A NEW TIME-DELAY COMPENSATING SCHEME FOR ELECTRO-HYDRAULIC SYSTEMS

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Abstract:

Operating time delay in the solenoid valves is very common for practical electro-hydraulic servo systems. It may sometimes cause performance degradation or even instability if it isn't treated carefully during control system designs. We propose in this paper a non-model-based design approach for hydraulic actuating systems based on a new time-delay compensation scheme. In the proposed system, two types of controllers are combined: a fuzzy-PID controller used to ensure primary tracking performance and an adaptable wavelet compensator used to compensate for the time delay resulting from the control valve. Performance of the proposed design is widely verified on a newly developed simulation platform to show its effectiveness.

Keywords: hydraulic system; position control; time-delay; fuzzy control; wavelet neural network

1 Introduction

Most of industrial of hydraulic systems use servo or proportional valves to interface electronic and power hydraulic systems. They are built by linear electrical motors or proportional solenoids. All these devices need better hydraulic platform, mechanical feedback or electrical sensors, to convert electrical signals and amplify in flow and pressure linearly. However, the developments of electronic machine, automation and robotics in agricultural and construction machinery need less expensive, robust hydraulic devices to interface power hydraulics (Malagut and Pregnolato, 2002). The traditional on/off solenoid is instead cheaper, robust, have higher power-mass ratio and could have small and large sizes. This is the reason why it was commonly adopted in practical applications (Gamble and Vaughan, 1996; Kajima and Kawamura, 1995; Malagut and Pregnolato, 2002; Rahman et al, 1996; Vaughan and Gamble, 1996).

Because of the magnetic hysteresis and the mechanical motion of the spool valve, the high speed solenoid valves usually have time delays. Considering the time-delay effect in a closed-loop system, it was pointed out in (Agrawal et al, 1999; Dorf and and Bishop, 1995; Huang and Wang, 2000) when the apparent delay time exceeded the dominant time constant of the system, the peak offsets, following a change in load, could approach those of the uncontrolled situation, even with the best PID tuning. In particular, the problem becomes conspicuous for the networked control system as time delay cannot be avoided as the information transmission on networks. The time delay problem was first tackled by Smith in 1957 with the focus put on the process control scheme. The primitive idea of the Smith predictor design is to cancel the time delay factor from the closed-loop characteristics equation (Huang and Wang, 2002; Majhi and Atherton, 1998; Zhang and Xu, 1999). However, success of the approach is heavily dependent on the known plant model which is, in general, inconvenient for the controlled plant with complicated characteristics.

In this paper, the hydraulic cylinder is operated by pulse width modulation (PWM) driven solenoid valves. Our design concept is initiated by the traditional Smith predictor in process system control which avoids the necessity of lowering the control gain to maintain stability due to the presence of time delay. We propose a new control scheme to compensate for the time-delay phenomenon which may occur in the control of hydraulic systems, especially when the hydraulic systems were setup in the networked control environment as the terminal devices. There were papers which presented

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strategies of auto-tuning PID controller based on the dynamic neural network to construct a Smith predictor (Cominos and Munro, 2002; Yonghong and Keyser, 1994; Zhang et al, 1996). However, in the current stage, the fuzzy logic-based control designs are still superior from either the viewpoint of reliability or implementability, especially for industrial applications. Our objective here is to develop a non-model-based control design methodology which ensures positioning accuracy while there were time delays and modeling uncertainties. A fuzzy-PID-PWM controller is used to provide primarily tracking control operations. Although a recurrent neural network can also identify the nonlinear plant due to their capability of mapping arbitrary continuous nonlinearities, the slow convergence is a serious issue (Yonghong and Keyser, 1994). To be more responsive for hydraulic systems, the basis function of neural network must react to the violent variation fast enough. A wavelet basis function neural network compensator is thus applied here as a time-delay compensator. Recently, Wavelet basis function networks were used for adaptive control and nonlinear systems (Daubechise, 1990; Jinhua and Ho, 1999; Mallet, 1984; Zhang et al, 1995). The efficacy of this type of network incorporated with the Smith predictor offers satisfactory time-delay compensation for the complicated plant. Furthermore, the proposed approach can, to some extent, offer the effect of disturbance rejection. Extensive studies presented here confirm effectiveness and feasibility of the proposed approach. Performance robustness of the hydraulic control system is examined as well.

2 Hydraulic System Modelling

A hydraulic system under consideration consists of the fluid supply unit and actuator, see the hydraulic cylinder shown in Fig. 1 (MATLAB, 1998). The actuator consists of a cylinder and a PWM hydraulic valve where p_s is supply pressure, Q is the oil flow directed from the valve to the actuator, p_c is control pressure, y is piston position, A_p is piston areas of the hydraulic cylinder, K_{sp} is spring rate. The PWM duty cycle is used for which the valve supplies oil to the control pressure. The pulse duty cycle shows the ratio of input signal pulse width to pulse cycle. The frequency of modulation is limited by the operating speed of the valve. To achieve acceptable control, the valve must be cycled very fast. Then, the valve vents p_c to exhaust. The oil flow Q controls p_c, which develops an actuating force against the piston. This forces the spring-loaded piston to its position y to track the reference input r_{in}.

Figure 2 shows the cross-sectional view of a typical solenoid valve. The solenoid is a pull-type device that pulls the armature into the coil when the coil is energized, thus shifting the ball of the valve. The armature and pole come together and the pressure force shuttles the ball to open the supply port and block the exhaust port. If there is no current, the internal spring forces the armature and ball to the right against the hydraulic force.

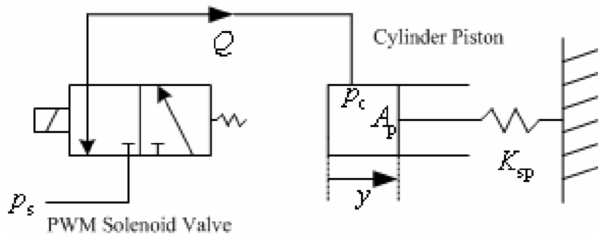

Fig. 1: *Hydraulic mechanism*

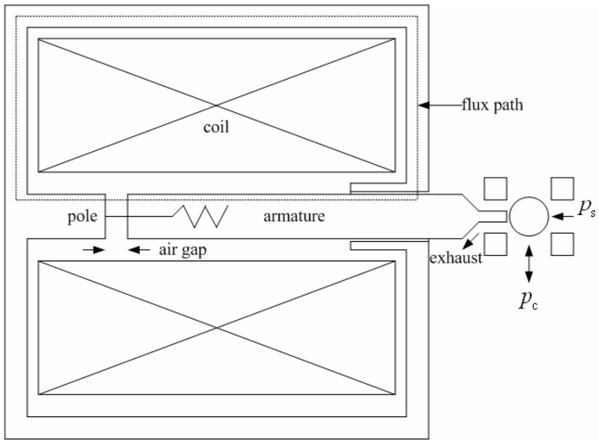

Fig. 2: *Theoretically modeled hydraulic system*

For the magnetic circuit we assume that leakage flux is negligible. The flux is determined by

$$\dot{\phi} = \frac{v_{sol} - iR}{N} \tag{1}$$

where ϕ is flux, v_{sol} is solenoid voltage, i is current, R is winding resistance and N is number of winding turns. The magnetomotive force required to develop this flux is broken up into components for the steel and the air gap. The magnetic circuit is characterized by

$$MMF = H_{air}g + H_{steel}L_{steel} \tag{2}$$

where MMF is magnetomotive force, H is magnetic field intensity, g is length of air gap and L_{steel} is magnetic circuit length in steel. Within the steel, the flux density B is a nonlinear function of H, dependent upon the material properties. The cross-sectional area A at air gap relates ϕ and B at the air gap, applies uniformly for the steel path.

$$B = \frac{\phi}{A} = f(H_{steel}) = \mu_0 H_{air} \tag{3}$$

where μ_0 is permeability of air. The electromechanical force F_e is determined by

$$F_e = \frac{1}{2\mu_0} B^2 A$$

The current i is given by

$$i = \frac{MMF}{N}$$

The hydraulic electrovalve, made mainly by solenoid and valve ball, is characterized by the typical relationship of mechanical dynamics:

$$m\ddot{x} + K_s x + C_v \dot{x} = F_e + A_0 p_s \tag{4}$$

where x is armature position, m is mass, A_0 is supply orifice area, p_s is supply pressure, K_s is spring rate and C_v is damping rate. If there is no current, the inner spring forces the armature and ball to the right against the hydraulic force. It obstructs the supply pressure p_s, and opens a path from the pressure to exhaust.

The oil flow directed from the valve to the actuator Q is the inlet flow less the outlet flow:

$$Q = Q_i - Q_o \tag{5}$$

where

$$Q_i = \begin{cases} K_0 A_0 \operatorname{sgn}(p_s - p_c)\sqrt{|p_s - p_c|}, & x > 0 \\ 0, & x = 0 \end{cases}$$

$$Q_o = \begin{cases} K_0 A_0 \sqrt{p_c}, & x < d \\ 0, & x = d \end{cases}$$

in which p_c is control pressure, K_0 is flow coefficient and d is ball travel.

The actuator moves the piston against a spring as a function of the control pressure developed behind it. Assuming negligible leakage, the actuator's motion equation is

$$\dot{p}_c = \frac{\beta_f}{V}(Q - \dot{y}A_p)$$

$$M_p \ddot{y} = p_c A_p - K_{sp} y \tag{6}$$

where β_f is fluid bulk modulus, V is fluid volume, y is piston position, A_p is area of piston, M_p is net actuator mass and K_{sp} is spring rate.

3 Tracking Control Design

The class of PID controllers is most widely used in industry due to its simple structure and easy of design. Therefore, good control techniques and algorithms for tuning the PID control gains are particularly important in industrial control systems. However, traditional control techniques and algorithms for tuning the PID gains are operated off-line and hence they are unable to adapt to plant or environment variations.

A fuzzy control system is a real-time expert system, implementing a part of process engineer's expert, preserves the simple control structure and doesn't demand the mathematical modeling knowledge. This motivates a fuzzy-PID control scheme described in the follows.

Conventional PID controllers need to regulate three variables, i.e. proportional, integral and derivative gains. It is usually desirable in practice to design a controller by using only the error and error change rate as its inputs to simplify the design work so that it will have the fine characteristics of a PID controller. A modified PID control scheme is proposed which preserves advantages of the traditional PID controller:

$$\begin{aligned} u &= \alpha(K_1 e + K_2 se) + \tfrac{1}{s}\beta(K_1 e + K_2 se) \\ &= (k_P + k_I \tfrac{1}{s} + k_D s)e \end{aligned} \tag{7}$$

where
$k_P = \alpha K_1 + \beta K_2$, $k_I = \beta K_1$, $k_D = \alpha K_2$

The two parameters $\alpha > 0$ and $\beta > 0$ are viewed as the weighting factor while choosing for the controller's type.

In the present control scheme, the fuzzy-PID controller is used to regulate the duty cycle. Therefore, Eq. 7 is actually replaced by the following control law:

$$\text{duty\_cycle} = \alpha\left[u'(e,\dot{e}) + \kappa \int u'(e,\dot{e})dt)\right] \tag{8}$$

where $u' = K_1 e + K_2 \dot{e}$ is the PD control command, $\kappa = \beta/\alpha$, $e = r_{in} - y$, i.e. the fuzzy-PID controller is applied to modulate the duty cycle. The solenoid driving circuit uses the computed duty cycle to generate the PWM waveform. Then the solenoid voltage is applied in order to achieve the desired current, force and valve flow.

For the convenience of design set κ in Eq. 8 to be constant. It is usually desirable in practice to design a controller by using only the error and error change rate as its inputs. The fuzzy logic system is used to infer the variables u' and α. The first output u' is control command, and the second output is used to tune the parameter α. Increasing α is equivalent to speed up the transient response of the fuzzy-PID controller. The linguistic values are expressed by linguistic sets as shown in Fig. 3. A rule table analogous to the standard PD-type rule table is constructed in Table 1.

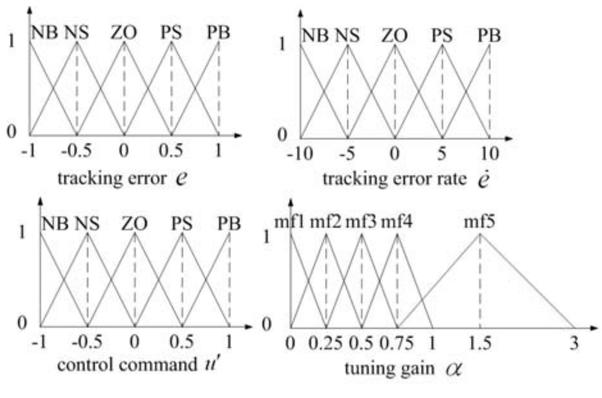

Fig. 3: *Membership functions for the (a) the tracking error e, (b) tracking error rate $\dot{e}$, (c) control command u', and (d) tuning gain α*

From the above-mentioned inference, the fuzzy rule table can be divided in three groups: In Groups 1 and 3, the difference between position output and reference input is big, and control commands are intended to speed up the approach to the reference position. In Group 2, the difference is small or zero, and control commands are set to be small or close to zero and are intended to correct small deviations from the reference position. The control scheme is managed to keep the difference between the controlled output and desired setpoint as small as possible.

It should be particularly emphasized that the previous mathematical model describing for the dynamic behavior of the hydraulic system is used for the simulation purpose alone. As it will be seen from the subsequent development that the present approach is not model based, the model information required is the valve delay time, however, its value can be experimentally measure or provided by the manufactory.

4 Time-Delay Compensation

A Smith predictor is a useful technique for better control of process systems with delay times. Suppose that the time-delay plant $G_p(s)$ containing the delay factor $e^{-\tau s}$ is described by

$$G_p(s) = G(s)e^{-\tau s} \tag{8}$$

The compensation network applied to reduce the influence of the delay time is shown in Fig. 4. For the closed-loop system, we get

$$\frac{Y(s)}{Y_r(s)} = \frac{G_c(s)G_p(s)}{1+G_c(s)G_p(s)+G_c(s)G_s(s)-G_c(s)G_s(s)e^{-\tau s}} \tag{10}$$

where $G_s(s)$ is the time-delay compensator. If τ and $G(s)$ are known precisely and $G_s(s)$ is precisely chosen to be $G(s)$ then Eq. 10 simplifies to

$$\frac{Y(s)}{Y_r(s)} = \frac{G_c(s)G(s)e^{-\tau s}}{1+G_c(s)G(s)} \tag{11}$$

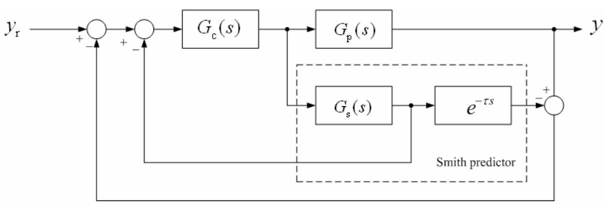

Fig. 4: *Conventional control system with time-delay compensation*

Clearly, the closed-loop stability won't be influenced by the time-delay effect. However, a major drawback of this scheme is that it cannot be applied to compensate for nonlinear systems or systems that aren't exactly modeled. Since a neural network can tune the connecting weights via a learning process to export ideal control signals, it could be applied to replace the predictor and resolve the problem. Due to similarity between wavelet decomposition and neural networks, combining wavelets with neural networks can hopefully remedy the weakness of each other. Because wavelet transformation has the ability of representing a function and revealing the properties of the function in the localized regions, the wavelet neural network could adjust the area by choosing the scale parameter and shift parameter. For the fast training of wavelet neural network, it is extremely suitable to identify the nonlinear systems.

To proceed, let us briefly introduce the structure of a wavelet neural network (WNN). The hidden layer performs nonlinear transformation via the activation function of each unit of the layer. And the function $\psi_{ab}(x)$ is the compact supported non-orthogonal function.

A wavelet transform W with respect to the function $f(x)$ can be mathematically expressed as (Daubechise, 1990; Jinhua and Ho, 1999; Mallet, 1984)

$$Wf = |a|^{-\frac{1}{2}} \int_{-\infty}^{\infty} f(x)\psi(\frac{x-b}{a})dx = \langle f(x), \psi_{ab}(x) \rangle \tag{12}$$

where a and b are, respectively, the dilation and translation factors, and

$$\psi_{ab}(x) = |a|^{-\frac{1}{2}} \psi(\frac{x-b}{a}) \tag{13}$$

The fundamental waveform of the wavelet function is called Mexican Hat, which was derived from the second derivative of Gaussian function $e^{-x^2/2}$. The function has excellent localization in time and frequency. The basis chosen here is simple and appropriate for industry control. If a and b are discrete numbers then this is called the discrete wavelet transform. It is convenient for choosing $a = 2^{-i}$, $b = aj$, i and $j \in Z$ then

$$Wf_{ij} = 2^{\frac{i}{2}} \int f(x)\psi(2^i x - j)dx = \langle f(x), \psi_{ij}(x) \rangle \tag{14}$$

where $\psi_{ij}(x) = 2^{\frac{i}{2}}\psi(2^i x - j)$.

The wavelet series expansion of $f(x)$ can be expressed as

$$f(x) = \sum_i \sum_j \theta_{ij}\psi_{ij}(x) \tag{15}$$

where $\theta_{ij} = \int_{-\infty}^{\infty} f(x)\psi_{ij}(x)dx = \langle f(x), \psi_{ij}(x) \rangle$. In theory, we could use the following wavelet function neural network to approximate a function $f(x)$ with arbitrary accuracy:

$$\hat{f}(x,\theta) = \sum_{i=M_1}^{M_m} \sum_{j=N_1}^{N_n} \theta_{ij}\psi_{ij}(x) = \theta^T W(x) \tag{16}$$

where $M_m, N_n \in Z$. The degree of approximation accuracy depends on the values of M_m and N_n chosen. Based on the Smith predictor, a WNN is used as a modified time-delay compensator, see Fig. 5. For the positioning control problem, we define the predicted position error as $\tilde{e}(t_k) = y(t_k) - \hat{y}(t_k)$ with $t_k = kT$ and T being the sampling period. We seek to determine the WNN as follows

$$NN_1(x,\theta_1) = \sum_{i=M_1}^{M_m} \sum_{j=N_1}^{N_n} \theta_{1ij}\psi_{ij}(c^T x) = \theta_1^T W_1(c^T x) \tag{17}$$

where $x(t_k) = \begin{bmatrix} u(t_k) & y(t_k) & y(t_{k-1}) & \cdots & y(t_{k-m}) \end{bmatrix}^T$ so that the instantaneous error cost $E(t_k) = 0.5\tilde{e}^2(t_k)$ is minimized with respect to the network parameter vector θ_1. It has been well known that if the network structure is sufficiently large, the activation function can approximate any continuous function within an arbitrary accuracy, i.e.

$$\lim_{M_i, N_j \to \infty} E^{\theta_1}_{M_i, N_j} = 0 \tag{18}$$

where

$$\sum_{i=1}^{m}\sum_{j=1}^{n} E^{\theta_1}_{M_i, N_j} = \frac{1}{2}\int_0^1 \cdots \int_0^1 [f(x) - \hat{f}(x)]^2 dx \tag{19}$$

To determine the adjustable weights θ_1, the gradient algorithm and chain rule are adopted:

$$\Delta\theta_1(t_k) = -\eta\tilde{e}(t_k)\frac{\partial\tilde{e}(t_k)}{\partial\theta_1(t_k)} \tag{20}$$

where $\Delta\theta_1(t_k) = \theta_1(t_{k+1}) - \theta_1(t_k)$ and

$$\begin{aligned}\frac{\partial\tilde{e}(t_k)}{\partial\theta_1(t_k)} &= \frac{\partial\left[y(t_k) - \hat{y}(t_k)\right]}{\partial\theta_1(t_k)} \\ &= \frac{-1}{1+P(t_k)}\left(P(t_k)\frac{\partial u_N(t_k)}{\partial\theta_1(t_k)} + \frac{\partial\hat{y}(t_k - \tau)}{\partial\theta_1(t_k)}\right) \\ &= \frac{-1}{1+P(t_k)}\left(P(t_k)W_1(c^T x(t_k)) + W_1(c^T x(t_{k\text{-}N}))\right)\end{aligned}$$

in which $P(t_k) = \dfrac{\partial y(t_k)}{\partial u_c(t_k)}\dfrac{\partial u_c(t_k)}{\partial e(t_k)}$, $e(t_k) = y_r(t_k) - \tilde{e}(t_k) - u_N(x(t_k), \theta_1(t_k))$ with $u_N(x, \theta_1) = NN_1(x, \theta_1)$; $\tau = NT$, N is an integer constant; η is the learning rate satisfying $0 < \eta < 1$. For the WNN, the PWM signal $u_p(t_i)$ and the measured cylinder position $y(t_i)$, $i = k, k\text{-}1, \ldots k\text{-}m$, are used as the input variables.

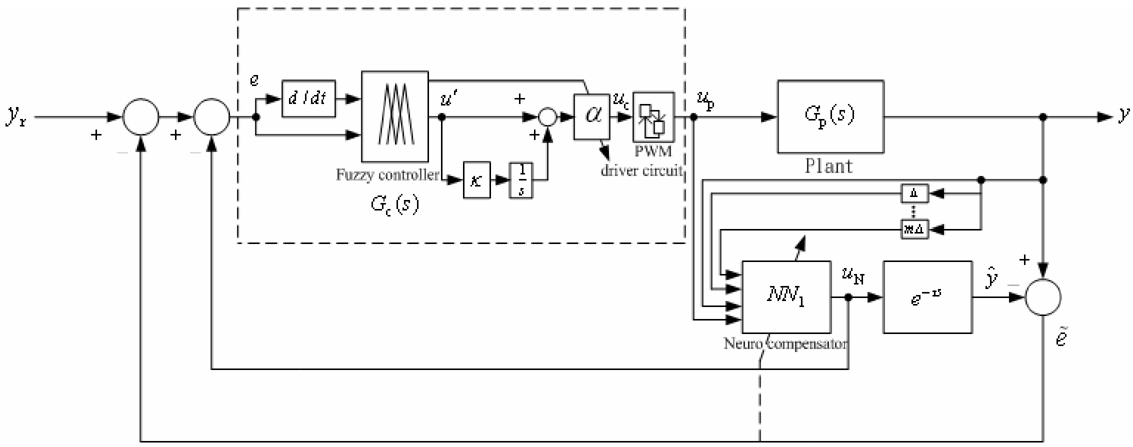

Fig. 5: *Proposed control scheme*

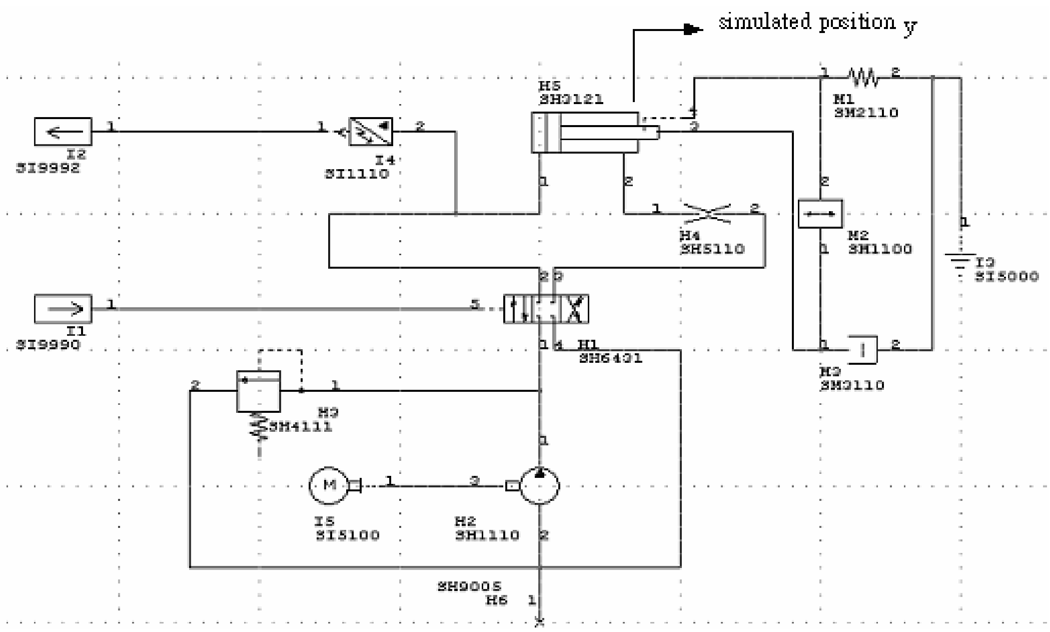

Fig. 6: *Configuration of the hydraulic system* $G_p(s)$

The proposed control scheme not only compensates for delay-time but also contributes to the robustness of tracking performance. Referring to Eq. 10, the wavelet compensator first adaptively adjusts its weights to provide the best cancellation to the perturbed plant. With reference to Eq. 11, we can know that tracking performance is further dominated by the fuzzy-PID controller. Embedded with the versatile characteristics, the compensator ensures satisfactory stability and performance of the hydraulic control system.

5 Demonstration and Verification

The schematic diagram for the hydraulic system $G_p(s)$ in Fig. 5 is realized in Fig. 6, in which I1 is the generated PWM signal, I2 is the measured pressure signal, I3 is the ground reference, I4 is the sensor, I5 is the motor, H1 is a four-way two-position solenoid valve, H2 is the pump, H3 is the relief valve, H4 is the flow valve, H5 is a linear double-acting cylinder, H6 is a tank, M1, M2 and M3 indicate, respectively, the load, spring and damper. To initiate the operation of the system, the pump speed went from zero to 1800 rpm for a step function. The performance and effectiveness of our proposed design scheme are fully confirmed in this newly developed simulation platform.

For the convenience of checking feasibility of the presented design, the reference input r_{in} was set to be a unit step command. Performance of the primary design with no time-delay compensation was first examined. The controlled system was required to respond to the step input with rise time less than 0.5 sec and 2% settling time less than 2 sec, less than 3% overshoot and zero steady state error. The goal was achieved by adjusting membership functions and fuzzy control rules in the fuzzy PID control design. Figure 7 shows the normalized transient response of the proposed PWM electro-hydraulic servo control system without time delay. The PWM carrier signal is displayed in Fig. 8. These show that the resulting response meets the specifications.

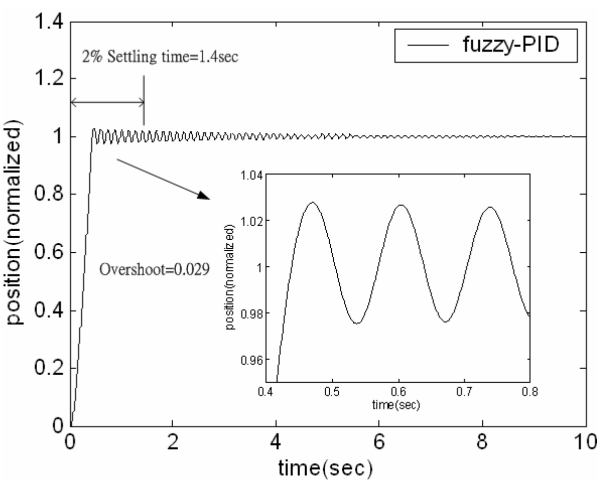

Fig. 7: *Positioning response to the step input*

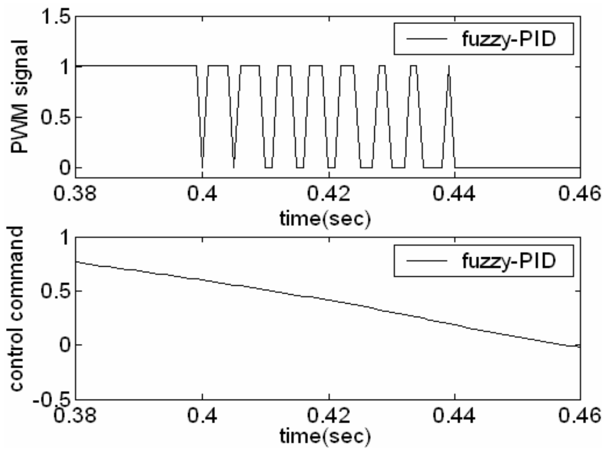

Fig. 8: *Generated PWM carrier signal of the fuzzy-PID controller*

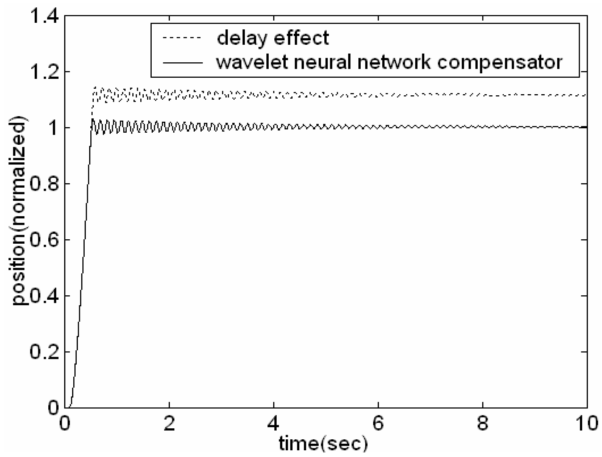

Fig. 9: *Effect of WNN compensation for the delay time $\tau = 0.1$ sec (dashed and solid lines represent, respectively, transient responses without (only with fuzzy-PID control) and with time-delay compensation)*

Next consider time-delay effect of the control valve and its influence to the closed-loop response. Figure 9 shows that tracking performance of the system without time-delay compensation may deviate from the desired steady state when the delay time τ exceeds 0.1 sec. It's also found that if one intentionally increases the value of τ, the response may even gradually diverge. The phenomenon goes conspicuously when the delay time becomes larger. If there is no time-delay compensation, the dashed line in the figure shows that the settling time (2%) is 7.3 sec, and the overshoot reaches 15%. This confirms assertion described in Section IV that explicit time delay may lower the phase margin and degrade the system performance. The system with the time-delay compensator does improve transient responses. For hydraulic control systems, the delay time up to 1 sec is practically unrealistic. However, if the hydraulic system is connected with a remote controller via internet or intranet, the situation would probably be occurred. Under the situation, the time delay effect should be taken into serious consideration.

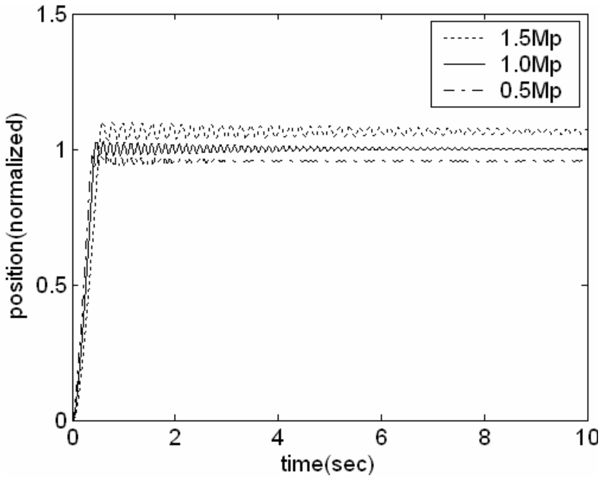

Fig. 10: *Transient responses of the closed-loop system with different loads and without WNN compensation*

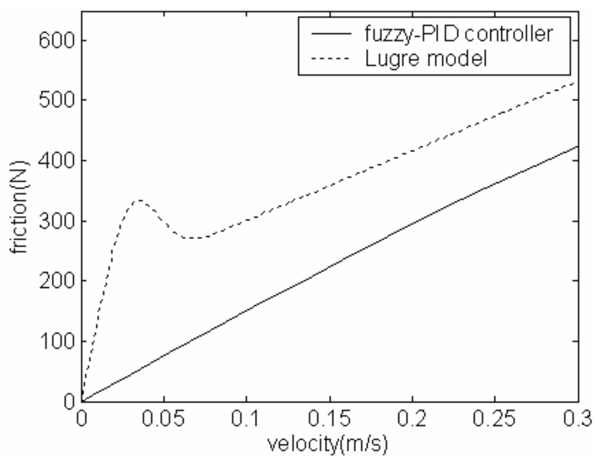

Fig. 12a: *Velocity versus LuGre friction (a) with and without fuzzy-PID control*

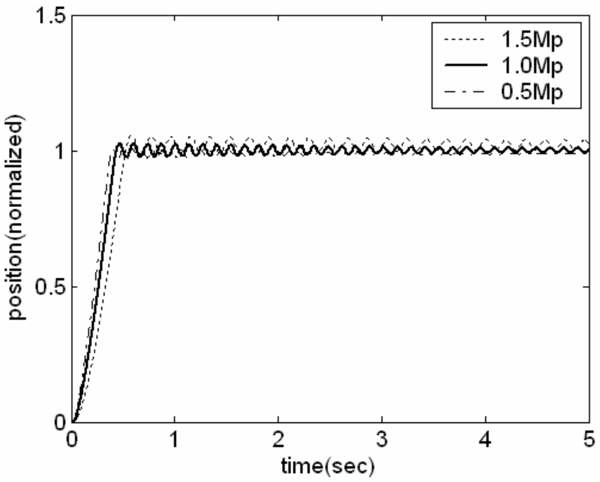

Fig. 11: *Test of performance robustness for the closed-loop system with WNN compensation*

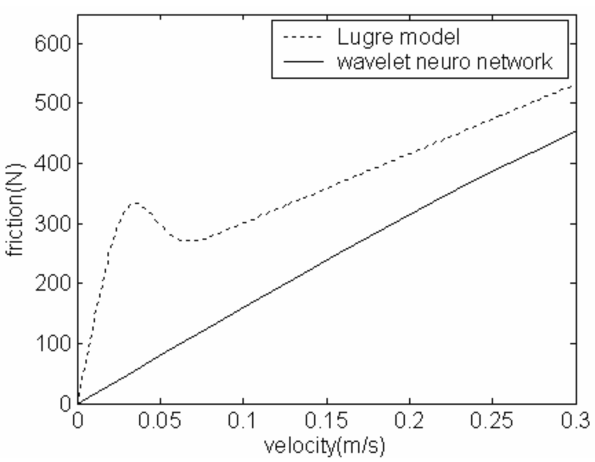

Fig. 12b: *Velocity versus LuGre friction (b) with and without WNN compensation*

We proceed to examine performance robustness of our proposed feedback control scheme. Figures 10 and 11 show transient responses of the closed-loop system with different loads ($M_p = 1.0$ kg , $0.5M_p$, and $1.5M_p$). Figure 12(a) shows velocity versus the LuGre friction with and without the fuzzy-PID controller, and Fig. 12(b) shows velocity versus friction with and without wavelet neural compensator. Because of the stribeck region, the curve of LuGre model is nonlinear without using the fuzzy-PID controller and wavelet neural compensator. Simulations have shown that the fuzzy-PID controller and wavelet neural compensator linearize the curve of velocity versus LuGre friction. These further demonstrate superiority of the present design.

6 Conclusions

An effective time-delay compensator design for the PWM electro-hydraulic servomechanism is proposed. An integrated fuzzy-neuro control scheme is constructed in which a modified fuzzy-PID controller is adopted to ensure tracking performance and a WNN is proposed to compensate for the time-delay effect in the solenoid valve to avoid performance degradation and probable instability. It is shown that the incorporated adaptable neural network ensures performance robustness while there are physical variations.

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