An Investigation into the PID Control for the Electro-Hydraulic Servo System of Skin Pass Mill

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ABSTRACT

This study is to investigate the problem of the SPM(Skin Pass Mill) system which is a finishing treatment of steel sheet, and to develop a PID control scheme to minimize process instability. An electrohydraulic servo system with conventional proportional controller used to regulate the force on the strip works inadequately to yield very undesirable transient responses at the moments welding parts of the strip come into and pass through the rolls. Both linearized and nonlinear models of a typical SPM system were simulated first by using Simulink. Then Ziegler-Nichols ultimate cycling method was used for an initial reference guide to tune PID gains, and further fine tuning was performed to get a desirable response. The test result in the plant show that proposed PID control scheme successfully improves the process instability in a SPM system.

Keywords : SPM(Skin Pass Mill), PID, Ziegler-Nichols method, Electrohydraulic servo

Nomenclature

A_p, A_r : Cylinder cap area, and cap area subtracted by rod area respectively
B : Viscous damping coefficient of piston and load
F : Acting force against the load
G_c : Controller gain function
H : Sensitivity of load cell
K : Flow-pressure coefficient of a servovalve
K_f : Load spring constant
K_c, K_{cp} : Linearized flow coefficients of a servovalve
K_p, K_i, K_d : Proportional, integral and derivative gain coefficients of PID controller respectively
K_q, K_{qp} : Flow gains of a servovalve
L_m : Internal leakage coefficient of piston
M : Total mass of piston and load
P_1, P_2 : Pressure at forward and return chamber respectively
P_s : Supply pressure into a servovalve
Q_1, Q_2 : Flow rates into forward and out of return chamber respectively
V_1, V_2 : Volume of forward and return chamber respectively
W : Weight of load
k_v : Gain factor of a servovalve
x_o, x_t : piston position at initial operation and total piston stroke respectively
x_v, x_p : Servovalve spool displacement and piston rod displacement respectively
u : Input current into a servovalve
β : Bulk modulus of hydraulic oil
ω : Servovalve frequency
ξ : Damping constant of a servovalve

1. Introduction

The SPM, also called superficial rolling, is a finishing treatment of steel sheet. It consists of a light roll pass made by two high mills as shown on the schematic diagram of Fig. 1. Fig. 2 is the schematic of the hydraulic servo control for SPM in POSCO(Pohang Iron & Steel Co., LTD), Korea.

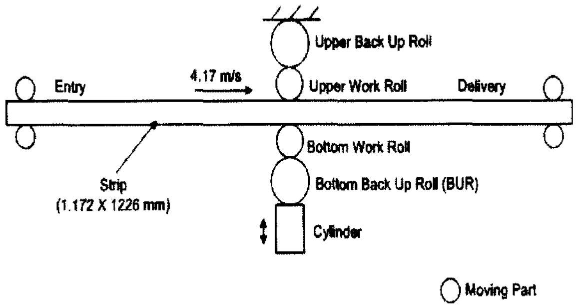


Fig. 1 Schematic Diagram of SPM

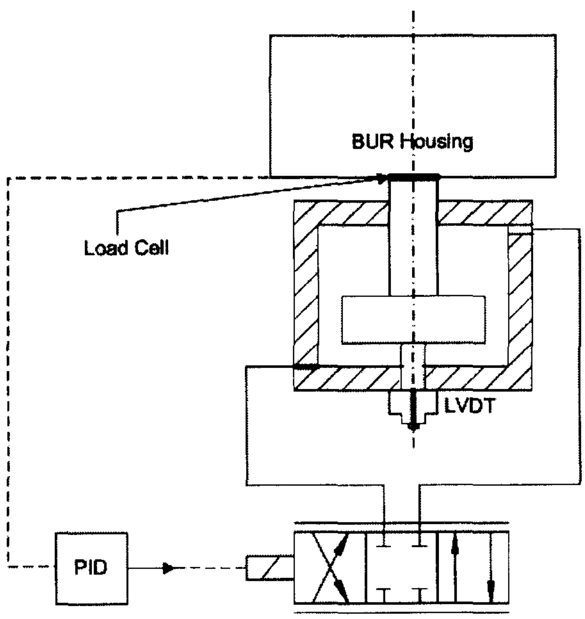


Fig. 2 Hydraulic servo control for SPM

The aims of SPM are suppression of vermicular type of defects during drawing operation, sheet surface finish marks due to rolls surface roughness, improvement of strip flatness, and increase of yield point and tensile strength of the strip. The purpose of the electrohydraulic servo system in SPM is to control the force of the hydraulic cylinders, that is, the force on the strip. The hydraulic servo system provides a selected constant force on the strip during the process except when the welding parts of the strip come into the rolls. When welding parts of the strip come into the rolls, operators usually reduce the pressing force (82 ton to 62 ton) to avoid tearing the strip, and return to the original set value after the welding parts pass through.

A conventional proportional controller to control the force on the strip works adequately during the set point control. However when the control force variates, undesirable transient responses occur frequently. A large overshoot and oscillatory transient responses cause a "pinch tree" which in turn let the strip to stuck between the rolls. The cost of damage resulting from the failure of SPM process is about $5,000 per minute.

The objective of this study is to investigate the problem in SPM and provide a practically useful control scheme to minimize the process instability. Many advanced control schemes have been theoretically investigated for the application into steel processes such as optimal LQ control[1], Fuzzy controllers[2,3], preview control[4], and H_∞ control[5]. However PID control schemes are still most widely used in process industries up to dates, because PID control has functional simplicity and robust performance characteristics in a wide range of operating conditions and processes.

Over the years, many methods have been developed to tune PID controllers: Ziegler-Nichols tuning formula[6-7], Cohen-Coon[8], integral of time absolute error optimum (ITAE)[9], frequency response techniques[10,11], s-plane root locus method[11], dominant pole design method[12], and some self-tuning algorithms[13,14,15]. However, these methods provide basic tools and not completely satisfactory answers for any kind of real system yet, because most of those formulas are for low-order, over-damped step response systems and the selection of the three coefficients of PID controller gains is a search problem in a three-dimensional space. In this study, Ziegler-Nichols ultimate cycling method was chosen for an initial reference guide. Then further fine tuning was performed to get a desirable response.

A mathematical model of SPM force control system was obtained first by using the expression for the dynamics of an electrohydraulic servo system and experimental data of a real SPM plant in the following manner.

2. System Modeling

The valve-piston combination in a SPM is shown schematically in Fig. 3.

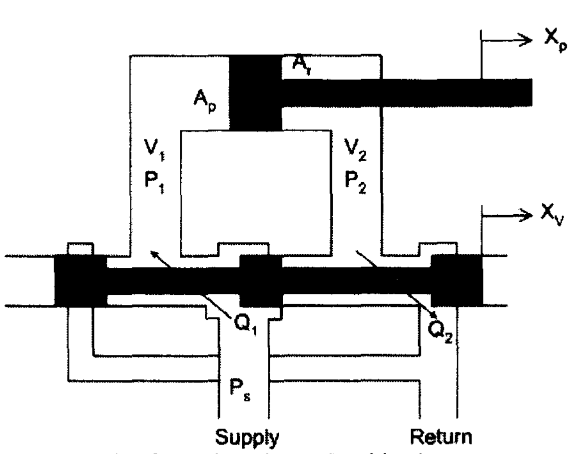


Fig. 3 Valve-Piston Combination

The first stage dynamics of torque motor with flapper-nozzle for an electohydraulic servovalve can be represented by second-order differential equation as follows[16]:

$$\frac{d^2 x_v}{d t^2} + 2\xi\omega \frac{d x_v}{dt} + \omega^2 x_v = k_v \omega^2 u \tag{1}$$

Suppose the spool orifices are symmetrical. Then, the flow rates are given by,

$$Q_1 = K x_v \sqrt{P_s - P_1}, \quad Q_2 = K x_v \sqrt{P_2} \tag{2}$$

Eq. (2) can be linearized for the small variation from the null position of the valve as follows:

$$\delta Q_1 = K_q \delta x_v - K_c \delta P_1, \quad \delta Q_1 = K_{qp} \delta x_v - K_{cp} \delta P_1 \tag{3}$$

Applying the continuity equation to each of the piston chamber yields

$$\delta Q_1 = L_m (P_1 - P_2) + \frac{V_1}{\beta}\frac{d P_1}{dt} + A_p \frac{d x_p}{dt} \tag{4}$$

$$\delta Q_2 = L_m (P_1 - P_2) - \frac{V_2}{\beta}\frac{d P_2}{dt} + A_r \frac{d x_p}{dt} \tag{5}$$

The force balance equation of the piston rod is represented by,

$$F = A_P P_1 - A_r P_2 = M\frac{d^2 x_p}{d t^2} + B\frac{d x_p}{dt} + K_f x_p + W \tag{6}$$

Then, in order to derive simplified linear transfer function, the terms of right-hand side of Eq. (6) was ignored. Because the piston displacement is very small (micrometer scale) for this particular force control system of SPM, and the load could be often omitted in system design so that only the transfer function from input is of interest. In addition, K_c and K_{cp} of Eq. (3) are also neglected, in other words we consider just flow gains, to get simplified transfer function. Then after extensive manipulations using Eqs. (1), (3)-(6), the open-loop system transfer function between the valve current u(t) and acting force F(t) on the strip is obtained as follows:

$$\frac{F}{U} = \frac{k_v \omega^2 \left[\left(\frac{A_p \beta K_q}{V_1} + \frac{A_r \beta K_{qp}}{V_2} \right) s + \frac{\beta^2 L_m}{V_1 V_2} (A_p - A_r)(K_q - K_{qp}) \right]}{s\left\{ s^3 + \left(2\xi\beta + \frac{\beta L_m}{V_1} + \frac{\beta L_m}{V_2} \right) s^2 + \left[2\xi\omega \left(\frac{\beta L_m}{V_1} + \frac{\beta L_m}{V_2} \right) + \omega^2 \right] s + \left(\frac{\beta L_m}{V_1} + \frac{\beta L_m}{V_2} \right) \omega^2 \right\}} \tag{7}$$

The sensitivity of load cell for the feedback gain of the system was $H = 2.5 \times 10^{-5}\ volt / kg_f$ and the parameter values measured in SPM plant at POSCO are summarized in Table 1. The dimensional parameters like the area, length and volume are measured. Internal leakage coefficient L_m was adopted from the typical value of Meritt [17]. Flow-pressure coefficient of servovalve K and flow gains K_q , K_{qp} were estimated by referring to hydraulic oil density and servovalve manufacturer's data of Rexroth[18]. Load mass M and spring constant K_f was measured by using the load cell and gap sensor in the SPM plant.

Table 1 Parameter Values for a SPM

Parameter	Measured Value	Parameter	Measured Value
A_p	2,043 cm^2	β	10,540 kg_f / cm^2
A_r	591 cm^2	P_s	250 kg_f / cm^2
x_o	10.7 cm	K_q	400.8 cm^3 / sec/ mA
x_i	16 cm	K_{qp}	115.9 cm^3 / sec/ mA
V_1	22,510 cm^3	V_2	3,500 cm^3
L_m	0.8951 cm^5 / kg_f sec	K	4,160 $cm^4 / kg_f^{1/2}$ sec
M	20,300 kg_m	K_f	1.3x10^[illegible] kg_f / cm
B	6.5x10^[illegible] kg_f sec/ cm	W	20 ton

Also the dynamics of servovalve of Eq. (1) between the input current, u, and second-stage spool displacement, x_v, was obtained at the serovovalve laboratory in Kwang-Yang Plant of POSCO as follows:

$$\frac{X_v}{U} = \frac{12}{s^2 + 130 s + 1200} \tag{8}$$

Then, based on the data in Table 1, the SPM control system of linear transfer function of Eq. (7) was represented on the block diagram as shown in Fig. 4. And, in order to get better results of the real system, nonlinear system model of the flow Eq. (2) was developed using Simulink as shown on Fig. 5.

The maximum input current of amplifier into the servovalve was saturated at $10mA$. In the simulation, uncertain parameter values of effective bulk modulus β

model of the flow Eq. (2) was developed using Simulink as shown on Fig. 5.

The maximum input current of amplifier into the servovalve was saturated at $10mA$. In the simulation, uncertain parameter values of effective bulk modulus β
and damping coefficient B were estimated as shown in Table 1 so that the shapes of the transient response is close to measured result of experiment. The open-loop system transfer function of Eq. 7 was obatined by using the parameter values in Table 1 at last as follows:

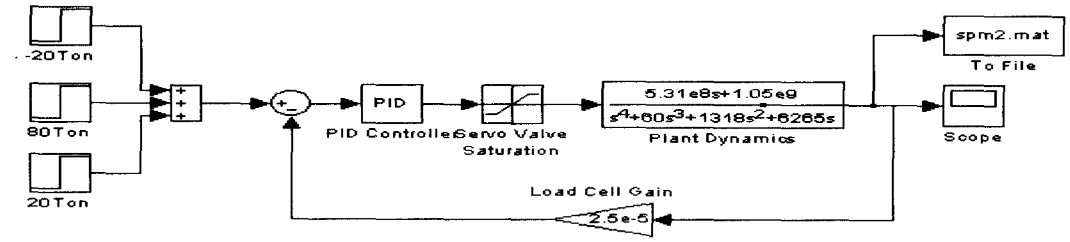


Fig. 4 Linearized Model for SPM Control System

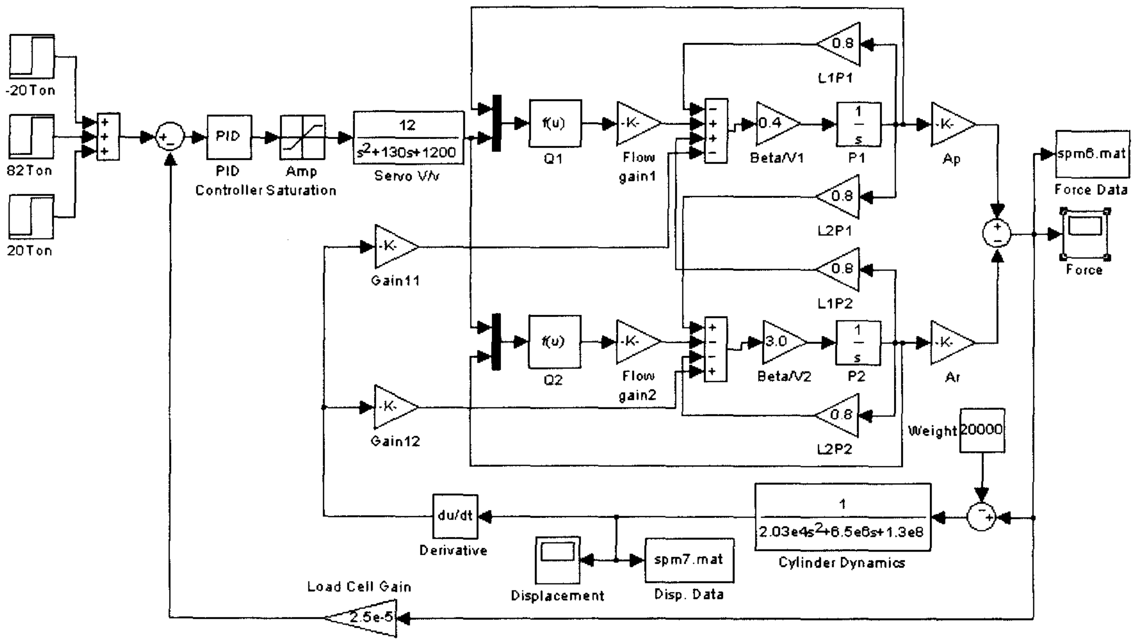


Fig. 5 Nonlinear Model of SPM Control System

$$\frac{F}{U}=\frac{5.31\times10^{8}s+1.05\times10^{9}}{s^{4}+60s^{3}+1{,}318s^{2}+6{,}265s} \qquad (9)$$

In order to validate the system model of Fig. 4 and Fig. 5, a test was conducted when the proportional gain of the amplifier was $K_p=4mA/volt$. Fig. 6 shows the comparison between the measured transient response of a real SPM plant and corresponding computer simulation results to a step input of $-20{,}000kg_f$ from the initial set force $82{,}000kg_f$. Fig. 6 reveals that the simulation results are quite close to the test result of SPM system.

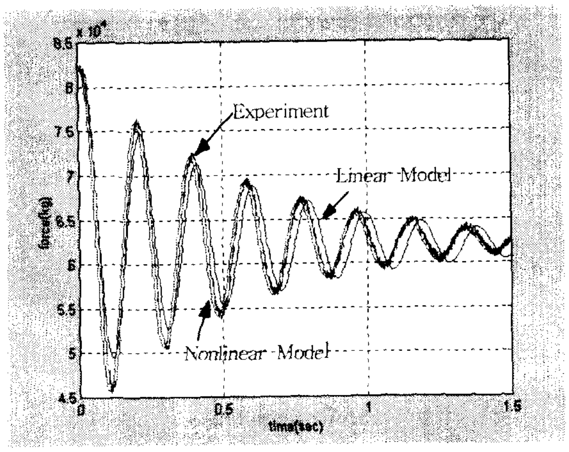


Fig. 6 Comparison of Test and Simulation Results ($K_p=4mA/volt$)

3. PID Controller Design

By considering the productivity and "pinch tree" problem caused by big overshoot, following performance specifications are required: percent overshoot < 5%; 2% settling time < 1 sec.; steady state error < 1%. The selection of the gain coefficients for a PID controller in $G_c(s)=K_p+K_i/s+K_d s$ is basically a search problem in a three-dimensional space. Ziegler-Nichols tuning method was applied to the nonlinear system model in Fig. 5 to find the coefficients to satisfy the specifications required in this study. In this case, "stability-limit" method was used, that is while the derivative and integral terms were initially set to zero, the proportional gain was increased from zero to a critical gain value, K_{cr}, where the closed-loop system exhibits sustained oscillations for a step input (0.5volts=20tons) as shown on Fig. 7.

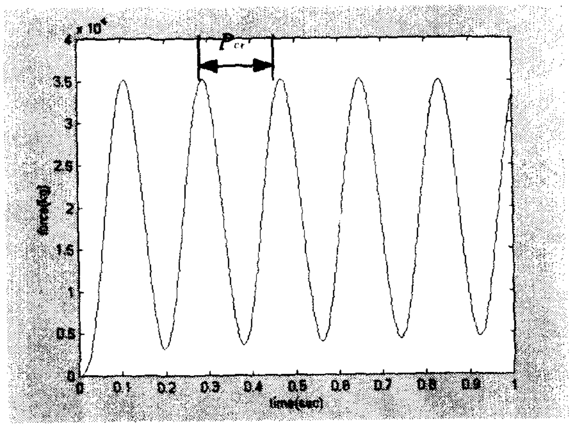


Fig. 7 Step response of SPM at marginally stable condition ($K_v=4.95mA/volt$)

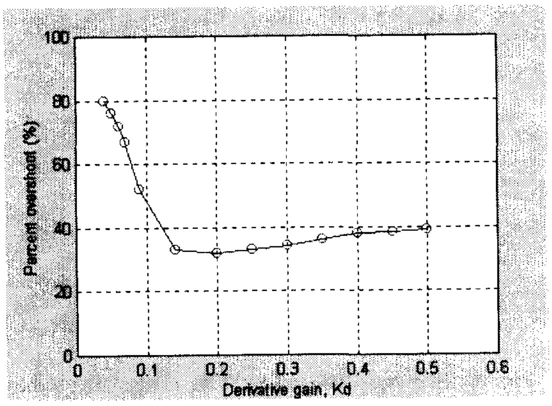
Using the value of $K_{cr}=4.95mA/volt$ and the period of oscillation, $P_{cr}=0.181sec$, the values of three PID coefficients were determined according to the following formulas (Ziegler and Nichols, 1942):

$$\begin{aligned} K_d &= K_p(0.125P_{cr}) = 0.067mA/volt \\ K_i &= K_p/(0.5P_{cr}) = 32.82mA/volt, \\ K_p &= 0.6K_{cr} = 2.97mA/volt, \end{aligned} \qquad (10)$$

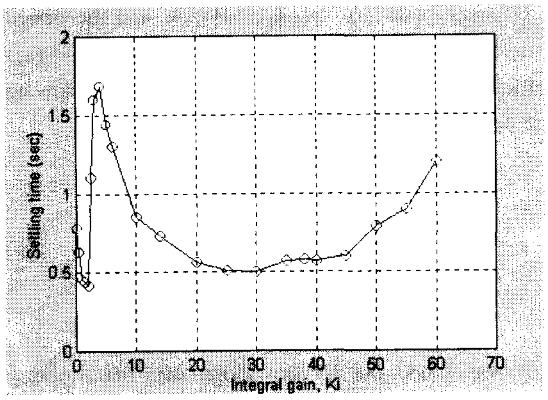
It was found out by simulation that the resulting response with controller of the gains in Eq. (10) for a step input (0.5volts=20tons) satisfies the requirement of settling time. However, percent overshoot was not improved at all. Thus the following tuning procedure for controller was necessarily proposed to obtain the desired response for this particular system.

First, the derivative gain was varied to find the optimal value yielding a minimum percent overshoot, while the proportional and integral gains obtained by the Ziegler-Nichols method were maintained. Then in a similar way, the optimal integral gain was searched for a minimum settling time. As shown on Fig. 8, the bounds of useful gains should be $0.15<K_d<0.25$, $0.8<K_i<2.0$. Using these values and tuning in the region around the proportional gain, three coefficients were determined in the following manner at last.

$$K_p = 3.0mA / volt, \quad K_i = 1.0mA / volt, \quad K_d = 0.2mA / volt \tag{11}$$



(a) Percent overshoot vs. derivative gain



(b) Settling time vs. integral gain

Fig. 8 Dynamic characteristics of various control gains

4. Validation Test

Based on the coefficients in Eq. (11), the PID controller in SPM system was tuned for the test. The reference control input into the system is initially constant 2.05 volts (equivalent to the 82 tons on the strip) and maintained until a welding part of the strip comes in. At the moment for the welding part to come into the rolls, the input signal decreases to 1.55 volts (reducing 20 tons of force) for a second. After one second, the input signal recovers to 2.05 volts.

The simulation results and test result are shown in Fig. 9. The results reveal that the performance of the system is greatly improved and satisfies the specified requirements by using the proposed PID control scheme. Also Fig. 9 shows that nonlinear simulation result was very close to the test result.

The PID control scheme proposed in this study was revealed to be practically acceptable in real plant.

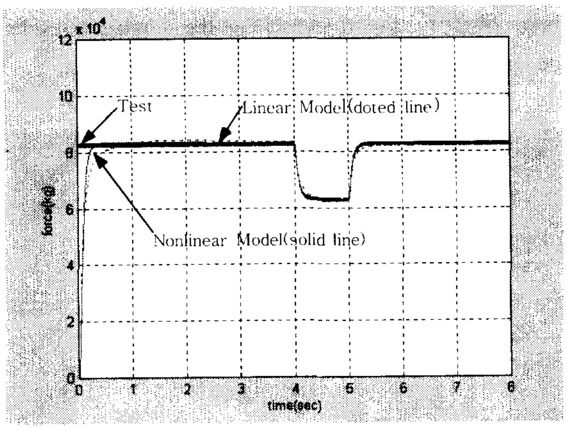


Fig. 9 Controlled responses of SPM system

5. Conclusion

A PID control scheme was proposed to prevent the "pinch tree' problem due to the instable transient response of the SPM system. To determine the three gain coefficients of the PID controller, Ziegler-Nichols ultimate cycling method was applied on the dynamics model of SPM for an initial reference guide. Then further fine tuning was performed to get a desirable response. The established PID gains by Simulink were implemented into the SPM plant for the test. The test revealed that the PID control scheme proposed in this study was practically acceptable, since it successfully satisfied the specified requirements for working of SPM system.

References

1. Kim, J. S., and Kim, C. M., 1994, "Noninteracting and Loop-shaping LQ Controller Design for Tandem Cold Mills," Transactions of the Korean Society of Mechanical Engineers, Vol. 18, No. 10., pp. 2629-2639.

2. Jung, J. Y. et al., 1994, "Simulation of Shape Control in Cold Rolling Using Fuzzy Control," Transactions of the Korean Society of Mechanical Engineers, Vol. 18, No. 2, pp. 302-312.
3. Jung, J. Y., and Im, Y. T., 1996, "Simulation of Fuzzy Shape Control for Cold-Rolled Strip with Randomly Irregular Strip Shape," Transactions of the Korean Society of Mechanical Engineers, Vol. 20, No. 3, pp. 861-871.
4. Kim, J. S. et al., 1995, "Digital Preview and Noninteracting Control of Tandem Cold Mills," Journal of the Korean Society of Precision Engineering, Vol. 12, No. 7, pp. 114-122.
5. Kim, J. S., and Kim, S. S., 1998, "Thickness Control of Tandem Cold Mills Using Control Techniques," Journal of the Korean Society of Precision Engineering, Vol. 15, No. 8, pp. 145-155.
6. Ziegler, J. G., and Nichols, N. B., 1942, "Optimum Settings for Automatic Controllers," Trans. ASME, Vol. 64, pp. 759-768.
7. Hang, C. C. et al., 1991, "Refinements of The Ziegler-Nichols Tuning Formula," Proc. IEE, Pt. D, Vol. 138, No. 2, pp. 111-118.
8. Cohen, G. H., and Coon, G. A., 1953, "Theoretical Consideration of Retarded Control," Trans. ASME, Vol. 75, pp. 827-834.
9. Simth, C. A., and Corripio, A. B., 1985, Principle and Practice of Automatic Process Control, Wiley.
10. Ho, W. K. et al., 1995, "Tuning of PID Controllers Based on Gain and Phase Margins Specifications," Automatica, Vol. 31, No. 3, pp. 497-502.
11. Dorf, B. C., and Bishop, R. H., 1995, Modern Control Systems, 7th ed., Addison Wesley.
12. Astrom, K. J., and Hagglund, T., 1995, PID Controllers: Theory, Design, and Tuning, Research Triangle Park, NC: ISA.
13. Astrom, K. J., and Hagglund T., 1984, "Automatic Tuning of Simple Regulators with Specifications on Phase and Amplitude Margins," Automatica, Vol. 20, No. 5, pp. 645-651.
14. Zhaung, M., and Atherton, D. P., 1993, "Automatic Tuning of Optimal PID Controllers," Proc. IEE, Pt. D, Vol. 140, No. 3, pp. 216-224.
15. Ho, W. K. et al., 1996, "Self-tuning PID Control of a Plant with Under-damped Response with Specifications on Gain and Phase Margins," IEEE Trans. on Control Applications, Vol. 5, No. 4, pp. 446-462.
16. Martin, D. J., and Burrows, C. R., 1976, "The Dynamic Characteristics of an Electrohydraulic Servovalve," Trans. ASME, J. Dynamic Systems, Measurement, Control, pp. 395-406.
17. Meritt, H. E., Hydraulic Control Systems, 1966, John Wiley & Sons, Inc., p. 142.
18. Rexroth, Mannesmann, Proportional and Servo- valves, Electronic Components, Mannesmann Rexroth GmbH, 1991.