Journal of Zhejig University SCIENCE A
ISSN 1673-565X (Print); ISSN 1862-1775 (Online)
www.zju.edu.cn/jzus; www.springerlink.com
E-mail: jzus@zju.edu.cn

Active disturbance rejection control for hydraulic width control system for rough mill*

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Received Dec. 14, 2006; revision accepted Feb. 14, 2007

Abstract: The highly nonlinear behavior of the system limits the performance of classical linear proportional and integral (PI) controllers used for hot rolling. An active disturbance rejection controller is proposed in this paper to deal with the nonlinear problem of hydraulic servo system in order to preserve fast response and small overshoot of control system. The active disturbance rejection (ADR) controller is composed of nonlinear tracking differentiator (TD), extended state observer (ESO) and nonlinear feedback (NF) law. An example of the hydraulic edger system case study is investigated to show the effectiveness and robustness of the proposed nonlinear controller, especially, in the circumstance of foreign disturbance and working condition variation, compared with classic PI controller.

Key words: Active disturbance rejection (ADR), Hydraulic servo system, Width control, Rough mill
doi:10.1631/jzus.2007.A1429 **Document code:** A **CLC number:** TH137

INTRODUCTION

Electrohydraulic servo systems, which can generate very high forces, exhibit rapid responses and have a high power to weight ratio compared with electrical counterparts, are widely used in many kinds of industrial processes, such as in rolling mills, power plants. It is well known that the dynamics of hydraulic systems is nonlinear. There are many factors such as flow characteristics, fluid compressibility and friction in the cylinders contributing to this nonlinear behavior. However, the traditional approach to control hydraulic servo systems is based on the local linearization of the nonlinear dynamics about a nominal operating point. Taking nonlinearity and time varying parameter into consideration, the traditional constant-gain controllers have become inadequate. As a result, many control methods such as robust control (Laval *et al*., 1996; Li and Khajepour, 2005), fuzzy control (Zhao *et al*., 1998; You *et al*., 2006), adaptive control (Yun and Cho, 1988; Sekhavat *et al*., 2006), state feedback control (Yu *et al*., 2004), immune control (Zou *et al*., 2005) and variable structure control (Bonchis *et al*., 2001), are used for hydraulic servo systems.

In hot strip mills, a heated slab is rolled in roughing mills and finishing trains, and cooled down on run-out tables and finally coiled at the coiler. In rough mills, it is necessary to control strip width and thickness at their target values. To control strip width, hydraulic edger screwdown systems are installed in front of the rolling stands and controlled by proportional and integral (PI) control method.

Precise width control can reduce the loss of strip trimming and increase the benefit-cost ratio. Therefore, it is necessary to renew the present control systems. One of the challenges is to refresh hydraulic edger screwdown systems to get better width control performance. Mainly the width gage fluctuation is caused by strip temperature deviation called skid marks. Slabs are placed on skids in reheated furnaces. Skids are cooled down by flowing water inside, so the

* Project supported by the National Basic Research Program (973) of China (No. 2006CB705400) and the National Natural Science Foundation of China (No. 50575200)

parts of a slab on skids are also cooled down. Because of skid marks, the rolling force fluctuates, during rolling process.

In this paper, the active disturbance rejection (ADR) control method (Han, 1998) is employed to develop a nonlinear controller for the hydraulic edger screwdown system in rough mill. An example of the hydraulic edger system case study is investigated to show the effectiveness and robustness of the proposed nonlinear controller.

DYNAMIC MODEL OF HYDRAULIC EDGER

The hydraulic edger system shown in Fig.1 consists essentially of an electrohydraulic servo valve, which controls fluid flow direction, a hydraulic cylinder, which output the force to impel the edger to roll the strip in the dimension of width, and a stand, on which the case of hydraulic cylinder is fixed firmly. A relief valve and an accumulator are used to maintain a steady pressure supply at the output of the pump.

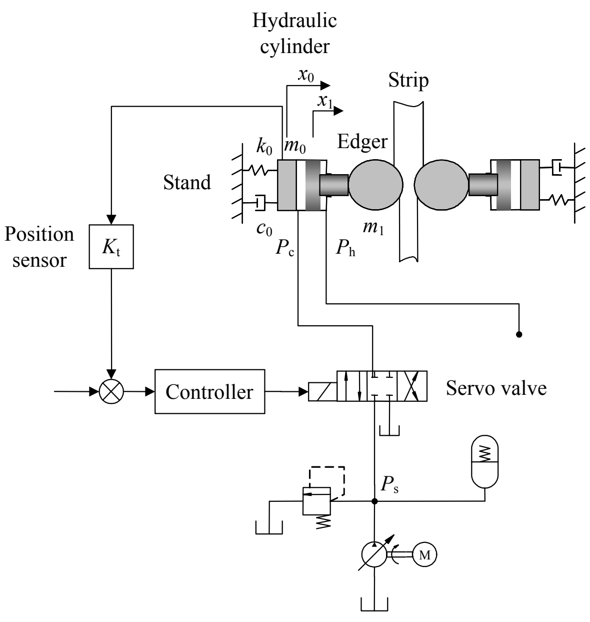

Fig.1 The diagram of the hydraulic edger control

The differential equation governing the dynamics of the hydraulic actuator is given in (Merrit, 1967). The control pressure dynamics are given by

$$\dot{p}_c = \frac{\beta_e}{V_c + A_c x_p}(q_v - A_c \dot{x}_p - c_l(p_c - p_h)), \quad (1)$$

where V_c is initial trapped fluid volume in control side, x_p is relative displacement of piston, c_l is coefficient of total leakage, p_c is control pressure acting on control side, p_h is pressure acting on uncontrolled side, A_c is piston area on control side, β_e is bulk modulus of oil, q_v is load flow.

For the servo valve, assuming it is critically centered and the orifices are matched and symmetrical, the dynamic model for spool movement can be given as

$$\frac{A_v(s)}{I_v(s)} = \frac{K_v}{s\tau_v + 1}, \quad (2)$$

where τ_v is time constant of valve torque motor, K_v is servo valve constant, s is Laplace operator, A_v is opening area of servo valve, I_v is input control current.

The load flow is a nonlinear function of control pressure and control state and is given by

$$q_v = \begin{cases} C_d A_v \sqrt{\frac{2}{\rho}(p_s - p_c)}, & p_s > p_c \cap O+; \\ -C_d A_v \sqrt{\frac{2}{\rho}(p_c - p_s)}, & p_s < p_c \cap O+; \\ C_d A_v \sqrt{\frac{2}{\rho} p_c}, & p_c > 0 \cap O-; \end{cases} \quad (3)$$

where C_d is flow discharge coefficient, ρ is mass density of fluid oil, p_s is supply pressure, $O+$ is positive position of servo valve, through which the oil from pump flows to the control side of hydraulic cylinder, $O-$ is negative position of servo valve, through which the oil from hydraulic pump returns to the tank at atmospheric pressure.

Considering the inertial load, viscous friction coefficient and external load force, the output equation of hydraulic cylinder can be described as

$$p_c A_c - p_h A_h = m_1 \ddot{x}_1 + \mu \dot{x}_p + F_w, \quad (4)$$

$$x_p = x_1 - x_0, \quad (5)$$

where A_h is piston area on uncontrolled side, m_1 is total mass of piston and edger roll, μ is viscous friction coefficient, F_w is external load force (rolling force), x_1 is absolute displacement of piston, x_0 is absolute displacement of case of hydraulic cylinder.

For the stand, assuming it is critically centered, the dynamic model for stand vibration can be given as

$$p_{\mathrm{h}}A_{\mathrm{h}} - p_{\mathrm{c}}A_{\mathrm{c}} + \mu\dot{x}_{\mathrm{p}} = m_0\ddot{x}_0 + k_0x_0 + c_0\dot{x}_0, \tag{6}$$

where k_0 is rigidity of stand, c_0 is viscous damping coefficient, m_0 is total mass of stand and hydraulic cylinder case.

CONTROLLER DESIGN

The active disturbance rejection control theory was firstly put forward by Han (1988). Compared with traditional PID control theory, it gets over the differential obsession by integral transformation, combines control engineering experience by nonlinear combination of error and reduces disturbances by error estimation (Han, 1990; 1994).

ADR control system is composed of nonlinear tracking differentiator (TD), extended state observer (ESO) and nonlinear feedback (NF) law as shown in Fig.2.

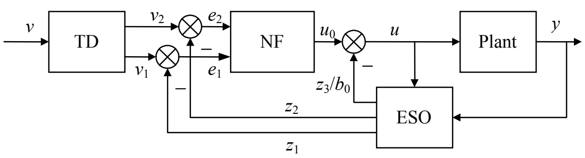

Fig.2 The diagram of ADR control system

Nonlinear tracking differentiator

Proportional, integral and derivative feedback is based on the past (I), present (P) and future (D) control error. Nevertheless, most loops are in fact PI because the noise, interfused in input signals or feedback signals, are easily amplified by derivative. The PID equation can be given as

$$u(k) = \left(K_{\mathrm{P}} + \frac{K_{\mathrm{I}}}{z^{-1} - 1} + K_{\mathrm{D}}(1 - z^{-1}) \right) e(k), \tag{7}$$

where $e(k)$ is control error at the Kth instant, K_{P} is proportional coefficient, K_{I} is integral coefficient, K_{D} is differential coefficient, $u(k)$ is system input signal at the Kth instant.

Consider a TD model:

$$\begin{cases} v_1' = v_2, \\ v_2' = -M\mathrm{sgn}\left(v_1 - v_0 + |v_2| v_2 / (2M)\right), \end{cases} \tag{8}$$

where v_1 is tracking result of input signal v, v_2 is differential of v_1, M is tracking coefficient. Instead of derivative operation, v_2 can derive from integral operation. It has been found that the integral transformation is beneficial for avoiding differential disaster (Han, 1988).

The step response of TD model with different value M is shown in Fig.3.

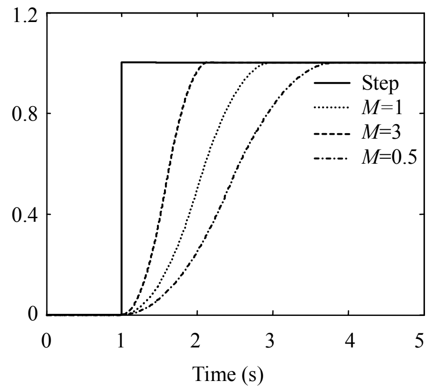

Fig.3 The diagram of step response of TD model

Extended state observer

Consider an ESO model:

$$\begin{cases} e = z_1 - y, \\ \dot{z}_1 = z_2 - \beta_{01} fal(e, \alpha, \delta_1), \\ \dot{z}_2 = z_3 - \beta_{02} fal(e, \alpha, \delta_1) + b_0 u, \\ \dot{z}_3 = -\beta_{03} fal(e, \alpha, \delta_1), \end{cases} \tag{9}$$

$$fal(x, \alpha, \delta) = \begin{cases} |x|^{\delta} \mathrm{sgn}(x), & |x| \ge \delta; \\ x / \delta^{1-\alpha}, & |x| < \delta, \end{cases} \tag{10}$$

where z_1 is tracking result of output signal y, z_2 is approximate differential of z_1, z_3 is estimate of disturbance, δ is turning point of fal function, α is nonlinear coefficient, β_{01}, β_{02}, β_{03} are coefficients of observer.

Note Eq.(9) is a continual, yet non-differential, function, which is the arithmetic fit of engineering experience, larger error with less regulated gain, while less error with larger regulated gain.

The characteristics of fal function with different value α, while δ is 0.7, are shown in Fig.4.

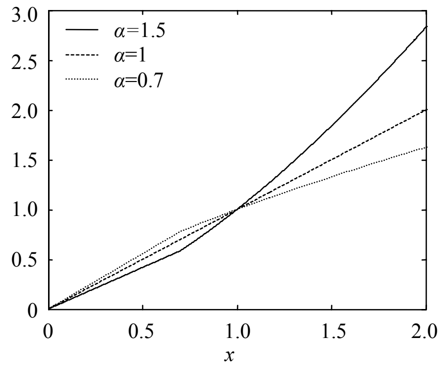

Fig.4 The characteristics of *fal* function

Table 1 The simulation parameters in the hydraulic width control system

Parameter	Value
τ_v	0.00825
K_v	0.5
V_c (m^3)	0.0063
p_s (MPa)	28
β_e (MPa)	700
C_d	0.2594
A_h	0.1052
A_c	0.1948
μ (N/(m·s))	1200

Nonlinear feedback law

A PID controller, which is designed based on the linear combination of proportional, integral and derivative feedback, is independent of the model of control plant. Because of simple linear combination, there is a limitation in resolving the conflict of fast response and overshoot of control system.

Consider an ESO model:

$$\begin{cases} e_1 = v_1 - z_1, \\ e_2 = v_2 - z_2, \\ u_0 = K_{p1} fal(e_1, \alpha, \delta) + K_{p2} fal(e_2, \alpha, \delta), \\ u = u_0 - z_3 / b_0, \end{cases} \tag{11}$$

where e_1 is system error, e_2 is system differential error, u_0 is control signal, u is system input signal. K_{p1} is proportional gain, K_{p2} is differential gain. The control signal u_0, which inherits the model-independent characteristic of PID controller and compromises effectively the conflict between fast response and overshoot, is a nonlinear combination of proportional and derivative feedback.

SIMULATIONS AND DISCUSSION

Simulations were performed to investigate the performance of the proposed nonlinear controllers. The dynamic model of the hydraulic system and ADR controller described before are used in the simulation. The simulation parameters of the hydraulic width control system are listed in Table 1.

Note the parameters of ESO, β_{01}, β_{02}, β_{03}, can be searched efficiently by the evolution strategies (Kim and Lee, 2006; Zou *et al*., 2006).

The following values were used for the ADR controller:

$$\beta_{01}=75.58, \beta_{02}=85.27, \beta_{03}=0.1,$$
$$M=30, \delta=0.9, b_0=0.0047.$$

The tracking performance of ESO, compared with system output, for a step reference trajectory is shown in Fig.5. The ESO tracks the system output almost perfectly, disregarding of nonlinear characteristic of hydraulic servo system.

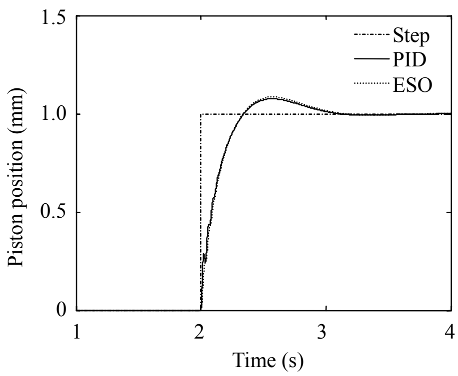

Fig.5 The tracking performance of ESO

The step responses of PI controller and ADR controller are compared in Fig.6. The proposed controller has better performance, especially much smaller settling time compared with the PI controller. It should be pointed out that the control system with ADR controller has zero overshoot while the rise time is almost the same as that of the system with PI controller.

The system performance, under sinusoidal rolling force disturbance with f=2 Hz, of PI controller and ADR controller is compared in Fig.7. The proposed controller has better performance, especially much smaller response gain compared with the PI controller.

The system performance, under the step change of rolling force with ΔF=0.7 MN, of PI controller and ADR controller is compared in Fig.8. Again the proposed controller has better performance. The settling time of the ADR controller is significantly shorter than that of PID controller.

After long-time operation, the leakage of hydraulic servo system will drastically increase because of wear and aging. It results in the deterioration of the performance of the hydraulic width control system. The closed-loop outputs achieved by these two controllers, when an abrupt leakage increase occurs, are shown in Fig.9. The hydraulic width control system with ADR controller can restore more quickly than that with PI controller.

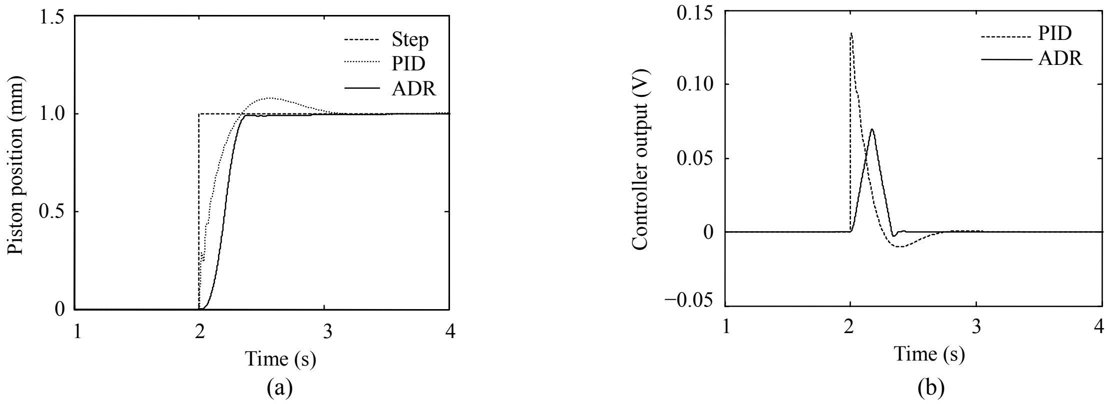

Fig.6 The step response of PID and ADR controllers. (a) Step response; (b) Controller output

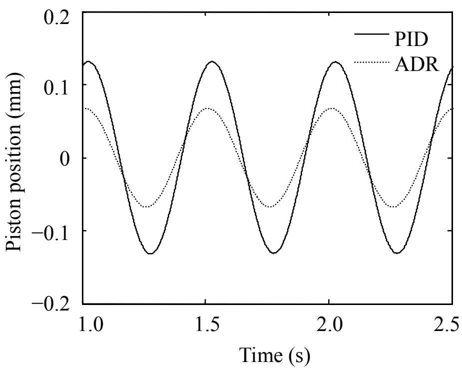

Fig.7 The performance of PID and ADR controllers under sinusoidal rolling force disturbance condition

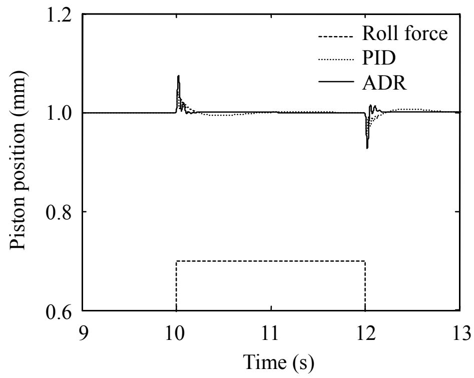

Fig.8 The performance of PID and ADR controllers under step change rolling force condition

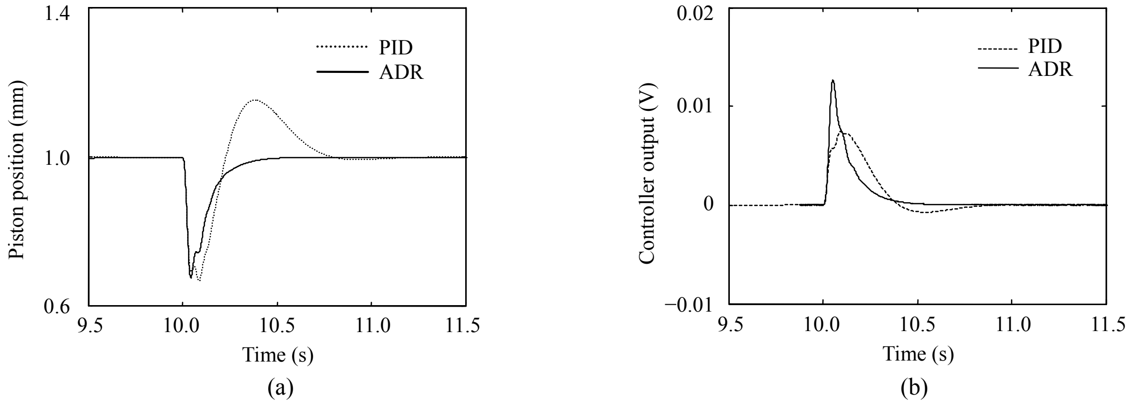

Fig.9 The performance of PID and ADR controllers with an abrupt leakage increase. (a) System response; (b) Controller output

CONCLUSION

The highly nonlinear behavior of the system limits the performance of classical linear PID controllers used for hot rolling. An ADR controller is proposed in this paper. The proposed controller, which is model-independent, can observe the outer disturbance and compensate it. The good performance of ADR controller is verified by the application of hydraulic edger screwdown system in rough mill. The simulation results showed that the ADR controller achieves better performance than the classical one, especially, in the circumstance of foreign disturbance and working condition variation.

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