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OPTIMAL-TUNING PID CONTROLLER DESIGN FOR HYDRAULIC POSITION CONTROL SYSTEMS

G. P. Liu* and S. Daley †

*School of Mechanical, Materials, Manufacturing Engineering and Management
University of Nottingham, University Park, Nottingham NG7 2RD, U.K.
Fax: +44 (0)115 9514000, Email: guoping.liu@nottingham.ac.uk

† ALSTOM Power Technology Centre, Cambridge Road, Leicester LE8 6LH, U.K.
Fax: +44 (0)116 2015464, Email: steve.daley@power.alstom.com

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Abstract

An optimal-tuning PID controller design scheme is presented for hydraulic position control systems. After analysis on physical dynamics of the systems, a nonlinear model is derived. The PID parameters are searched by optimisation methods to satisfy some desired time-domain performance requirements. Based on an estimated process model, the optimal-tuning PID controller provides optimal PID parameters even in the case where the process dynamics changes. To implement the optimal-tuning PID control in practice safely, a control performance prediction strategy is incorporated into the scheme. The proposed PID control technique is implemented on a PID controller unit and also tested on a hydraulic position control test rig.

1 Introduction

Since Ziegler and Nicholes (1942) proposed the design formula of the proportional, integral and derivative (PID) controller in 1940s, the PID controller has been widely applied in industry in spite of the development of a vast array of advanced control strategies in the last four decades. Its popularity stems from its applicability and robust performance in a wide variety of operating scenarios. Moreover, there is a wide conceptual understanding of the effect of the three terms involved amongst non-specialist plant operators that makes manual tuning a relatively straightforward task. A number of PID tuning rules have developed in the last 50 years use frequency response methods (see, e.g., Kessler, 1958; Voda and Landou, 1995; Liu *et al*., 1998; Daley and Liu, 1999; Liu and Daley 1999a,b). They provide simple tuning formulae to determine the PID controller parameters. However, since only a small amount of information on the dynamic behaviour of the process is used, in many situations they do not provide good tuning or produce satisfactory closed-loop response, for example, oscillatory response to setpoint changes. It is also difficult for them to cope with the dynamic change of the system.

In order to improve the performance of PID tuning for processes with changing dynamic properties, several automatic tuning and adaptive strategies have been proposed (Astrom and Hagglund, 1984; McCormack and Godfrey, 1998). These controllers have self-initialisation and recalibration features to cope with little *a priori* knowledge and significant changes in the process dynamics. However, the PID controller parameters are still computed using the classic tuning formulae and, as noted above, these do not provide a good control performance in all situations. Moreover, in some plants there can be several control loops the tuning of which can be highly time consuming and in addition, ageing and non-linear effects can lead to controller parameters that are far from the optimal. This can result in inefficient plant operation, unnecessary wear and unscheduled shutdowns.

To overcome the above limitations, this paper presents an optimal-tuning PID controller design scheme. It mainly includes four parts: nonlinear model estimation, a definition of desired system specifications, optimal-tuning mechanism and PID controller. The nonlinear model for the process is estimated using a recursive method. The desired system specifications are directly given in the time-domain by the time response of a desired closed-loop system. The optimal-tuning mechanism searches optimal PID parameters so that the desired specifications are satisfied. The paper demonstrates the feasibility of using this technique for reducing commissioning time and improving the control performance of the hydraulic systems.

2 Hydraulic Position Control Test Rig

The hydraulic position control test rig as shown in Figure 1 was designed and built by ALSTOM Power Technology Centre, England for dynamic performance testing of position controllers. The rig consists of two hydraulic cylinders: main cylinder and loading cylinder, which are coupled by a shaft. The main cylinder is controlled by a 3 way proportional

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valve. The loading cylinder is controlled by a Moog servo valve to simulate variations in load. Hydraulic pressure for operation of the rig is generated by the fluid power lab hydraulic power pack and supplied by the ring main system.

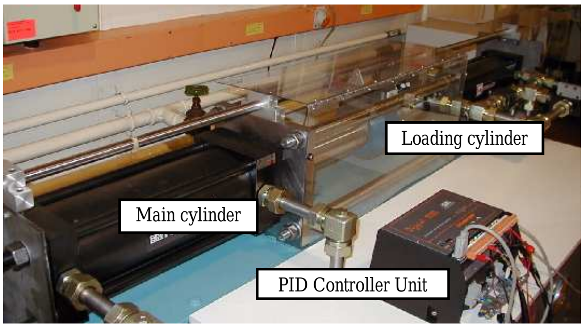

Figure 1 The hydraulic position control test rig

The flow of hydraulic fluid to the main cylinder is controlled by a 3 way proportional valve operated by a controller. The position of the main cylinder along its stroke is measured using a Linear Variable Displacement Transducer (LVDT) mounted above the cylinder - this signal is used for closed-loop control of the main cylinder position. In order to gather performance information about the position controllers under test, other sensors monitor the following: the coil current of the valve and the force of the main cylinder. The rig is fitted with an emergency stop button which if activated will shut down the power pack.

This test rig is used to simulate a control system of hydraulic turbines. The main cylinder represents the turbine servomotor. The loading cylinder represents the turbine adjusting mechanism, which may be a distributor on a Francis or pump turbine, a distributor and/or runner blades on a Bulb or Kaplan unit, or defectors and/or injectors on Pelton turbines.

The PID controllers are implemented by a device, called the PID Controller Unit. It mainly consists of three electronic boards. The first is the central processing unit board including the microprocessor, the RAM, EPROM and EEPROM memories, the logic (TOR) and analog input/output interfaces and the serial link. The second is the power amplifier board driving the actuator. The third is the power supply board for the entire unit. The PID Controller Unit configuration is shown in Figure 2. The PID parameters are K_P (the proportional gain), K_I (the integral gain) and K_D (the derivative gain). L_b and U_b are the lower and upper bounds of the actuator input signal. D_c is the DC offset compensation.

The main parameters of the PID Controller Unit can be accessed via a serial link to a data-processing software which enables communication with the PID Controller Unit and can be used with PC type computers. The software performs the function of measuring instruments, such as digital oscilloscope, synchroscope, voltmeter and etc. It has two operating modes. One is on-line functionality allowing real-time communication with the Controller Unit, which enables the operator to visualise, modify or record the governing parameters. The other is off-line functionality (not connected to the controller) allowing preparation of parameters for direct use or enquiry of records libraries.

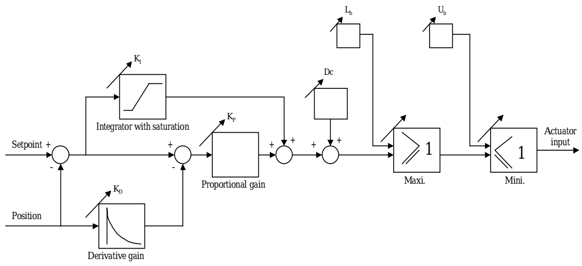

Figure 2 The PID Controller unit configuration

3 Modelling of the Hydraulic System

In order to design an optimal controller and predict control performance for the hydraulic test rig, the theoretical and practical modelling of the rig is needed. Taking nonlinearities into account, a model for the hydraulic position control system is derived below, which uses the 3 way proportional valve for position control. The dynamics of the spool-displacement of the valve is represented by the following second order differential equation:

$$\frac{d^2 x_v}{d^2 t} + 2\varsigma_v \omega_v \frac{dx_v}{dt} + \omega_v^2 x_v = \omega_v^2 K_v i \tag{1}$$

where x_v is the valve spool displacement from neutral, i the input current of the valve from the controller, ω_v the natural frequency of the valve, ς_v the damping ration of the valve and K_v the gain of the valve.

The general pressure-flow function of the valve has the form

$$q = f(x_v, p) \tag{2}$$

where q is the flow through the actuator, p the pressure drop across the actuator and $f(.,.)$ the non-linear function. The above equation can be linearised and expressed as a Taylor's series about a particular operating point. For $x_v = 0,\ p = 0$, the linearised equation of the pressure-flow function becomes

$$q = L_x x_v - L_p p \tag{3}$$

where

$$L_x = \left.\frac{\partial f}{\partial x_v}\right|_{x_v=0,\ p=0} \qquad L_p = -\left.\frac{\partial f}{\partial p}\right|_{x_v=0,\ p=0} \tag{4}$$

which vary in magnitude with the operating point.

With the assumption that the piston of the driving cylinder is centred, the continuity expression of the oil flow is

$$q = M_1 \frac{dy}{dt} - C_t\, p + \frac{V_t}{4\beta_t} \frac{dp}{dt} \tag{5}$$

where y is the displacement of the piston, M_1 the effective area of the piston in the left-side chamber of the main cylinder, C_t total leakage coefficient of the piston, V_t the total volume of the fluid under compression in both the left- and right-side chambers and β_t the effective bulk modulus of the system.

The force balance on the piston is

$$p = \frac{-1}{M_1}\left(m\frac{d^2 y}{d^2 t} + C_d \frac{dy}{dt} + F + (M_2 - M_1)P_0 \right) \tag{6}$$

where m is the total mass of the piston and the load, F the external load force on the piston which is provided by the force control system on the loading cylinder, P_0 the pressure of the right-side chamber of the main cylinder provided by hydraulic power supply, C_d the viscous damping coefficient of the piston and the load and M_2 the effective area of the piston in the right-side chamber.

Discretization of the above differential equations of the linearised hydraulic position system gives the following discrete-time model

$$Ay(t) = q^{-d} Bi(t) + d_0 \tag{7}$$

where d_o is the dc offset, d is the time delay, and the polynomials A and B are

$$A = 1 + \sum_{i=1}^{n} a_i q^{-i} \quad B = \sum_{i=0}^{m} b_i q^{-i} \tag{8}$$

To cope with the nonlinearity, an adaptive model based on an ARX structure is used. The adaptive model is calculated using a set of time varying parameters obtained from a least squares algorithm with a forgetting factor to track parameter variations caused by unmodelled nonlinearity. The model form can be written as

$$y(t) = \theta^T \phi_{t-1} \tag{9}$$

where

$$\theta = [a_1, a_2, \ldots, a_n, b_0, b_1, \ldots, b_m, d_0]^T \tag{10}$$

$$\phi_{t-1} = [-y(t-1),\ldots,-y(t-n), i(t-d), ,\ldots, i(t-d-m), 1]^T \tag{11}$$

The least squares algorithm with a freezing factor is

$$\hat{\theta}_t = \hat{\theta}_{t-1} + \beta P_t \phi_{t-1}(y(t) - \hat{\theta}_{t-1}^T \phi_{t-1}) \tag{12}$$

$$P_t = \begin{cases} \lambda^{-1}(P_{t-1} - P_{t-1}\phi_{t-1}\phi_{t-1}^T P_{t-1}(\lambda + \phi_{t-1}^T P_{t-1}\phi_{t-1)})^{-1}) & if\ \beta = 1 \\ P_{t-1} & if\ \beta = 0 \end{cases} \tag{13}$$

where $\hat{\theta}_t$ is the estimated parameter vector, $\lambda \in [0,1]$ is a forgetting factor, and $\beta \in \{0,1\}$ is the parameter freezing factor. The freezing factor β is set manually by the designer or automatically by a certain condition to turn adaptation on and off. For example, if the modelling error is within the desired bound, then set $\beta = 0$.

In practice, it has been found that the dynamics of the hydraulic position system is different in forward and backward moving directions of the main cylinder. Thus, a non-linear ARX model for the system may be expressed by

$$\begin{cases} A_f\, y(t) = q^{-d_f} B_f\, i_v(t) + d_{of} & \text{for the forward direction} \\ A_b\, y(t) = q^{-d_b} B_b i_v(t) + d_{ob} & \text{for the backward direction} \end{cases} \tag{14}$$

where A_f, A_b, B_f and B_b are the polynomials, d_{of} and d_{ob} the dc offset, d_f and d_b the time delay.

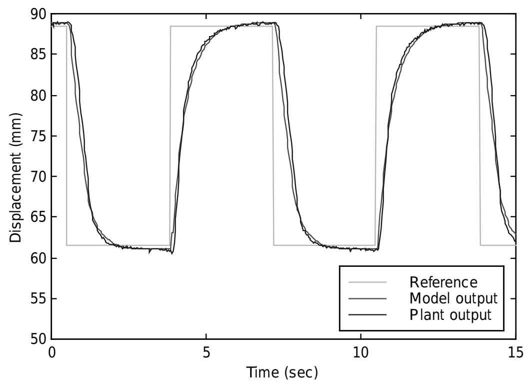

Figure 3 The responses of the plant and model

Based on the physical model derived above, the dynamics of each direction was modelled by a fifth-order ARX model, respectively. Using a PID controller ($K_P=1.5$, $K_I=0$ and $K_D=0.05$) which will be introduced in Section 4, the parameters of the model were estimated using the recursive least squares algorithm. The accuracy of the model was tested below. The system ran at the operating point that the amplitude, period and dc-offset of the square-wave reference input were 13.5mm, 6.7s and 75mm. The parameters of the model were firstly updated at this operating point and then fixed by setting the freezing factor $\beta = 0$. The responses of the plant and model are shown in Figure 3. It is clear that the model output is close to the output of the plant. From the experimental results, it shows that the direction dependent ARX model represents the plant quite well.

4 Optimal PID Controller Design

According to the PID Controller Unit configuration shown in Figure 2 , the PID controller is expressed by

$$i(t) = K_P\left(e(t) + K_D\frac{dy(t)}{dt}\right) + K_I\int_0^t e(t)dt \qquad (15)$$

where K_P, K_I and K_D are the PID parameters, $e(t) = r(t) - y(t)$ is the difference between the reference input *r(t)* and the position *y(t)*.

Since PID controller parameters are usually designed using either one or two measurement points of the system frequency response, their control performance may not satisfy the desired time-response requirements. To overcome this disadvantage, an optimal PID controller design is proposed in the time domain.

In the time domain, specifications for a control system design involve certain requirements associated with the time response of the system. The requirements are often expressed in terms of the standard quantities on the rise time, settling time, overshoot, peak time, and steady-state error of a step response. The time responses of two standard systems are widely used to represent these requirements. They are the first- and second-order systems. Their transfer functions are

$$G_1(s) = \frac{\omega_0}{s + \omega_0} \quad G_2(s) = \frac{\omega_n^2}{s^2 + 2\varsigma\omega_n s + \omega_n^2} \qquad (16)$$

where the parameters ω_0 and ω_n are natural frequencies, and ς the damping ratio. In order to have a good closed-loop time response, the following performance function needs to be considered during the design of a PID controller: for the continuous-time case

$$J(K_P, K_I, K_D) = \int_0^\infty (y_{step}(t) - y_{step}^d(t))^2 dt \qquad (17)$$

or for the discrete-time case

$$J(K_P, K_I, K_D) = \sum_{t=0}^{\infty}(y_{step}(t) - y_{step}^d(t))^2 \qquad (18)$$

where $y_{step}^d(t)$ is the desired step response which may be produced by the transfer function $G_1(s)$ or $G_2(s)$, and $y_{step}(t)$ the step response of the system with the PID controller. Since it is often not allowed to try different PID controller parameters on the plant for the sake of safety, $y_{step}(t)$ is replaced by the step response of the model with the PID controller. Thus, the optimal PID controller design may be stated as

$$\min_{K_P, K_I, K_D} J(K_P, K_I, K_D) \qquad (19)$$

There are a number of optimisation methods to solve the above problem. Many optimisation algorithms now exist in standard libraries of optimisation software, for example, the optimisation toolbox for use with MATLAB (Grace, 1994).

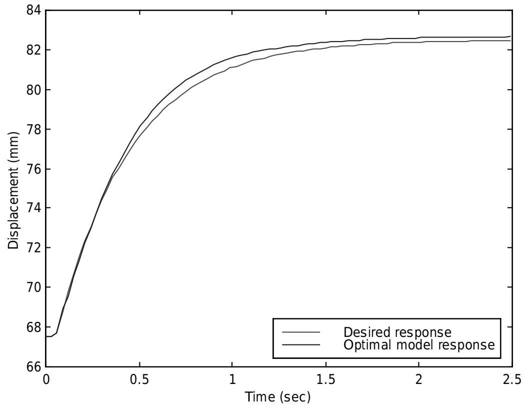

Figure 4 The desired response and optimal model response.

Now, the optimal PID controller for the hydraulic system is considered. The first-order transfer function with $\omega_o = 2.5$ was used as the tracking model. The cost function was given by $J(K_P, K_I, K_D)$. The reference input was a step signal, which changed from 67.5mm to 82.5mm. Using function 'fmins' in the optimisation toolbox, the optimal PID parameters *K_P=1.923, K_I=0.01* and *K_D=0.02* were found. The desired response of the tracking model and the closed-loop step response of the model with the optimal PID controller are shown in Figure 4.

5 Optimal-Tuning PID Control

When a system has different operating points with widely differing dynamic properties, it is not always possible to control with a fixed parameter controller, even if this is a highly robust controller. For this case, an optimal-tuning PID control scheme is proposed as shown in Figure 5. It mainly consists of four parts: model parameter estimation, desired system specifications, optimal-tuning mechanism and PID controller. The model parameters are estimated using the least squares identification method. The desired system specifications are represented by the time response of the desired standard first- or second-order systems. The optimal-tuning mechanism finds optimal parameters for the PID controller so that the desired system specifications are satisfied.

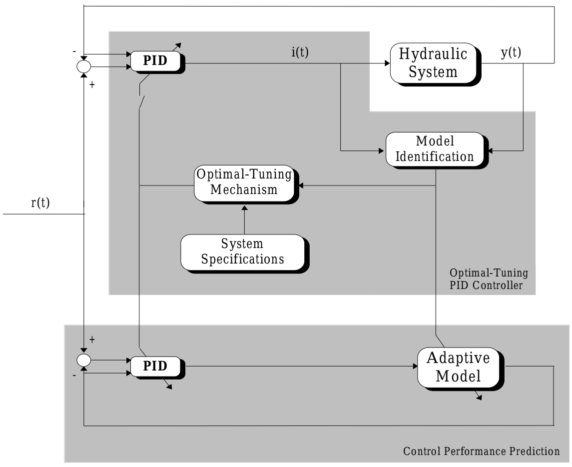

Figure 5 The optimal-tuning PID control scheme with control performance prediction.

The operating procedure of the optimal-tuning PID control is as follows. When the system operating point or dynamics changes, the new model parameters are re-estimated by switching on the estimation algorithm. Then, using this updated model parameters, the tuning mechanism searches for the optimal parameters for the PID controller to satisfy the desired system specifications. Finally, the PID controller is set to have the obtained optimal parameters. In this way, the PID controller may cope with all operating points of the system and the closed-loop system will have similar optimal control performance. But, compared with a fixed parameter control, the disadvantage of this strategy is that it needs slightly more computation to search for the optimal parameters.

The optimal-tuning PID control technique is applied to the hydraulic position control test rig, which is representative of many industrial systems which utilise fluid power. This is a particularly apposite application of the method since hydraulic systems are often very conservatively tuned due to the fact that the cost of getting the tuning wrong can be highly destructive and costly.

To implement the optimal-tuning PID control in practice safely, a control performance prediction strategy is proposed, as shown in Figure 5. It mainly comprises the adaptive model updated by an on-line system identification algorithm and a PID controller. The parameters of the PID controller is adjusted by the optimal-tuning PID algorithm. Roughly speaking, the implementation of PID control strategies consists of two stages. The first stage is to run a PID controller on the on-line identified model to predict the performance of the PID controller before it is used on the rig. The second stage is to apply the PID controller to the hydraulic system using the PID parameters which are verified to be safe on the adaptive model. In this way, unnecessary damage which could result from the wrong PID parameters can be prevented though a slight increase in computational load.

To effectively assess the performance of the proposed optimal-tuning scheme and demonstrate the operation of the control performance prediction strategy, three experimental cases have been considered on the hydraulic position control test rig. For all three cases, the amplitude, period and dc offset of the square-wave reference input are 7.5mm, 5 seconds and 75mm, respectively. The tracking model is a first order system with $\omega_o = 2.5$. The nonlinear model parameters of the plant were estimated using the recursive least squares algorithm.

Case A: The parameters of the PID controller were designed using optimisation methods, based on the estimated model for the backward direction. They were $K_P=0.8$, $K_I=0.011$ and $K_D=0.05$. This controller was applied to both the plant and the model. Their responses are given in Figure 6. It is clear that the model and plant responses in the backward direction are very close to the desired response but are away from it in the forward direction.

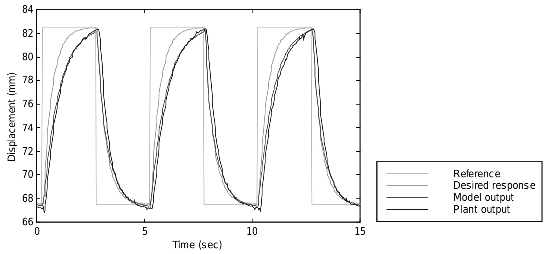

Figure 6 The responses of the plant, model and tracking model (Case A)

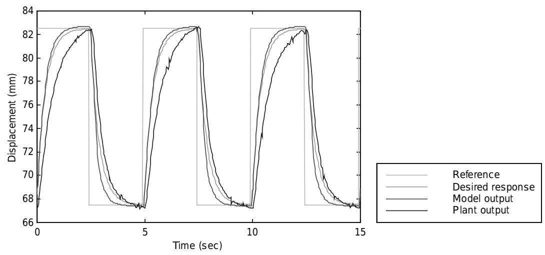

Figure 7 The responses of the plant, model and tracking model (Case B)

Case B: Based on the estimated model for the forward direction, the optimal PID parameters were found by the optimal-tuning mechanism to be $K_P=1.923$, $K_I=0.01$ and $K_D=0.02$. This optimal PID controller was applied to the model only. The closed-loop response of the model with the optimal PID controller is very close to the desired response in the forward direction, as shown in Figure 7.

Case C: The optimal PID controller (*K_P=1.923, K_I=0.01* and *K_D=0.02*) was applied to the plant as well. It can be seen from Figure 8 that the closed-loop responses of both the plant and the model with the optimal PID controller are very close to the desired response in the forward direction but are away from it in the backward direction.

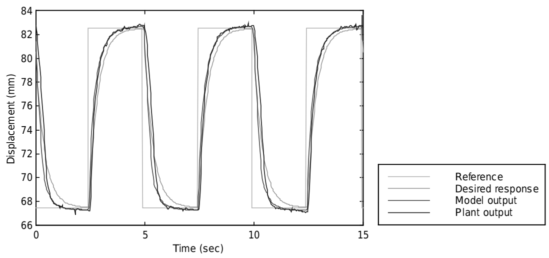

Figure 8 The responses of the plant, model and tracking model (Case C)

Clearly, the optimal-tuning PID control and control performance prediction work quite well. Since there exists different dynamics in the forward and backward directions in the system, the system cannot satisfy the desired requirements in both directions using a fixed-parameter PID controller. To have the same desired performance in both forward and backward directions, the parameters of the PID controller must be switched between two optimal parameter sets. Otherwise, some trade-offs should be considered.

6 Conclusions

The optimal-tuning PID controller design scheme has been proposed for the hydraulic position control system. Since the dynamics of the system for each movement direction is different, a non-linear direction dependent ARX model was used. The optimal-tuning PID design scheme mainly consists of four parts: non-linear model estimation, definition of desired system specifications, optimal-tuning mechanism and PID controller. The non-linear model for the process is estimated using the recursive least squares method. The desired system specifications are represented by the time response of the desired first- or second-order system. The optimal-tuning mechanism finds optimal parameters for the PID controller so that the desired system specifications are satisfied. To implement the optimal-tuning PID control in practice safely, a control performance prediction strategy has been proposed. It mainly includes the adaptive model updated by an on-line system identification algorithm and a PID controller. A PID controller runs firstly on the adaptive model to predict its performance before it is used on the rig. In this way, unnecessary damage which could result from the wrong PID parameters can be prevented. These techniques above have been successfully applied to the hydraulic position control test rig. The experimental results have shown that they can significantly improve system performance.

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