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APPLICATION OF OPTIMAL-TUNING PID CONTROL TO INDUSTRIAL HYDRAULIC SYSTEMS

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Abstract: This paper is concerned with real-time optimal-tuning PID control design for industrial hydraulic systems. Several optimal-tuning controller design techniques are used for a hydraulic position control system. After analysis on the system, a nonlinear dynamical model is derived. A nonlinear PID control scheme with inverse of dead zone is introduced to overcome the dead zone in this hydraulic system. An optimal PID controller is designed to satisfy desired time-domain performance requirements. Using the adaptive model, an optimal-tuning PID control scheme is proposed to provide optimal PID parameters even in the case where the system dynamics are time variant. To implement the optimal-tuning control in practice safely, a control performance prediction strategy is integrated. These control techniques are implemented on dSPACE and are also demonstrated by a number of experiments on a hydraulic position control test rig.

Keywords: Optimal-tuning control, real-time control, optimal control, hydraulic systems.

1. INTRODUCTION

Hydraulic systems are widely used in industry. Recently, most of controllers for industrial hydraulic systems are still the proportional, integral and derivative (PID) controller. The PID 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. Based on frequency response methods, many PID tuning rules have developed in the last 50 years, for example, Ziegler-Nichols rule (Ziegler and Nichols, 1942), symmetric optimum rule (Kessler, 1958; Voda and Landou, 1995), some-overshoot rule (Seborg *et al*. 1989), no-overshoot rule (Seborg *et al*. 1989), refined Ziegler-Nicholes rule (Hang *et al*. 1991), integral of squared time weighted error rule (Zhuang and Atherton, 1993), and integral of absolute error rule (Pessen, 1994). These methods are straightforward to apply since 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 responses.

In an effort to improve the performance of PID tuning for processes with changing dynamic properties, several automatic tuning and adaptive strategies have been proposed (Kraus and Mayron, 1984; Astrom and Hagglund, 1984; Radke and Issermann, 1987; McCormack and Godfrey, 1998). These controllers have self-initialisation and re-calibration 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. 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 deal with this problem, an optimal-tuning PID control scheme has been proposed in recent years (Liu and Daley, 1999, 2000; Daley and Liu, 1999; Liu *et al.* 2000).

The paper is to demonstrate the feasibility of using optimal-tuning PID control techniques for reducing commissioning time and achieving consistent control performance for hydraulic systems with wide operating range. The following sections detail modelling of hydraulic systems, nonlinear PID control, optimal PID controller design, optimal-tuning PID control scheme, PID control toolbox and experimental results.

2. MODELLING OF HYDRAULIC SYSTEMS

To test dynamic performance of the optimal-tuning PID controller, a hydraulic control test rig was designed and built by ALSTOM Power Technology Centre, UK, as shown in Figure 1. The rig consists of two hydraulic cylinders: main cylinder and loading cylinder. The main cylinder is coupled by a shaft to the loading cylinder. The main cylinder is controlled by a 3 way proportional 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.

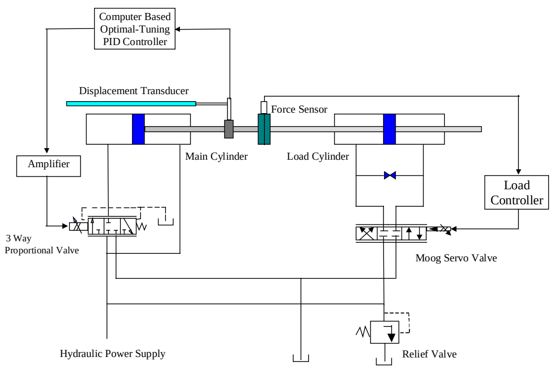

Figure 1 The hydraulic position control system

The flow of hydraulic fluid to the main cylinder is controlled by a 3 way proportional valve operated through an amplifier 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 via a computer based optimal-tuning PID controller. In order to gather performance information about the optimal-tuning controller under test, the sensors monitor the following: the main cylinder displacement, 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.

The hydraulic test rig is mainly 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 optimal-tuning PID controller is implemented in the environment that consists of dSAPCE, Matlab, Simulink and Real-Time Workshop.

In order to design an optimal controller and assess control performance for the hydraulic test rig, the theoretical and practical modelling of the rig is needed. Taking nonlinearities into account, the hydraulic system can be modelled by two components: the dead zone nonlinearilty and the linearised model.

The valve spool occludes the orifice with some overlap so that for a range of spool positions there is no fluid flow. Thus, a dead zone should be placed between the valve dynamics and actuator/load dynamics. For the sake of simplicity, this dead zone is equivalently moved to the position between the output of the controller and the input current of the valve. So, the dead zone nonlinearity may be approximately described as

$$i_v = \begin{cases} i - I_1 & \text{if } i > I_1 \\ 0 & \text{if } |i| \le I_1 \\ i + I_1 & \text{if } i < -I_1 \end{cases} \tag{1}$$

where i_v the input current of the valve from the dead zone, i is the input current from the controller and I_1 the width of the dead zone.

The linearised discrete-time model of the hydraulic position system is given by

$$A\hat{y}(t) = q^{-d} B i_v(t) + d_0 \tag{2}$$

where A and B are the polynomials. 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

$$\hat{y}(t) = \theta^T \phi_{t-1} \tag{3}$$

where θ is the parameter vector and ϕ_{t-1} the input-output vector. The least squares algorithm with a parameter-freezing factor is

$$\hat{\theta}_t = \hat{\theta}_{t-1} + \beta P_t \phi_{t-1}(y(t) - \hat{\theta}^T \phi_{t-1}) \tag{4}$$

$$P_t = \begin{cases} \lambda^{-1}(P_{t-1} - P_{t-1}\phi_{t-1}\phi_{t-1}^T P_{t-1}(\lambda + \phi_{t-1}^T P_{t-1}\phi_{t-1)})^{-1}) & if\ \beta = 1 \\ P_{t-1} & if\ \beta = 0 \end{cases} \tag{5}$$

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. It can also be used to compare fixed parameter control and adaptive parameter control.

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

$$\begin{cases} A_f\hat{y}(t) = q^{-d_f} B_f i_v(t) + d_{of} & \text{forward} \\ A_b\hat{y}(t) = q^{-d_b} B_b i_v(t) + d_{ob} & \text{backward} \end{cases} \quad (6)$$

where A_f, A_b, B_f and B_b are the polynomials.

Based on the physical model derived above, the dynamics of each direction is modelled by a fifth-order ARX model, respectively. The accuracy of the model was tested. The system ran at the operating point that the amplitude, period and dc-offset of the square-wave reference input were 20mm, 10s and 75mm. Using a non-linear PID controller ($K_P=1$, $K_I=0$, $K_D=0$ and $I_1=0.05$), which will be introduced in Section 5, the modelling performance of the hydraulic system is shown in Figure 2. . It is clear that the model output is very close to the output of the plant.

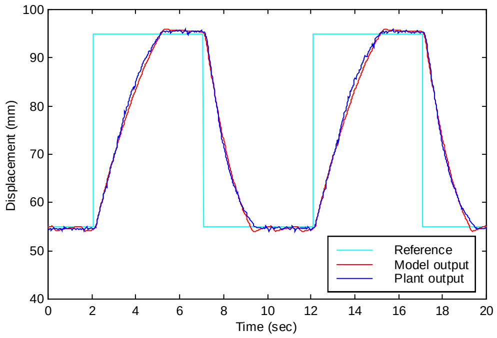

Figure 2 Responses of the plant and model

3. NON-LINEAR PID CONTROL

It is assumed that the ideal transfer function of a PID controller is given by

$$K(s) = K_P\left(1 + \frac{K_I}{s} + K_D s\right) \quad (7)$$

where K_P, K_I and K_D are the PID parameters. Since there exists dead-zone in the hydraulic position system, it is difficult to achieve high-precision tracking using only linear controllers. The approach is to add a non-linear compensator, which includes the inverse of dead zone, to the linear controller. The inverse of dead zone is employed to cancel the dead zone. This structure is shown in Figure 3.

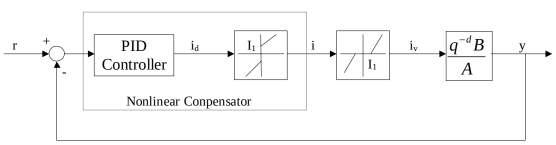

Figure 3 Closed-loop control system

The inverse of dead zone is described by

$$i = \begin{cases} i_d + I_1 & \text{if } i_d > 0 \\ 0 & \text{if } i_d = 0 \\ i_d - I_1 & \text{if } i_d < 0 \end{cases} \quad (8)$$

If the parameter I_1 is exactly known, the dead-zone inverse cancels the dead-zone effect, that is,

$$i_v = i_d \quad (9)$$

So, the dead zone in the system is removed by the inverse of dead zone in the controller. The PID controller design for the nonlinear system is simplified to the PID design for a linear system.

Since the signal i_v is not measurable, it is very difficult to know the width I_1 of the dead zone exactly. In practice, the width is often estimated by utilising the measurable signals i and y. There are some practical estimation method available (see, for example, Tao and Kokotovic, 1996).

4. OPTIMAL PID CONTROLLER DESIGN

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. 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 response of a standard second-order system is widely used to represent these requirements. Its transfer function is

$$G_d(s) = \frac{\omega_n^2}{s^2 + 2\varsigma\omega_n s + \omega_n^2} \quad (10)$$

where the parameter ω_n is natural frequency, 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:

$$J(K_P, K_I, K_D) = \int_0^\infty (y_{step}(t) - y_{step}^d(t))^2 dt \quad (11)$$

where $y_{step}^d(t)$ is the desired step response which may be produced by the transfer function $G_d(s)$, and $y_{step}(t)$ the step response of the system with the PID controller. Thus, the optimal PID controller design may be stated as

$$\min_{K_P, K_I, K_D} J(K_P, K_I, K_D) \quad (12)$$

There are a number of optimisation methods to solve the above problem (Liu *et al.*, 2001). Many optimisation algorithms now exist in standard libraries of optimisation software, for example, the optimisation toolbox for use with MATLAB (Grace, 1994).

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 4. It mainly consists of four parts: model parameter estimation, desired system specifications, optimal-tuning mechanism and nonlinear 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 second-order system. The optimal-tuning mechanism finds optimal parameters for the PID controller so that the desired system specifications are satisfied.

The operating procedure of the optimal-tuning PID control is as follows. When the system operating point or dynamics change, 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 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.

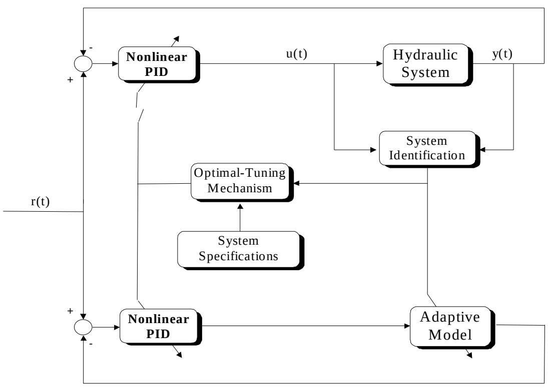

Figure 4 Optimal-tuning PID control scheme

To implement the optimal-tuning PID control in practice safely, a control performance prediction scheme is integrated, as shown in Figure 4. It mainly comprises an external control loop and an internal control loop. The external control loop includes the hydraulic system and a nonlinear PID controller. The internal control loop includes the adaptive model updated by an on-line system identification algorithm and a nonlinear PID controller.

The parameters of the two PID controllers are adjusted by the optimal-tuning PID algorithm. Roughly speaking, implementation of this PID control strategy consists of two stages. The first stage is to run a nonlinear 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.

6. PID CONTROL DESIGN TOOLBOX

The PID control design toolbox has specially been developed in MATLAB to enable the designer to have flexibility in the off-line design of PID controllers for the hydraulic test rig. The toolbox has two sub-menus, as shown in

Figure 5. One is the function menu and the other is the set-up menu. The former mainly covers real-time data acquisition, non-linear system identification, model validity test, optimal PID controller design, off-line control performance prediction, desire and real response comparison and PID parameter display. The latter mainly includes the setting of the reference input, desired time response specifications, PID initial proportional, integral and derivative gains.

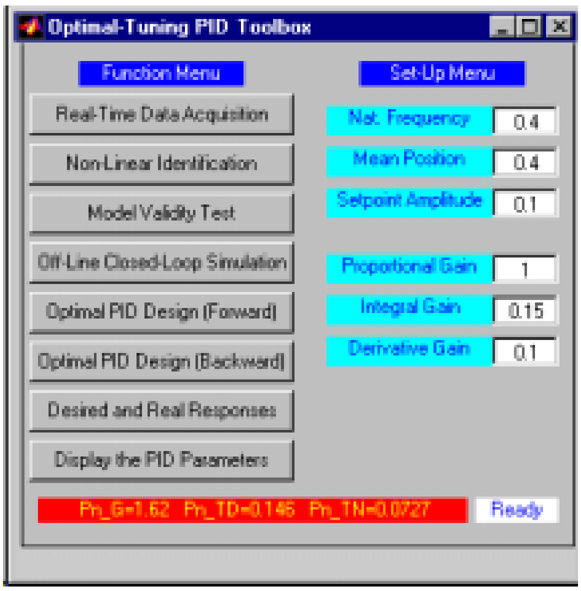

Figure 5 Optimal-tuning PID toolbox menu.

The function for the real-time data acquisition collects real-time input-output data from the real plant through AD/DA converters.

The function for the non-linear system identification uses the input-output data of the hydraulic system to estimate the parameters of a non-linear model, which represents the hydraulic system quite well.
The function for the model validity test provides test results of the non-linear model, which is estimated by the non-linear system identification, using a set of input-output data which are not used for the parameter estimation.
The function for the optimal controller design searches the optimal PID parameters in forward and backward directions of the hydraulic system to satisfy the desired time response specifications, based on the identified non-linear model.
The function for the off-line control performance prediction predicts the closed-loop performance of the hydraulic system when a PID controller is applied. It considers various nonlinearities which there exists in the systems, such as control input saturation, non-linear model, and sensor nonlinearities.
The function for the desired and real responses compares the real response of the system using the optimal PID parameters with the desired response to show how close both two responses.
The function for PID parameter display shows the latest PID parameters in the PID parameter display box.
The function for the reference input setting gives the shape of the square reference input by typing in the frequency, amplitude and mean level of the signal.
The function for the desired time response specifications determines the desired time response of the closed-loop system by typing in the natural frequency and/or damping ratio of the first- or second-order system.
The function for proportional, integral and derivative gains makes it possible to change the PID parameters manually.

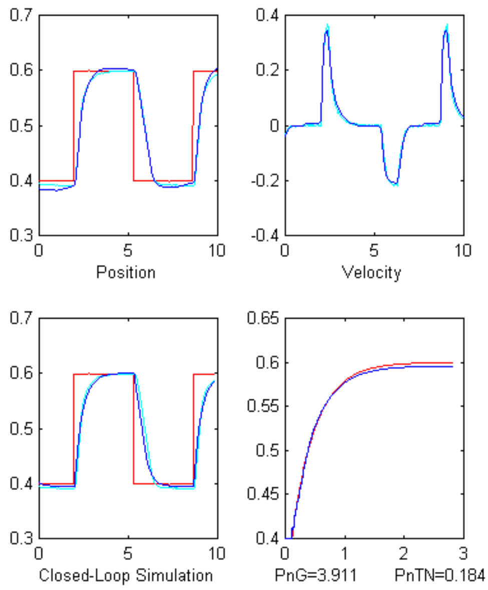

Figure 6 Display window of the toolbox.

The toolbox also provides a interface window to display the design performance, as shown in Figure 6. The non-linear identification results on both displacement and velocity are given in the two top subfigures in Figure 6, where the grey lines denote the real responses of the system and the dark lines the responses of the model. They are also used for displaying validity test results. The off-line closed-loop simulation performance is shown in the left lower subfigure in Figure 6. The progress of the optimisation design for PID parameters for the forward direction is given in the right lower subfigure in Figure 6.

7. EXPERIMENTAL RESULTS

To effectively assess the performance of the proposed tuning method, three experimental cases have been considered. For all three cases, the amplitude, period and dc offset of the square-wave reference input are 7.5mm, 5 seconds and 75mm, respectively. Here, only optimal-tuning PID controller for the forward direction is illustrated.
Case A: The parameters of the PID controller 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 7. It is clear that the responses are away from the desired response.

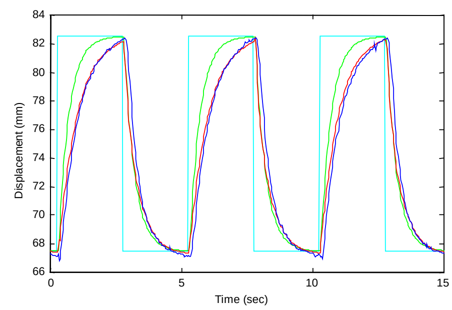

Figure 7 Responses of the plant, model and tracking model (Case A)

Case B: The optimal-tuning mechanism was switched on. The optimal PID parameters were found to be $K_P=1.923$, $K_I=0.01$ and $K_D=0.02$. This optimal PID controller was applied to the model. The closed-loop response of the model with the optimal PID controller is very close to the desired response, as shown in Figure 8.

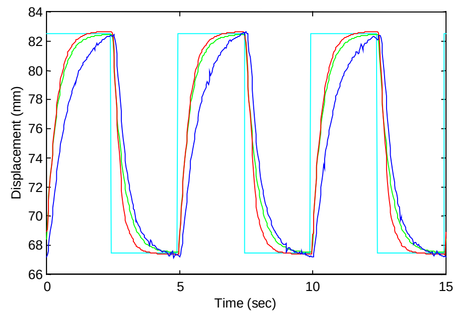

Figure 8 Responses of the plant, model and tracking model (Case B)

Case C: The optimal PID controller ($K_P=1.923$, $K_I=0.01$ and $K_D=0.02$) This optimal PID was applied to the plant. It can be seen from Figure 9 that the closed-loop responses of both the plant and the model with the optimal PID controller are very close to the desired response.

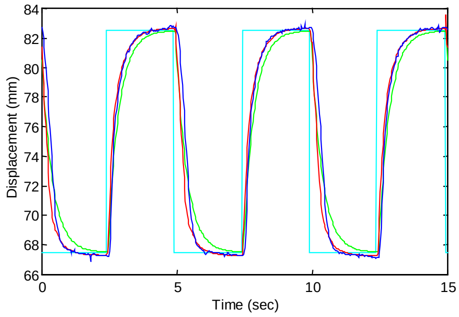

Figure 9 Responses of the plant, model and tracking model (Case C)

8. DISCUSSIONS AND CONCLUSIONS

The optimal-tuning PID controller design has been considered for the hydraulic position control system. Several design techniques have been addressed. Since the dynamics of the system for each movement direction is different, a non-linear direction dependent ARX model was used. As there exists dead-zone in the hydraulic position system, it is difficult to achieve high-precision tracking using only linear controllers. A non-linear PID controller, which includes the inverse of dead zone, was introduced to cancel the dead zone.

The optimal-tuning PID design strategy mainly consists of four parts: non-linear model estimation, definition of desired 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 a desired second-order system. The optimal-tuning mechanism finds optimal parameters for the PID controller so that the desired specifications are satisfied. To implement the optimal-tuning PID control safely, a control performance prediction scheme has been integrated. It mainly comprises an external control loop and an internal control loop. A PID controller runs firstly in the internal loop to predict its performance before it is used in the real plant. In this way, unnecessary damage which could result from the wrong PID parameters can be prevented. The PID control design toolbox written in MATLAB enables the designer to have flexibility in the off-line design of PID controllers for hydraulic systems. These techniques above have been successfully applied to the hydraulic position control system. The experimental results have shown that they can significantly improve system performance.

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