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Optimal-tuning PID control for industrial systems

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Abstract

Three optimal-tuning PID controller design schemes are presented for industrial control systems in this paper. They are time-domain optimal-tuning PID control, frequency-domain optimal-tuning PID control and multiobjective optimal-tuning PID control. These schemes can provide optimal PID parameters so that the desired system specifications are satisfied even in case where the system dynamics are time variant or the system operating points change. They are applied to three industrial systems, a hydraulic position control system, a rotary hydraulic speed control system and a gasifier, respectively.

Keywords: Industrial systems; Multiobjective control; Optimal-tuning; PID control

1. Introduction

The PID control algorithm remains the most popular approach for industrial process control despite continual advances in control theory. This is not only due to the simple structure which is conceptually easy to understand and, which makes manual tuning possible, but also to the fact that the algorithm provides adequate performance in the vast majority of applications. Most of the PID tuning rules developed in the last 50 years use frequency-response methods. Examples include, Ziegler–Nichols rule (Ziegler & Nichols, 1942), symmetric optimum rule (Kessler, 1958; Voda & Landau, 1995), Ziegler–Nichols' complementary rule (Mantz & Tacconi, 1989), some-overshoot rule (Seborg, Edgar, & Mellichamp, 1989), no-overshoot rule (Seborg et al., 1989), refined Ziegler–Nichols rule (Hang, Astrom, & Ho, 1991), integral of squared time weighted error rule (Zhuang & 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 enough tuning or produce a satisfactory closed-loop response. For example, in practice, the Ziegler–Nichols rule often leads to a rather oscillatory response to setpoint changes.

In fact, to have required system performance specifications, PID controller design of industrial systems is complicated by a number of factors:

(a) The system has non-linearities such as directionally dependent actuator and plant dynamics;
(b) Various uncertainties, such as modelling error and external disturbances, are involved in the system.
(c) Sub-optimal tuning may be necessary to cater for changes in the system with time such as ageing and general wear.
(d) Commissioning is easiest without load, but the load is often variable and affects the dynamic performance.

As a result of these difficulties, the PID controllers are rarely tuned optimally and the engineers will need to settle for a compromise performance given the time available for the exercise. This makes system tuning more subjective, with the potential for different engineers to achieve different, sub-optimal performance from the same equipment. Poor tuning can lead to

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mechanical wear associated with excessive control activity, poor control performance and even poor quality products.

To improve the performance of PID tuning for processes with changing dynamic properties, several tuning strategies have been proposed, for example, automatic tuning PID (Astrom & Hagglund, 1984) and adaptive PID (Kraus & Mayron, 1984). These controllers have self-initialisation and recalibration features to cope with little a priori knowledge and significant changes in the process dynamics, based on the automatic measurement of the ultimate gain and period. Various techniques, such as relay excitation feedback (Astrom & Hagglund, 1984) and rule-based autotuning (McCormack & Godfrey, 1998), have been developed. However, the PID controller parameters are still computed using the classic tuning formulae and, as noted above, these do not provide good control performance in all situations.

In order to provide consistent, reliable, safe and optimal solution to industrial control problems, three novel optimal-tuning PID control schemes are presented in this paper. They are the time-domain optimal-tuning PID control, the frequency-domain optimal-tuning PID control and the multiobjective optimal-tuning PID control. These schemes generally consist of four basic parts: model estimation, desired system specifications, optimal-tuning mechanism and an online PID controller. The model estimation provides a parametric/non-parametric model for the process. The desired system specifications include a set of control requirements of the process. The optimal-tuning mechanism finds optimal parameters for the PID controller so that the desired system specifications are satisfied. These optimal-tuning PID control schemes are demonstrated through their applications to a hydraulic position control system, a rotary hydraulic speed control system and a gasifier.

2. Time-domain optimal-tuning PID control

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 (Daley & Liu, 1998). They are the first- and second-order systems, whose transfer functions are

$$G_1(s) = \frac{\omega_0}{s + \omega_0}, \tag{1}$$

$$G_2(s) = \frac{\omega_n^2}{s^2 + 2\varsigma\omega_n s + \omega_n^2}, \tag{2}$$

where the parameters ω_0 and ω_n are natural frequencies, and ς the damping ratio. In the time domain, the PID controller is assumed to be of the form:

$$u(t) = K_p\left(e(t) + K_d\frac{\mathrm{d}y(t)}{\mathrm{d}t}\right) + K_i\int_0^t e(t)\,\mathrm{d}t, \tag{3}$$

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 $y(t)$, and $u(t)$ is the control input.

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:

$$J(K_p, K_i, K_d) = \int_0^\infty (y_{step}(t) - y_{step}^d(t))^2\,\mathrm{d}t, \tag{4}$$

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

$$\min_{K_p, K_i, K_d} J(K_p, K_i, K_d). \tag{5}$$

Statistical considerations show that the performance function J is the most appropriate choice for data fitting when errors in the data have a normal distribution. The function J is often preferred because it is known that the best fitting calculation is straightforward to solve. But, the solution to the minimisation equation (5) may not give good control results for the system. For this case, some other performance functions should be taken into account. For example, if the control error between the desired output and actual output is greater than a tolerated value, its L_∞-norm function may need to be considered during the controller design.

When a system has different operating points with widely different dynamic properties, it is not always possible to exercise control with a fixed parameter controller, even if this is a highly robust controller. A time-domain optimal-tuning PID control scheme is proposed as shown in Fig. 1. It mainly consists of model parameter estimation, desired system specifications, optimal-tuning mechanism, an online PID controller and control performance prediction. The model parameters are estimated using identification methods, e.g., the least squares algorithm. 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 time-domain system specifications are satisfied. The online PID controller is connected directly to the real plant. The control performance prediction, which is composed of an adaptive model of the plant and an offline PID controller, is used to predict the closed-loop control performance of the system with the PID parameters which are found by the optimal-tuning mechanism. The operating procedure of the time-domain 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 the updated model parameters, the tuning mechanism searches for the optimal parameters for the PID controller to satisfy the desired system specifications. Before the online PID is updated by the new optimal parameters, these parameters are tested in the control performance prediction loop. Finally, the PID controller is set to have the obtained optimal parameters that have been verified as being safe on the adaptive model. In this way, any unnecessary damage resulting from the wrong PID parameters can be prevented.

The time-domain optimal-tuning PID control was applied to the hydraulic position control test rig in Fig. 2. The rig has two hydraulic cylinders: main cylinder and loading cylinder, which are coupled by a shaft. An ALSTOM 3 way proportional valve controls the main cylinder. The loading cylinder is controlled by a Moog servo valve to simulate variations in load. This test rig is used to represent a part of 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.

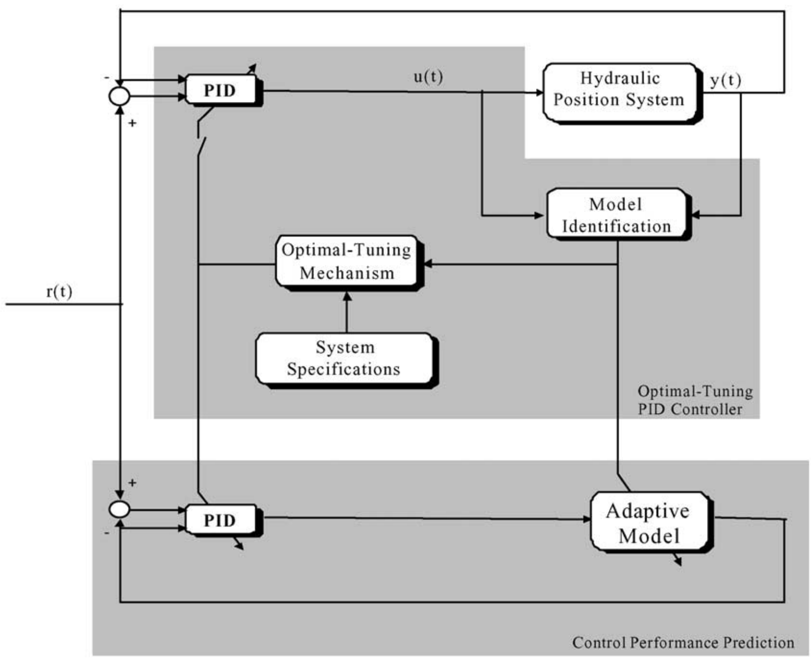

Fig. 1. The time-domain optimal-tuning PID control scheme.

Fig. 2. The hydraulic position test rig.

The optimal-tuning PID control scheme depicted in Fig. 1 is implemented using ALSTOM's DIGIPID 1000 governor with MATLAB and SIMULINK. The main control aim is to produce a positioning loop that is as fast as possible without overshoot. Because of the potential for damage, a key issue is implementation safety. As a result the decision whether to implement a set of optimal parameters is left with the operator or commissioning engineer. Only when the operator is satisfied with the accuracy of the model and suitability of the optimal parameters is the main loop controller (in this case ALSTOM's DIGIPID 1000 governor) modified.

Because of the restriction on overshoot the specification is made using a first order response. To effectively assess the performance of the proposed tuning method, three experimental cases have been considered. The tracking model is a first order system with $\omega_0 = 2.5$. The step change of the reference input is from 67.5 to 82.5 mm.

Case A: The parameters of the PID controller were $K_p = 0.8$, $K_i = 1e - 5$ and $K_d = 0.05$. This controller was applied to both the plant and the model. The closed-loop step responses of the plant and the model are given in Fig. 3. It is shown clearly that the plant and model responses are almost the same, which implies that the model represents the plant very well. But, the plant response is significantly different from the desired response.

Case B: The optimal-tuning mechanism was switched on. The optimal PID parameters were found to be $K_p = 1.923$, $K_i = 2e - 5$ and $K_d = 0.02$. This optimal PID controller was applied to the performance prediction loop only. The closed-loop response of the model with the optimal PID controller is very close to the desired response, as shown in Fig. 4.

Case C: The optimal PID controller ($K_p = 1.923$, $K_i = 2e - 5$ and $K_d = 0.02$) was applied to the plant. It can be seen from Fig. 5 that the closed-loop responses of both the plant and the model with the optimal PID controller are very close to the desired response.

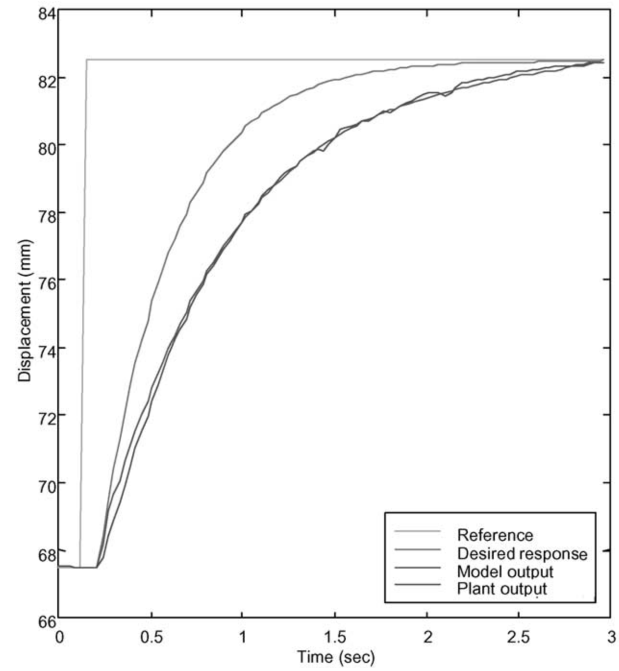

Fig. 3. Responses of the plant, model and reference model (Case A).

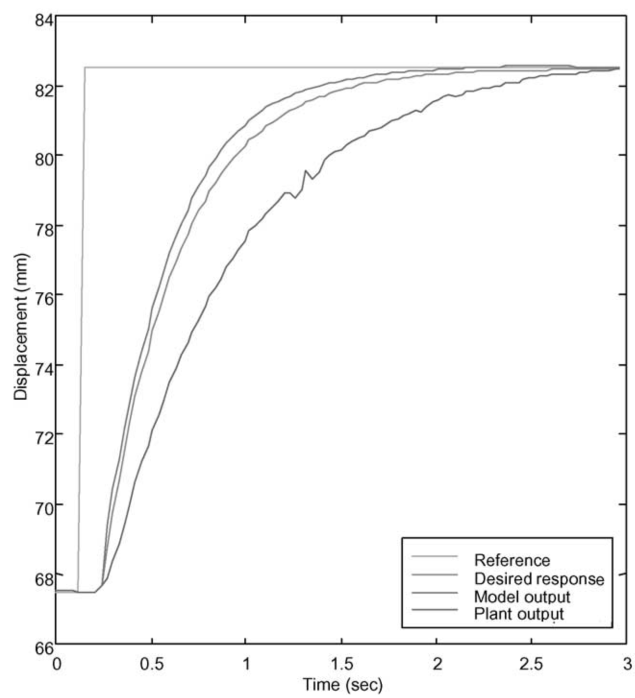

Fig. 4. Responses of the plant, model and reference model (Case B).

3. Frequency-domain optimal-tuning PID control

The design of feedback control systems in industry using frequency-response methods is more popular than any other. This is primarily because the frequency-response method provides good designs in the face of uncertainty in the plant model and can easily use experimental information for design purposes.

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

$$K(s) = K_c\left(1 + \frac{1}{T_i s} + T_d s\right), \tag{6}$$

where K_c, T_i and T_d are the PID parameters. Although the traditional PID design methods give simple tuning rules for the controller parameters using either one or two measurement points of the system frequency-response, their control performance may not satisfy the desired requirements. To overcome this disadvantage, an optimal PID controller design is proposed in the frequency-domain.

In the frequency-domain, there are two quantities used to measure the stability margin of the system. One is the gain margin, which is the factor by which the gain is less than the neutral stability value. The other is the phase margin, which is the amount by which the phase of the system exceeds -180° when the system gain is unity. The gain and phase margins are also related to the damping of a system. In addition to the stability of a design, the system is also expected to meet a speed-of-response specification like bandwidth. The crossover frequency, which is the frequency at which the gain is unity, would be a good measurement in the frequency domain for the system's speed of time response. Also, the larger the value of the magnitude on the low-frequency asymptote, the lower the steady-state errors will be for the closed-loop system. This relationship is very useful in the design of suitable compensation. Thus, the following performance functions need to be considered during the design of a PID controller (Liu & Daley, 1999):

$$\phi_1(K_c, T_i, T_d) = \left\{ \frac{|K(j\omega)G(j\omega)|}{G_M}, \quad \angle K(j\omega)G(j\omega) = -180^\circ \right\}, \tag{7}$$

$$\phi_2(K_c, T_i, T_d) = \left\{ 2 - \frac{180^\circ + \angle K(j\omega)G(j\omega)}{P_M}, \quad |K(j\omega)G(j\omega)| = 1 \right\}, \tag{8}$$

$$\phi_3(K_c, T_i, T_d) = \left\{ \frac{\omega}{2\pi f_d}, \quad |K(j\omega)G(j\omega)| = 1 \right\}, \tag{9}$$

$$\phi_4(K_c, T_i, T_d) = \left\{ \frac{1}{e_{ss}|1 + K(j\omega)G(j\omega)|}, \quad \omega = 0 \right\}, \tag{10}$$

where $\phi_i(K_c, T_i, T_d)$, for $i = 1, 2, 3, 4$, are the normalised gain-margin, phase-margin, crossover-frequency and the steady-state error functions with the desired values G_M, P_M, f_d and e_{ss}, respectively. Thus, the optimal PID controller may be obtained by

$$\min_{K_c, T_i, T_d} \sum_{i=1}^{4} w_i(\phi_i(K_c, T_i, T_d) - 1)^2, \tag{11}$$

where w_i is the weighting factor. The choice of the weighting factor is used to balance the effect of the performance functions to the closed-loop system. Usually, the weighting factor is chosen by the designer using the trial and error method. In this paper, all weight factors are simply set to be 1, i.e. $w_i = 1$, for $i = 1, 2, 3, 4$.

A frequency-domain optimal-tuning PID control scheme is proposed, as shown in Fig. 6. It mainly consists of four parts: frequency-response estimation, desired system specifications, optimal-tuning mechanism and PID controller. The frequency-response estimated using frequency-domain identification methods provides a non-parametric model for the process. The desired

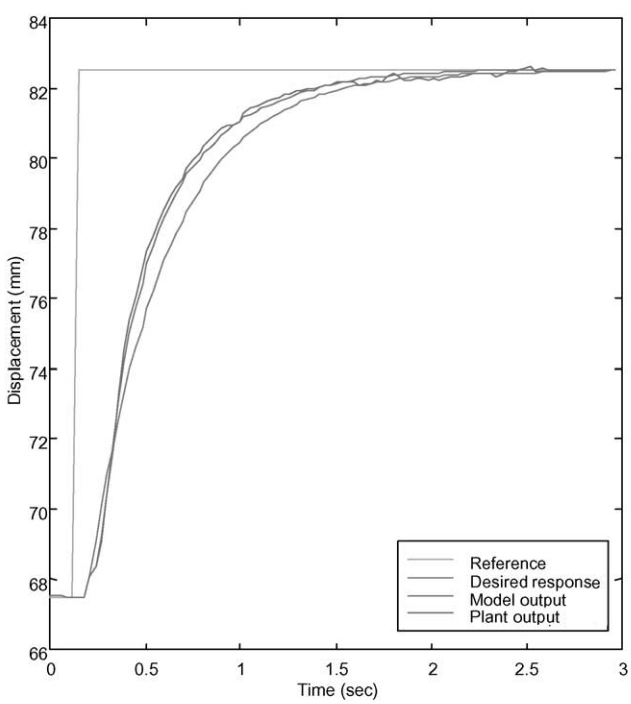

Fig. 5. Responses of the plant, model and reference model (Case C).

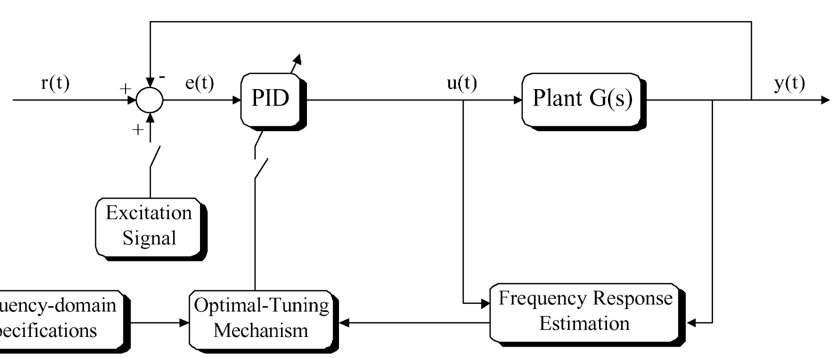

Fig. 6. Frequency-domain optimal-tuning control scheme.

system specifications include a set of requirements in the frequency domain: gain margin, phase margin, crossover frequency and steady-state error. The optimal-tuning mechanism uses the process frequency-response to find 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's operating-point or dynamics change, the new process frequency-response is re-estimated by switching on the excitation signal that is added to the reference input. Then, using this updated frequency-response, 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 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.

The optimal-tuning PID control scheme was applied to a rotary hydraulic test rig, as shown in Fig. 7, which is representative of many industrial systems that 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, by virtue of the power available. The rotary hydraulic test rig comprises an electro-hydraulic servo control valve driving a fixed displacement hydraulic motor up to 8000 rpm with a maximum operating pressure of 21 MPa. The motor is coupled by a rigid shaft to a hydraulic pump of the same displacement as the motor and a solenoid controlled relief valve is used to simulate variations in load. This type of hydraulic system is typically applied to mixer drives, centrifuge drives and machine tool drives where accurate speed control with fast response times is required, and large changes in load can be expected.

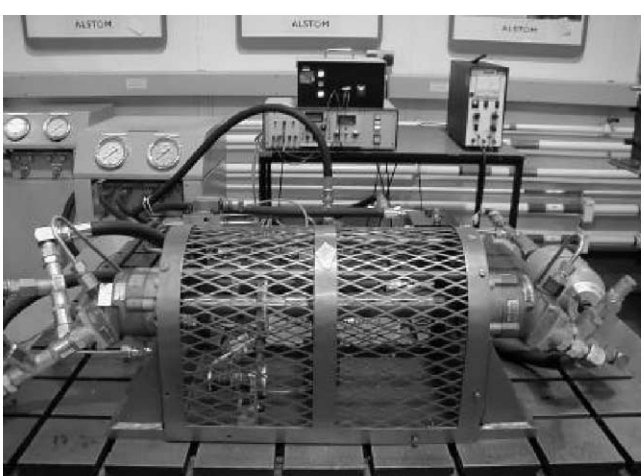

Fig. 7. Rotary hydraulic speed control test rig.

The implementation of the frequency-domain optimal-tuning PID control for the hydraulic speed control system is carried out using the MathWorks Real-Time Workshop connected to a dSPACE DSP board based around the TMS320, MATLAB and SIMULINK. During the experiment, the rotary hydraulic system was operated in two working conditions: without load and with load. Also, four PID control rules were compared, which are Ziegler–Nichols rule (Ziegler & Nichols, 1942), integral of absolute error rule (Pessen, 1994), symmetric optimum rule (Kessler, 1958; Voda and Landau, 1995) and optimal-tuning rule proposed in this section.

Case I (without load): A small periodic multi-sine excitation signal was directly toped on to the reference input. Based on the input–output data, the frequency-response of the system was estimated using the frequency-domain identification methods. The estimated magnitude and phase of the system with respect to frequency are obtained. Following the four PID tuning rules provides the parameters which are given in Table 1. The speed responses of the hydraulic motor using four PID controllers are shown in Fig. 8.

Case II (with load): In this case, there was load on the hydraulic motor. This means the system dynamics are different. When the PID parameters for Case I were still used, two PID controllers failed to stabilise the system,

Table 1
PID parameters for case I

PID rule	K_c	T_i	T_d
Ziegler-Nichols rule (ZNR)	0.1012	0.0549	0.0137
Integral of absolute error rule (IAER)	0.1180	0.0439	0.0165
Symmetric optimum rule (SOR)	0.0844	0.0994	0.0056
Optimal-tuning rule (OTR)	0.0317	0.0262	0.0385

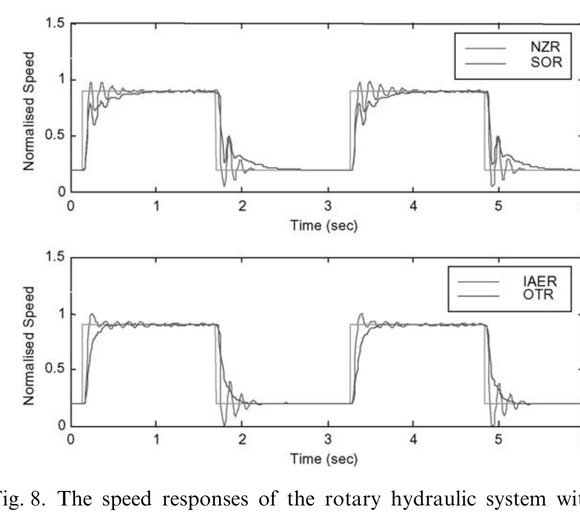

Fig. 8. The speed responses of the rotary hydraulic system without load.

namely the Ziegler–Nichols and Integral of absolute error rules. The speed responses using the other two PID control rules are given in Fig. 9. Note that the symmetric optimum rule gives a significantly oscillatory response. Though the optimal-tuning rule does not give a speed response as good as that for Case I, its performance is reasonably good and for some practical cases may be considered acceptable.

Since the system dynamics has changed significantly a more appropriate response would be to re-tune. The periodic multi-sine excitation signal was applied to the hydraulic system again. The estimated frequency response of the load case was re-estimated. In terms of the four PID tuning rules, the parameters of the PID controllers for this case are given in Table 2. The performance results of the four PID controllers for this case are shown in Fig. 10. It is clear that the relative performance is similar to that displayed for Case I except the IAER PID controller causes an oscillation in

Table 2
PID parameters for case II

PID rule	K_c	T_i	T_d
Ziegler-Nichols rule (ZNR)	0.1662	0.0606	0.0152
Integral of absolute error rule (IAER)	0.1939	0.0485	0.0182
Symmetric optimum rule (SOR)	0.1268	0.1141	0.0065
Optimal-tuning rule (OTR)	0.0632	0.0372	0.0351

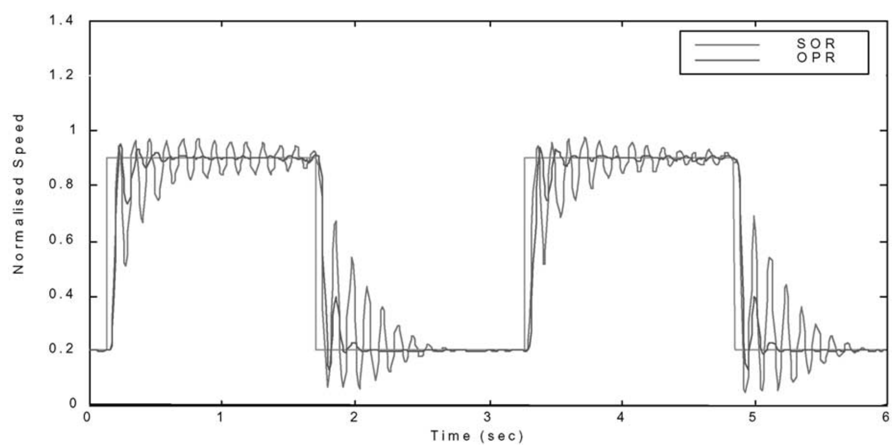

Fig. 9. The speed responses of the rotary hydraulic system with load.

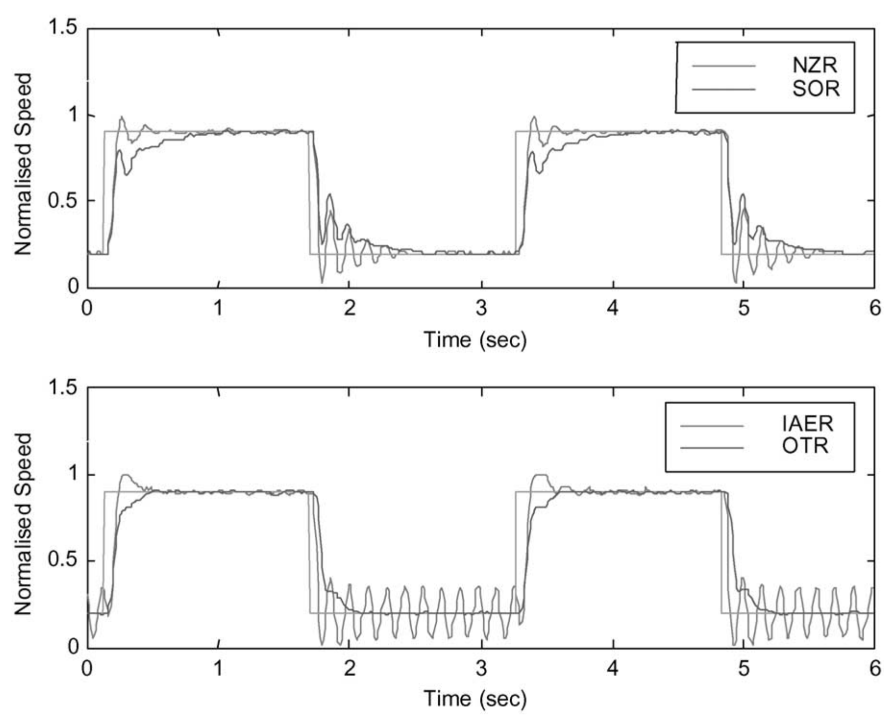

Fig. 10. The speed responses of the rotary hydraulic system with load.

the low speed range, which shows the system has non-linearities.

4. Multiobjective optimal-tuning PI control for a gasifier

Integrated gasification combined cycle (IGCC) power plants are being developed around the world to provide environmentally clean and efficient power generation from coal. To provide environmentally clean and efficient power generation from coal, a gasifier plays an important role in IGCC power plants. The gasifier is based on the spouted fluidised bed gasification concept (originally developed by British Coal CTDD) and can be considered as a reactor where coal is gasified with air and steam. In the ALSTOM benchmark challenge on gasifier control (Dixon, Pike, & Donne, 1998), the gasifier, as shown in Fig. 11, is a non-linear multivariable system, having four inputs (coal, limestone, air and steam) and four outputs (pressure, temperature, bed mass and gas quality) with a high degree of cross coupling between them (Donne, Dixon, Pike, Odeku, & Ricketts, 1998). The aim of the benchmark challenge is to design a controller to satisfy a set of specifications on the system outputs, inputs and input rates when a step pressure disturbance is applied to the system, based on a linearised model of the gasifier at the 100% load operating point. In addition, the challenge requires that the controller also be evaluated on models representing the gasifier at 50% and 0% load operating points and on rejection to a sine-wave/step pressure disturbance applied to the gasifier at the above three operating points.

There are a wide variety of control techniques which can be used to design a controller for the gasifier. Here, a multiobjective optimally-tuning PI controller is considered for the gasifier. A set of multiobjective performance criteria based on the specifications for the system outputs, inputs and input rates are formulated as functions of the parameters of the multi-input multi-output (MIMO) PI controller. The controller is designed using multiobjective optimisation methods and also evaluated on three linearised gasifier models.

Since the gasifier outputs, inputs and input rates are constrained, a set of performance functions should be considered during the design of the controller. For this purpose, the following performance functions of the m-input r-output gasifier with n states are introduced.

$$\phi_i(K) = \frac{|\tilde{y}_i|}{y_i^d}, \quad i = 1, 2, \ldots, r, \tag{12}$$

$$\phi_{r+j}(K) = \frac{|\tilde{u}_j|}{u_j^d}, \quad j = 1, 2, \ldots, m, \tag{13}$$

$$\phi_{r+m+k}(K) = \frac{|\dot{u}_k|}{\delta u_k^d}, \quad k = 1, 2, \ldots, m, \tag{14}$$

$$\phi_{r+2m+1}(K) = 1 + \max_{j=1,2,\ldots,n} \{\mathrm{Re}(\lambda_j)\} + \varepsilon, \tag{15}$$

where $\tilde{y}_i$ is the ith output fluctuation and $\tilde{u}_j$ the jth input fluctuation, the parameter vector K denotes the parameters of the PI controller, λ_j is the jth poles of the closed-loop system, Re(.) denotes the real part of a number, the positive real number ε represents the requirements on the closed-loop poles, y_i^d, u_j^d, δu_k^d are the upper bounds on the output fluctuation, input fluctuation and input rates, respectively. According to the requirements on the gasifier, the following multi-objective performance criteria should be satisfied:

$$\phi_i(K) \leqslant 1, \quad i = 1, 2, \ldots, r + 2m + 1. \tag{16}$$

If the above inequalities are satisfied, then the problem is solved. Thus, the design problem is to find a PI controller to make (16) hold. If the above inequalities are met, then the problem is solved. Clearly, the design problem is to find a PID controller to make (16) hold. There are a number of methods to solve the performance criteria problem (16), for example, the minimax optimisation method (Gill et al., 1981) and the method of inequalities (Zakian & Al-Naib, 1973; Liu, 1992).

The linearised model of the gasifier at the 100% load operating point was used for the gasifier control design. This model has twenty-five state variables ($n = 25$), four control inputs ($m = 4$) and the four outputs ($r = 4$). The disturbance d on the gasifier is the sink pressure which represents the pressure upstream of the gas turbine. Thus, the MIMO PI controller is a 4×4 PI controller, which consists of 16 P-parameters and 16 I-parameters. Also, there are 13 performance function criteria which need to be met during the controller design.

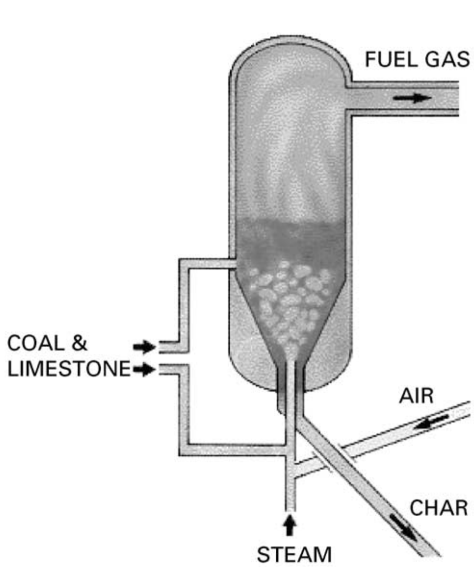

Fig. 11. The gasifier.

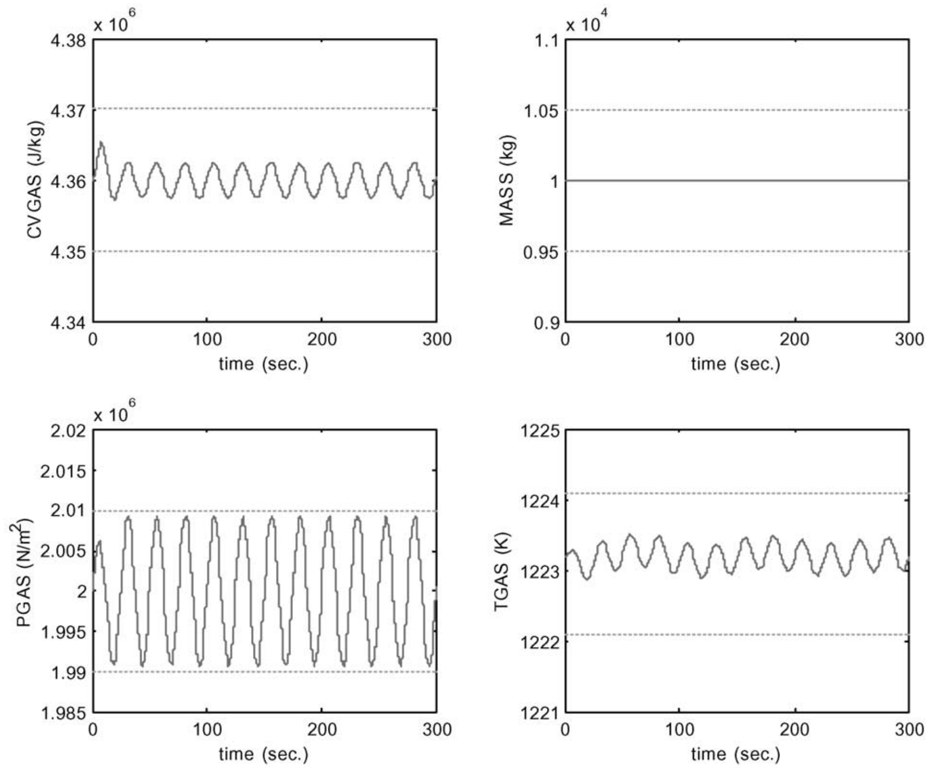

Fig. 12. Output responses to sine wave disturbance for 100% load.

Based on the linearised model of the gasifier at the 100% load operating points with a sine-wave pressure disturbance, all 13 performance function criteria were successfully satisfied by the 4×4 PI controller using the multiobjective optimisation algorithms provided by the optimisation toolbox for use with MATLAB (Grace, 1994). The outputs (fuel gas calorific value (CVGAS), bed mass (MASS), fuel gas pressure (PGAS) and fuel gas temperature (TGAS) of the gasifier at 100% load are shown in Fig. 12, where the dot lines represent the upper bounds and lower bounds of the output constraints. Clearly, the outputs of the gasifier are within the required bounds and have a good rejection to the sine-wave pressure disturbance.

The same controller that was designed for the 100% load operating point was also applied to the linearised models at the 50% load operating points. All performance requirements of the gasifier at 50% load are satisfied as well. But, some performance criteria at 0% load are violated by use of this controller. Thus, to cope with the 0% load case, the multi objective optimal-tuning mechanism was switched on again to adapt the change in operating point of the gasifier. The new optimal PI parameters resulted in similar control performance results as those at 100% load.

5. Conclusions

This paper has considered three optimal PID controller design schemes: the time-domain optimal-tuning PID control, the frequency-domain optimal-tuning PID control and the multiobjective optimal-tuning PID control. These schemes mainly consist of four basic parts: model estimation, a definition of desired system specifications, an optimal-tuning mechanism and a PID controller. Different kinds of performance functions for the three control schemes were employed. These schemes have been successfully applied to three industrial systems: the hydraulic position control system, the rotary hydraulic speed control system and the gasifier. The experimental results have shown that the optimal-tuning PID controller can significantly improve system performance, and copes well with changes in the process dynamics.

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