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Tuning of PID controllers for boiler-turbine units

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Abstract

A simple two-by-two model for a boiler-turbine unit is demonstrated in this paper. The model can capture the essential dynamics of a unit. The design of a coordinated controller is discussed based on this model. A PID control structure is derived, and a tuning procedure is proposed. The examples show that the method is easy to apply and can achieve acceptable performance. © 2004 ISA—The Instrumentation, Systems, and Automation Society.

Keywords: Boiler-turbine unit; PID tuning; Robustness; Industrial applications

1. Introduction

A common way to generate electric power is to use drum boilers to produce steam and make the steam drive turbogenerators to generate electricity. Two types of configurations exist for this purpose:

1. A header is used to accommodate all the steam produced from several boilers, and the steam is then distributed to several turbines through the header. The capacity of the boilers used in this configuration is usually small. The steam can be used for generating electricity as well as other purposes. This configuration is commonly used in industrial utility plants.
2. A single boiler is used to generate steam that is directly fed to a single turbine. This configuration is usually called a boiler-turbine unit. The capacity of the boilers used in this configuration is very large compared with the first configuration.

In this paper we will concentrate on a boiler-turbine unit since this configuration is more common in a modern power plant due to the possibly quick response to the electricity demand from the power grid or network. The control system for a boiler-turbine unit usually needs to meet the following requirements:

- Electric power output must be able to follow the demand by the dispatch.
- Throttle pressure must be maintained despite variations of the load.
- The amount of water in the steam drum must be maintained at a desired level to prevent overheating of the drum or flooding of steam lines.
- Steam temperature must be maintained at a desired level to prevent overheating of the superheaters and to prevent wet steam from entering turbines.
- The mixture of fuel and air in the combustion chamber must meet standards for safety, efficiency, and environment protection, which is usually accomplished by maintaining a desired level of excess oxygen.

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Nomenclature

Parameters	Description
B	Boiler firing rate (fuel demand)
μ	Governor valve position
N	Electricity generated
P_T	Throttle pressure
P_D	Drum pressure
S_G	Steam generation
S_F	Turbine steam flow
C_B	Boiler storage constant
K_{SH}	Superheater friction drop coefficient

To fulfill the complex control objectives listed above, the control system for a power plant is usually divided into several subsystems [1]. For example, the feedwater control subsystem is used to regulate the drum level; the temperature control subsystem is used to regulate the steam temperature; and the air control subsystem is used to regulate the excess oxygen. Since the couplings between the drum level, the steam temperature and the excess oxygen are not strong, these three subsystems can be designed independently. Thus the boiler-turbine unit can be modeled as a 2×2 system. The two inputs are boiler firing rate (or fuel flow rate, assuming air flow rate is regulated well by air control subsystem) and governor valve position. The two outputs are electric power and throttle pressure.

Two conventional techniques for the control of a boiler-turbine unit are:

1. Boiler follows turbine (BFT). The governor valve is responsible for following the power demand and the firing rate for controlling the throttle pressure.
2. Turbine follows boiler (TFB). The firing rate is responsible for following the power demand and the governor valve for controlling the throttle pressure.

Note that both methods use single-loop controllers but different pairs for control. Since the throttle pressure and the electric power are tightly coupled, an advanced control techniques might give better performance than a decentralized one. This control technique is called ‘‘coordinated control’’ in power plants since it coordinates the control inputs based on both the electric power demand and the throttle pressure.

The controller design for a boiler-turbine unit has attracted much attention in the past years. Modern control techniques have been applied to improve unit performance, e.g., LQG/LTR [2], H_∞ control [3,4]. predictive control [5–9], and fuzzy control [10,11]. These results are encouraging; however, conventional PID controllers are easier and quicker to implement.

Ref. [3] proposed a PID reduction procedure for a centralized controller and showed that the performance of the final PI controller for a boiler-turbine unit did not degrade much from the original loop-shaping H_∞ controller. Encouraged by this result, this paper will examine PID tuning for a boiler-turbine unit. PID tuning for a single-variable process is well known, e.g., Refs. [12–14], and there are papers discussing PID tuning for two-by-two processes, e.g, Refs. [15–18]. However, the dynamics for a boiler-turbine unit is different from the first-order plus deadtime dynamics discussed in those papers, and a literature search for PID tuning for a boiler-turbine unit did not yield results.

It should be noted that modern control techniques might achieve better performance than the proposed method, since our controller is a traditional PID controller. The comparison of PID with modern control techniques, such as MPC, LQG, H_∞, can be found in the open literature. Generally the advantage of a PID controller is its ease of implementation and tuning, while the advantage of a controller designed by modern techniques is its performance improvement. There is always a tradeoff between ease to use and cost to implement and tune.

In Section 2 a simple model for a boiler-turbine unit is derived. In Section 3 controller design for this model is discussed, and a control structure is found. A method is proposed to tune the parameters. Examples are given in Section 4 to illustrate the proposed tuning method, and conclusions are given in Section 5. Throughout this paper, Δ is used to denote the increment of a variable.

2. Simple boiler-turbine model for tuning

A first-order plus deadtime (FOPDT) model is often used for PID tuning for single-variable stable systems. The underlying idea is that this simple model can capture the essential dynamics of the system under consideration. So to study controller tuning for a boiler-turbine unit, it is

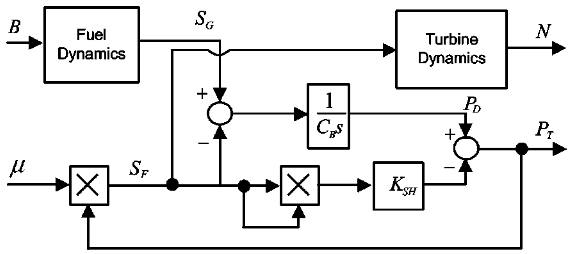

Fig. 1. A simple diagram of a boiler-turbine unit.

helpful to find a simple model that can capture the essential dynamics, especially the coupling effect between the generated electricity and the throttle pressure. However, the complete dynamics of a boiler-turbine unit are very complex and hard to model [19]. Cheres [20] and de Mello [21] proposed a nonlinear dynamic system with a simple structure to capture the essential dynamics of the boiler-turbine unit (Fig. 1).

The model in Fig. 1 shows the energy balance relation and the essential nonlinear characteristics of the boiler-turbine system.

- The energy balance relation: Drum pressure P_D relates the balance between the steam generation S_G and the turbine steam flow S_F:

$$\Delta S_G - \Delta S_F = C_B \frac{d\Delta P_D}{dt}. \tag{1}$$

- The two nonlinear characteristics are:

1. The pressure drop between the drum pressure P_D and the steam pressure P_T is related to the steam flow S_F by:
$$P_D - P_T = K_{SH} S_F^2. \tag{2}$$
2. The steam flow S_F is the product of the throttle pressure P_T and the turbine governor position μ:
$$S_F = \mu P_T. \tag{3}$$

Consider a linearized model of the boiler-turbine unit at a nominal operating point. Taking the increments on both sides of Eqs. (2) and (3), we have

$$\Delta P_D - \Delta P_T = R \Delta S_F, \tag{4}$$

$$\Delta S_F = \mu \Delta P_T + P_T \Delta \mu, \tag{5}$$

where $R = 2K_{SH} S_F$. Combining Eqs. (1), (4), and (5) we have

$$\begin{bmatrix} \Delta S_F \\ \Delta P_T \end{bmatrix} = \begin{bmatrix} \dfrac{1}{T_0 s + 1} & \dfrac{P_T C_B s}{\mu (T_0 s + 1)} \\ \dfrac{1}{\mu (T_0 s + 1)} & -\dfrac{P_T}{\mu} \dfrac{T_b s + 1}{T_0 s + 1} \end{bmatrix} \begin{bmatrix} \Delta S_G \\ \Delta \mu \end{bmatrix}, \tag{6}$$

where

$$T_0 := (1 + \mu R) C_B, \quad T_b := C_B R. \tag{7}$$

The fuel dynamics can be modeled as a first-order process,

$$\Delta S_G = \frac{k_1}{T_1 s + 1} \Delta B, \tag{8}$$

and the turbine dynamics can be modeled as

$$\Delta N = \frac{(\alpha T_2 s + 1) k_2}{T_2 s + 1} \Delta S_F, \tag{9}$$

where α is the ratio of the electric power generated by the high-pressure turbine to the total electric power generated by the turbine. Combining Eqs. (6), (8), and (9), a linearized model of a boiler-turbine unit at a certain operating point is obtained:

$$\begin{bmatrix} \Delta N \\ \Delta P_T \end{bmatrix} = \begin{bmatrix} \dfrac{m_{11}(\alpha T_2 s + 1)}{(T_1 s + 1)(T_0 s + 1)(T_2 s + 1)} & \dfrac{m_{12} s (\alpha T_2 s + 1)}{(T_0 s + 1)(T_2 s + 1)} \\ \dfrac{m_{21}}{(T_1 s + 1)(T_0 s + 1)} & -\dfrac{m_{22}(T_b s + 1)}{T_0 s + 1} \end{bmatrix} \times \begin{bmatrix} \Delta B \\ \Delta \mu \end{bmatrix}, \tag{10}$$

where

$$m_{11} := k_1 k_2, \quad m_{12} := \frac{P_T C_B k_1}{\mu}, \quad m_{21} := \frac{k_1}{\mu},$$

$$m_{22} := \frac{P_T}{\mu}. \tag{11}$$

Typical step responses for a boiler-turbine unit are shown in Fig. 2. The model is simple but captures the essential dynamics of the unit, and can serve as a base model for controller tuning.

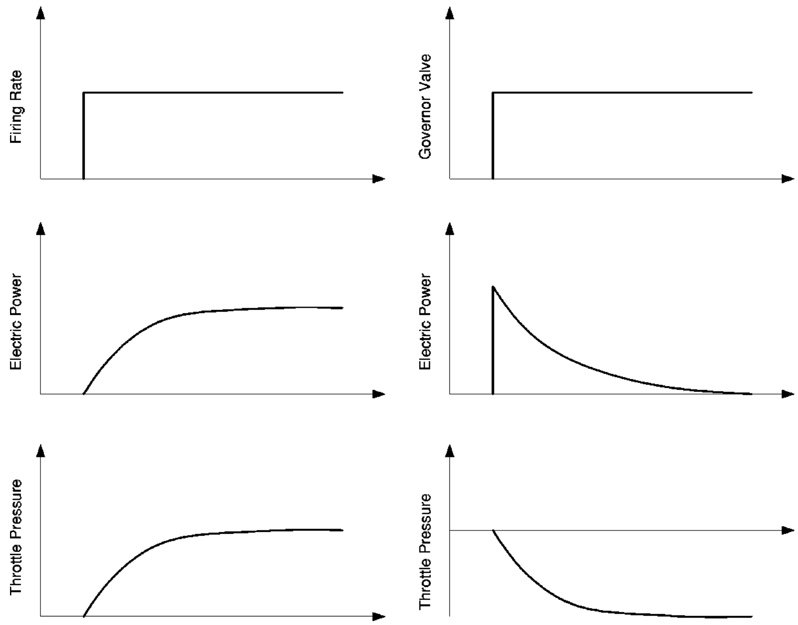

Fig. 2. Typical step responses for a boiler-turbine unit.

3. Design and tuning of coordinated PID controllers

3.1. Design

Consider a unity feedback system shown in Fig. 3, where G is the plant model, G_d is the disturbance model, and K is the controller.

It is well known that a well-designed control system should meet the following requirements besides the nominal stability:

- Set-point tracking,
- Disturbance attenuation,
- Robust stability and/or robust performance.

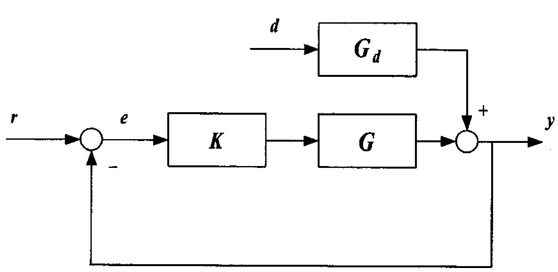

Fig. 3. Typical unity feedback configuration.

For a multivariable plant, set-point tracking requires that the system be decoupled. For a 2×2 plant, suppose the model and the controller are decomposed as

$$G=\begin{bmatrix} G_{11} & G_{12} \\ G_{21} & G_{22} \end{bmatrix}, \quad K=\begin{bmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{bmatrix}, \tag{12}$$

then open-loop system decoupling requires that GK is diagonal, i.e.,

$$G_{11}K_{12}+G_{12}K_{22}=0, \tag{13}$$

$$G_{21}K_{11}+G_{22}K_{21}=0. \tag{14}$$

So a complete decoupler for a 2×2 system takes the following form:

$$K=\begin{bmatrix} 1 & -G_{12}/G_{11} \\ -G_{21}/G_{22} & 1 \end{bmatrix}\begin{bmatrix} K_{11} & 0 \\ 0 & K_{22} \end{bmatrix}. \tag{15}$$

Now that the unit model is given by Eq. (10), a decoupler can then be designed according to Eq. (15),

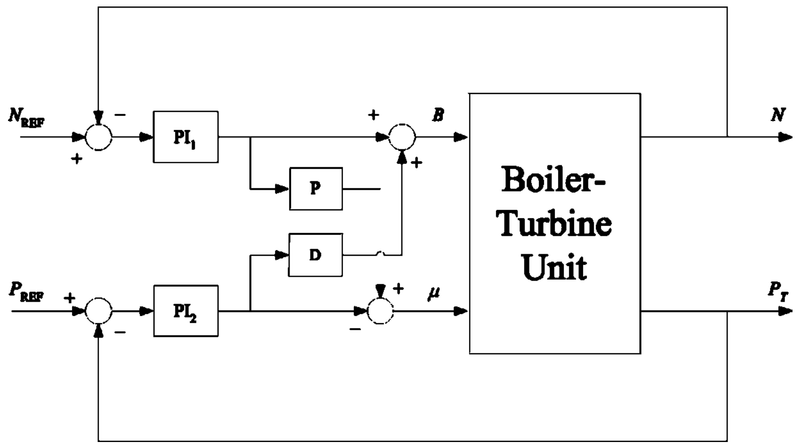

Fig. 4. Control structure of a boiler-turbine unit with a decoupler.

$$K=\begin{bmatrix} 1 & \dfrac{m_{12}}{m_{11}}s(T_1s+1) \\ \dfrac{m_{21}}{m_{22}}\dfrac{1}{(T_1s+1)(T_0s+1)} & -1 \end{bmatrix} \times\begin{bmatrix} K_{11} & 0 \\ 0 & K_{22} \end{bmatrix}. \tag{16}$$

However, the decoupler will be of high order and not easy to implement, so some simplifications should be made. Since the time constants T_1 and T_0 are usually larger than 10 s for a typical boiler-turbine unit, the dynamic effect of $1/(T_1s+1)(T_0s+1)$ can be ignored. The second-order derivative action of $s(1+T_1s)$ is not implementable so only the first-order derivative action is retained. The final PID controller for a boiler-turbine unit takes the following form:

$$K_d(s)=\begin{bmatrix} 1 & \dfrac{m_{12}}{m_{11}}s \\ \dfrac{m_{21}}{m_{22}} & -1 \end{bmatrix}\begin{bmatrix} \mathrm{PI}_1 & 0 \\ 0 & \mathrm{PI}_2 \end{bmatrix}, \tag{17}$$

where PI_1 and PI_2 are two PI controllers to be tuned to achieve the desired dynamic performance for each loop. The whole control structure is shown in Fig. 4.

An alternative option is to use a static decoupler:

$$K_s(s)=\begin{bmatrix} 1 & 0 \\ \dfrac{m_{21}}{m_{22}} & -1 \end{bmatrix}\begin{bmatrix} \mathrm{PI}_1 & 0 \\ 0 & \mathrm{PI}_2 \end{bmatrix}. \tag{18}$$

It is clear that if the two diagonal PI controllers in $K_s(s)$ and $K_d(s)$ are chosen as the same, then the two controllers will have the same tracking performance for the electric power N, but not for the throttle pressure P_T.

The decoupling effects of the decouplers obtained above are not quite satisfactory, as can be seen in the examples below. From the model in the previous section, the system is coupled only in Eq. (6). A new decoupler structure is described below.

Note that the inverse of Eq. (6) is

$$C(s)=\begin{bmatrix} T_bs+1 & C_Bs \\ \dfrac{1}{P_T} & -\dfrac{\mu}{P_T} \end{bmatrix}. \tag{19}$$

So a candidate for the decoupler of the whole unit can be chosen as

$$W(s)=\begin{bmatrix} \dfrac{T_1s+1}{k_1s} & 0 \\ 0 & \dfrac{1}{s} \end{bmatrix}C(s)\begin{bmatrix} \dfrac{1}{k_2} & 0 \\ 0 & 1 \end{bmatrix} =\begin{bmatrix} \dfrac{(T_1s+1)(T_bs+1)}{k_1k_2s} & \dfrac{(T_1s+1)C_B}{k_1} \\ \dfrac{1}{k_2P_Ts} & -\dfrac{\mu}{P_Ts} \end{bmatrix}. \tag{20}$$

Here integrators were added to achieve no offset set-point tracking.

Next, two single-loop controllers for the diagonal elements of the decoupled system need to be

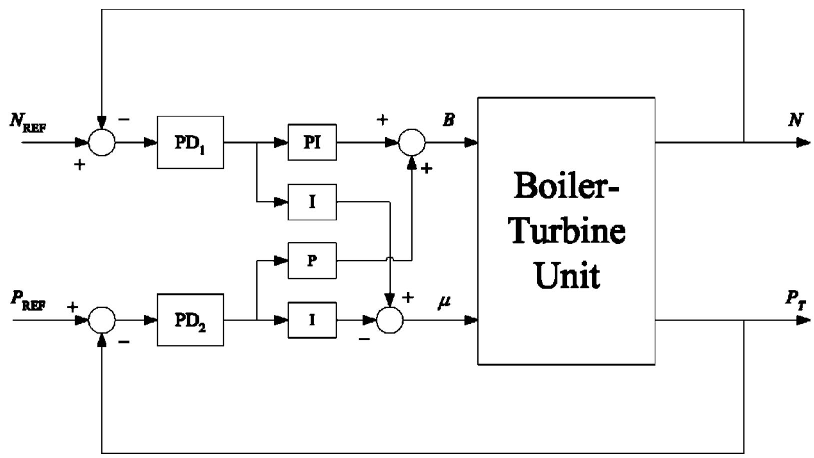

Fig. 5. Coordinated control structure of a boiler-turbine unit.

designed to improve the dynamic responses. They can be chosen as two PD controllers, since integral action is already added and PD controllers are known to be able to improve the dynamic performance. The final coordinated controller will be of the form

$$K(s)=\begin{bmatrix}\dfrac{(T_1s+1)(T_bs+1)}{k_1k_2s} & \dfrac{(T_1s+1)C_B}{k_1}\\ \dfrac{1}{k_2P_Ts} & -\dfrac{\mu}{P_Ts}\end{bmatrix}\times\begin{bmatrix}\mathrm{PD}_1 & 0\\ 0 & \mathrm{PD}_2\end{bmatrix}=\begin{bmatrix}\dfrac{(T_1s+1)(T_bs+1)}{m_{11}s} & \dfrac{(T_1s+1)m_{12}}{m_{11}m_{22}}\\ \dfrac{m_{21}}{m_{11}m_{22}s} & -\dfrac{1}{m_{22}s}\end{bmatrix}\times\begin{bmatrix}\mathrm{PD}_1 & 0\\ 0 & \mathrm{PD}_2\end{bmatrix}. \quad (21)$$

To ensure that each element of the final controller can be realized with a PID structure, the second-order polynomial $(T_1s+1)(T_bs+1)$ is approximated with a first-order one $(T_1+T_b)s+1$, which is possible as long as T_1T_b is small. Moreover, simulations show that the derivative action in the (1,2) block is very sensitive to process noise, so a static gain is used instead. The final coordinated PID controller for the boiler-turbine unit is

$$K_c(s)=\begin{bmatrix}\dfrac{(T_1+T_b)s+1}{m_{11}s} & \dfrac{m_{12}}{m_{11}m_{22}}\\ \dfrac{m_{21}}{m_{11}m_{22}s} & -\dfrac{1}{m_{22}s}\end{bmatrix}\times\begin{bmatrix}\mathrm{PD}_1 & 0\\ 0 & \mathrm{PD}_2\end{bmatrix}. \quad (22)$$

The whole control structure is shown in Fig. 5.

3.2. Tuning

Once the structure of the coordinated PID controller [Eqs. (18), (17), or (22)] is adopted, the parameters of the two single-loop PI or PD controllers need to be tuned to satisfy other performance of the system. Manual tuning of PI or PD controllers are well known. In this paper, robust tuning of PID controllers is used. The method is proposed in Ref. [22]. The basic idea is that PID controllers should be tuned to maximize the integral action under the constraint of a certain degree of robust stability, i.e.,

$$\max \underline{\sigma}(K_i) \quad (23)$$

under the constraint

$$\varepsilon_m := \mu_\Delta\left(\begin{bmatrix}I\\ K\end{bmatrix}(I+GK)^{-1}[I \quad G]\right) < \gamma_m, \quad (24)$$

where K_i is the integral gain of a PID controller, ε_m is the robustness measure, and γ_m is a given robust stability requirement. Extensive simulations show that ε_m should lie between 3 and 5 to have good tradeoff between time-domain performance and frequency-domain robustness [22]. In the examples below, iterative tuning of the PI or PD controllers is done: step responses are simulated to check if the parameters can achieve certain dynamic performance, and the robustness measure is computed to make sure that it is less than 4.

4. Simulation studies

Three examples are given in this section to illustrate the proposed PID structure and tuning method for boiler-turbine units.

Example 1: Consider a boiler-turbine unit with the following transfer function which was obtained by fitting the step response data:

$$G_1(s)=\begin{bmatrix} \frac{4.247(3.4s+1)}{(100s+1)(20s+1)(10s+1)} & \frac{3.224s(3.4s+1)}{(100s+1)(10s+1)} \\ \frac{0.224}{(100s+1)(20s+1)} & -\frac{0.19(20s+1)}{100s+1} \end{bmatrix}. \tag{25}$$

The model is in the standard form. For this model, we have

$$m_{11}=4.247,\quad m_{12}=3.224,\quad m_{21}=0.224,$$
$$m_{22}=0.19,$$
$$T_b=20,\quad T_1=20,\quad T_0=100,\quad T_2=10,$$
$$\alpha=0.34.$$

The coordinated controllers discussed in the previous section are

$$K_{d1}(s)=\begin{bmatrix} 1 & 0.7591s \\ 1.1789 & -1 \end{bmatrix} \times \begin{bmatrix} 0.72+\frac{0.005}{s} & 0 \\ 0 & 6+\frac{0.2}{s} \end{bmatrix}, \tag{26}$$

$$K_{s1}(s)=\begin{bmatrix} 1 & 0 \\ 1.1789 & -1 \end{bmatrix} \times \begin{bmatrix} 0.72+\frac{0.005}{s} & 0 \\ 0 & 6+\frac{0.2}{s} \end{bmatrix}, \tag{27}$$

$$K_{c1}(s)=\begin{bmatrix} 9.418+\frac{0.2355}{s} & 3.995 \\ \frac{0.2776}{s} & -\frac{5.263}{s} \end{bmatrix} \times \begin{bmatrix} 0.1(1+25s) & 0 \\ 0 & 0.1(1+25s) \end{bmatrix}, \tag{28}$$

where the diagonal PI and PD controllers are tuned such that the robustness measure in Eq. (24) is less than 4.

The step responses for the closed-loop system (step starts from $t=50$) and the controller outputs are shown in Fig. 6. It is clear that a step on the electric power output has little effect on the throttle pressure, and in this case both the governor valve and the firing rate respond to the electric power demand quickly, so the unit can follow the demand and the resulting pressure oscillation can be damped quickly. However, the pressure is mainly regulated by the governor valve, so it will affect the electric power output. We can see that K_{c1} has the best decoupling effect.

Example 2: Consider a 300-MW coal-fired once-through boiler-turbine unit. At full load, the following transfer function was obtained by fitting the step response data:

$$G_2(s)=\begin{bmatrix} \frac{2.069(311s+1)}{(149s+1)^2(22.4s+1)} & \frac{4.665s(99s+1)}{(582s^2+50s+1)(4.1s+1)} \\ \frac{0.124(205s+1)}{(128s+1)^2(11.7s+1)} & -\frac{0.139(2.8s+1)}{70s+1} \end{bmatrix}. \tag{29}$$

The model is not exactly in the form in Eq. (10). However, we can still get the following parameters from the model:

$$m_{11}=2.069,\quad m_{12}=4.665,\quad m_{21}=0.124,$$
$$m_{22}=0.139,$$

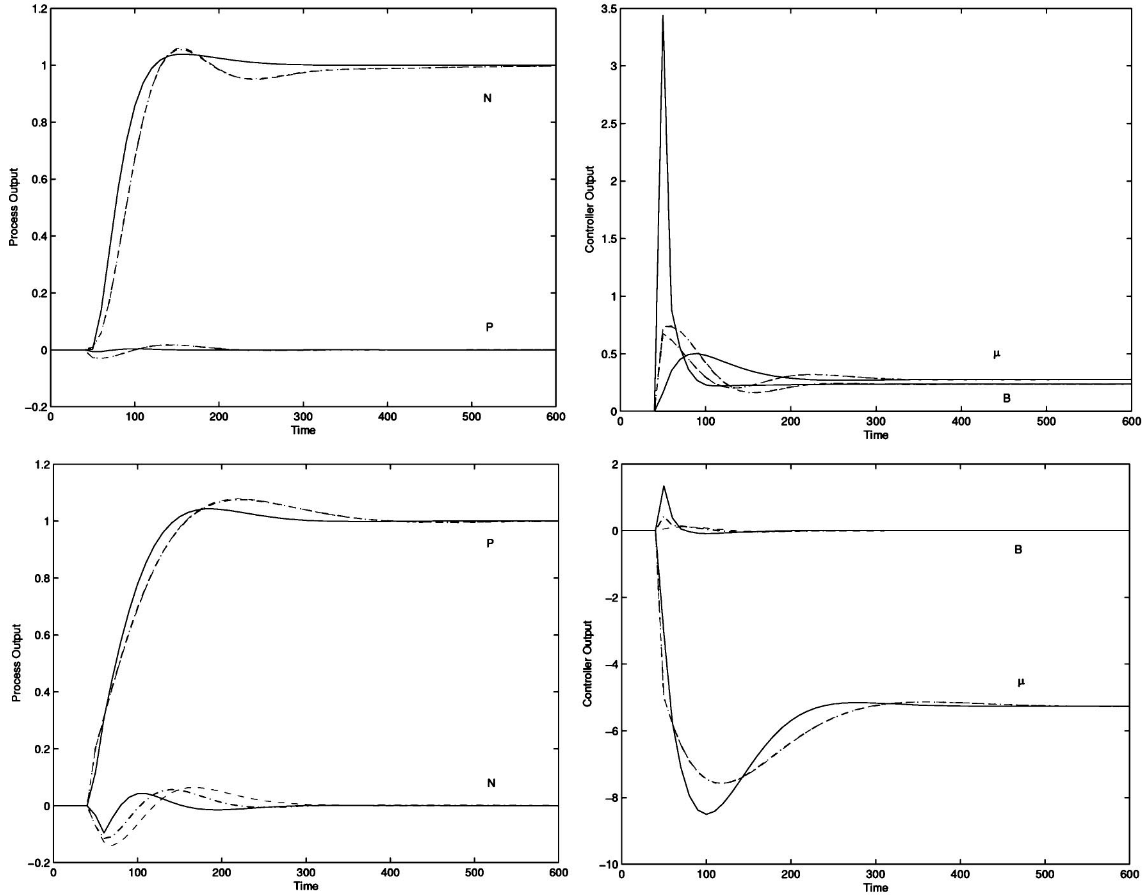

Fig. 6. Responses for example 1 (solid, K_{c1}; dash, K_{d1}; dash dotted, K_{s1}).

$$T_b = 2.8, \quad T_1 = 15.4689, \quad T_0 = 70.$$

The three coordinated controllers are

$$K_{d2}(s) = \begin{bmatrix} 1 & 2.2547s \\ 0.8921 & -1 \end{bmatrix} \times \begin{bmatrix} 0.427 + \dfrac{0.005}{s} & 0 \\ 0 & \dfrac{0.05}{s} \end{bmatrix}, \quad (30)$$

$$K_{s2}(s) = \begin{bmatrix} 1 & 0 \\ 0.8921 & -1 \end{bmatrix} \begin{bmatrix} 0.427 + \dfrac{0.005}{s} & 0 \\ 0 & \dfrac{0.05}{s} \end{bmatrix}, \quad (31)$$

$$K_{c2}(s) = \begin{bmatrix} 8.83 + \dfrac{0.4833}{s} & 16.22 \\ \dfrac{0.4312}{s} & -\dfrac{7.194}{s} \end{bmatrix} \times \begin{bmatrix} 0.08(1 + 72.8s) & 0 \\ 0 & 0.007 \end{bmatrix}. \quad (32)$$

The step responses for the closed-loop system (step starts from $t = 50$) and the controller outputs are shown in Fig. 7. Again the electricity demand can be followed quickly without affecting the throttle pressure. However, the pressure response is a bit slower since the coupling is more severe for this unit than the one in the previous example. Among the three controllers, K_{c2} has the best decoupling effect.

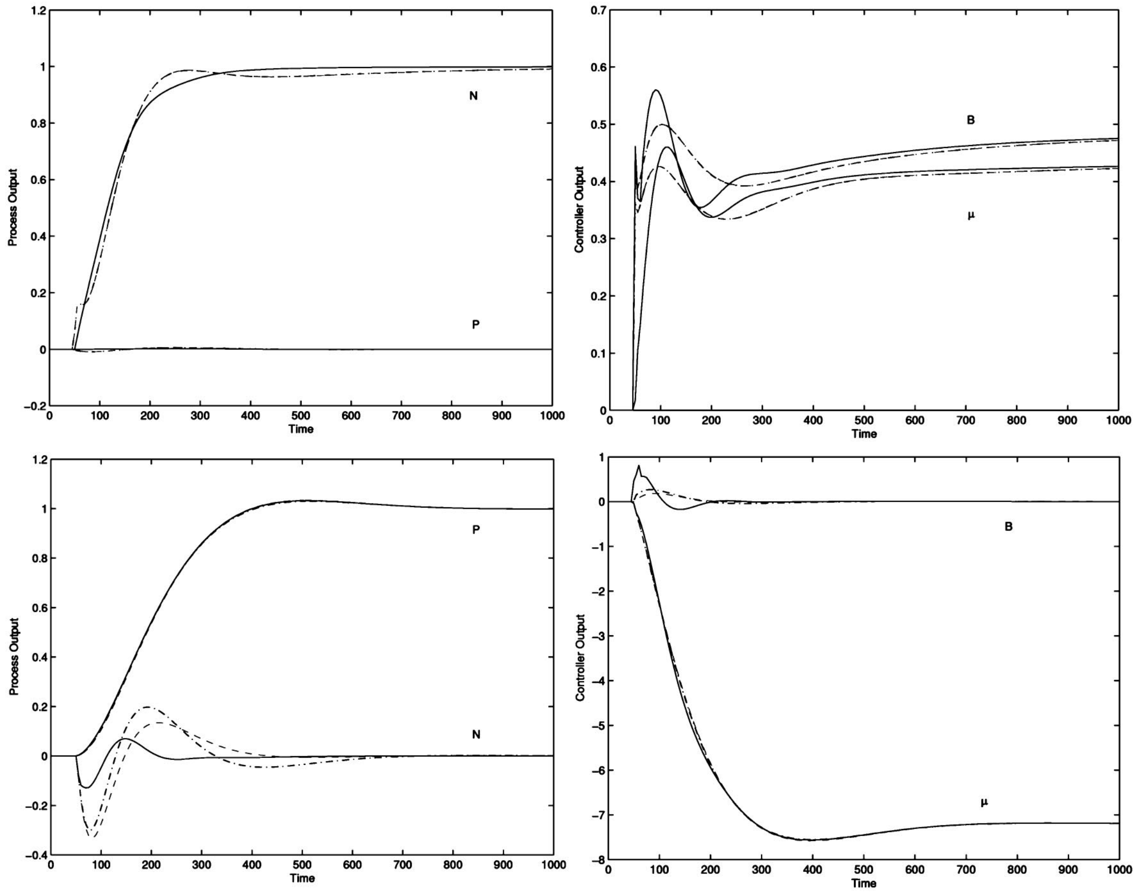

Fig. 7. Responses for example 2: full load (solid, K_{c2}; dash, K_{d2}; dash dotted, K_{s2}).

To test the robustness of the tuned controllers, the transfer function for the unit obtained at 70% load is obtained:

$$\begin{bmatrix} \dfrac{2.116(457s+1)}{(221s+1)^2(21.8s+1)} & \dfrac{1.483s(150s+1)}{(632s^2+40s+1)(2.7s+1)} \\ \dfrac{0.162(275s+1)}{(168s+1)^2(11.7s+1)} & -\dfrac{0.081(0.97s+1)}{97s+1} \end{bmatrix}. \tag{33}$$

At this load the step responses for the closed-loop system are shown in Fig. 8. Clearly the responses for the electric power degrade little for the three controllers.

Example 3: Consider a boiler-turbine unit [4] with the following transfer function:

$$G_3(s) = \begin{bmatrix} \dfrac{0.0595}{s^2+7.994s+0.0326}e^{-30s} & \dfrac{33333s+0.13}{2760s^2+424s+1} \\ \dfrac{0.6852}{s^2+7.994s+0.0326}e^{-30s} & -\left(\dfrac{0.0151}{5s+1}+\dfrac{80}{s^2+8.4s+0.049}\right) \end{bmatrix}. \tag{34}$$

The model appears quite different from our simple model; however, its step response is quite similar to the one shown in Fig. 2. So the simple model should capture the essential dynamics of the unit.

Since

$$m_{11}=1.8252, \quad m_{12}=33333,$$

$$m_{21}=21.0184, \quad m_{22}=1632.7,$$

and by curve fitting,

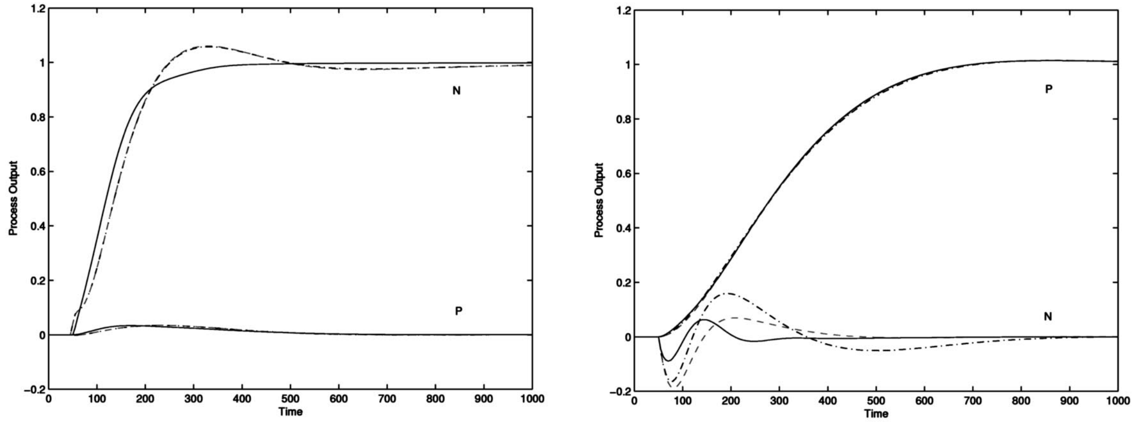

Fig. 8. Responses for example 2: 70% load (solid: K_{c2}; dash: K_{d2}; dash dotted: K_{s2}).

$$T_b \approx 0, \quad T_1 \approx 37.1, \quad T_0 \approx 171.3.$$

The coordinated controllers are

$$K_{d3}(s)=\begin{bmatrix} 1 & 1825s \\ 0.0129 & -1 \end{bmatrix} \times \begin{bmatrix} 0.08+\dfrac{0.004}{s} & 0 \\ 0 & 0.0042+\dfrac{0.00003}{s} \end{bmatrix}, \tag{35}$$

$$K_{s3}(s)=\begin{bmatrix} 1 & 0 \\ 0.0129 & -1 \end{bmatrix} \times \begin{bmatrix} 0.08+\dfrac{0.004}{s} & 0 \\ 0 & 0.0042+\dfrac{0.00003}{s} \end{bmatrix}, \tag{36}$$

$$K_{c3}(s)=\begin{bmatrix} 20.26+\dfrac{0.5479}{s} & 11.19 \\ \dfrac{0.007053}{s} & -\dfrac{0.0006125}{s} \end{bmatrix} \times \begin{bmatrix} 0.008(1+30s) & 0 \\ 0 & 0.02(1+10s) \end{bmatrix}. \tag{37}$$

K_{d3} has a very large derivative action on the (1,2) block, which makes the closed-loop system unstable, so the responses are not shown here. The step responses for the closed-loop system (step starts from $t=50$), and the controller outputs are shown in Fig. 9. The unit is slow in following the electricity demand due to a large time constant T_0 (over 2 min), however, the performance is still acceptable.

It should be noted that in practical implementation the derivative action must be followed by a bound and/or a rate limit or a filter to soften the unexpected sudden change on the control input, and a filter can also be used to filter out the noise on the measurement. In this case the filter dynamics should be considered when tuning the PD controllers. If the time constant of the filter is small compared with the dynamics of the unit, then the effect of the filter can be ignored. Fig. 10 shows the responses of the unit controlled by K_{c3} when there are white noises in the measurement of P_T and N. The power density of the noise is 0.1 for both measurements and the filter is chosen as $1/(10s+1)$ for both channels. It can be shown that except the derivation due to the noise the overall responses are similar to those without noise.

5. Conclusions

A simple model for a boiler-turbine unit was derived in the paper and a design and tuning method for the coordinated PID controller was proposed based on this model. Examples showed that the

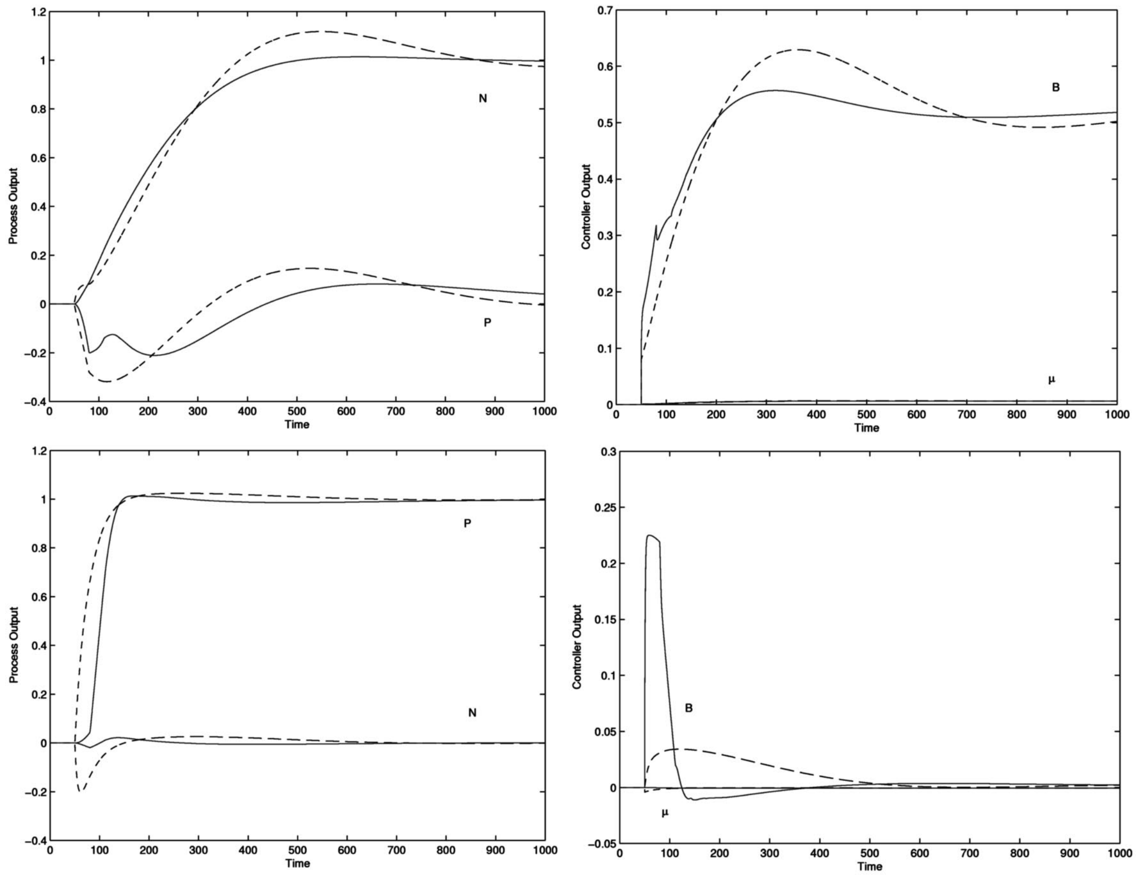

Fig. 9. Responses for example 3 (solid, K_{c3}; dash, K_{s3}).

method is easy to apply and can achieve acceptable performance. To achieve better performance, further researches should be directed to the following:

- *Modeling*. Though the model used in this paper is sufficient for PID design and tuning, however, a more sophistic model for a boiler-turbine unit can reveal more informa-

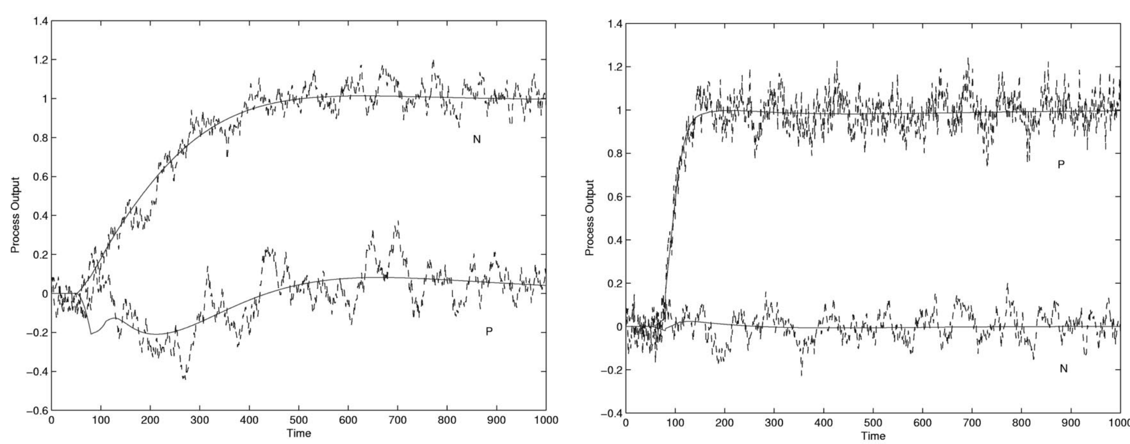

Fig. 10. Responses for example 3: Noisy measurements (solid, without noise; dash, with noise).

tion on the dynamics of a unit, and thus is more likely to have a better control.

- *Control structure*. Structures other than PID might be better suited for the control of a unit. For example, model predictive technique is one of the options that can be used to improve the overall performance.

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