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Linear Analysis and Control of a Boiler-Turbine Unit

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Abstract: Boiler-turbine units are multivariable nonlinear systems. The control of such systems is not easy considering the practical tuning, implementing and maintaining problems. In this paper, the design of a linear controller for the Dalate No.4 unit is reported. Based on a nonlinear model of the unit, we analyze the nonlinearity of the unit and propose to choose the appropriate operating points so that a linear controller can achieve wide-range performance. Simulation results and field tests show that the designed controller works well for the specific range of load variations.

Keywords: Boiler-turbine unit; linear control; wide-range performance; co-ordinated control

1. INTRODUCTION

A boiler-turbine unit is a configuration that is widely used in modern power plants. The configuration uses a single boiler to generate steam and directly feeds the steam to a single turbine to generate electricity.

The control system for a boiler-turbine unit usually needs to meet the following requirements:

- Megawatt output must be able to follow the load 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.

While each of the requirements listed above is important, the first two are critical for the safe and economic operation of a power plant. From the control point of view, the boiler-turbine unit can be modeled as a 2×2 system [Tan et al., 2004a]. The two inputs are boiler firing rate and throttle valve position. The two outputs are megawatt output and throttle pressure.

The control of boiler-turbine units has been widely studied in the literature using various control techniques, e.g., robust control [Kwon et al., 1989, Hwang and Kim, 1995, Tan et al., 1999]; nonlinear control [Fang et al., 2004]; predictive control [Rossiter et al., 1991, Rovnak and Corlis, 1991, Prasad et al., 1998, Poncia and Bittanti, 2001, Peng et al., 2001]; and intelligent control [Dimeo and Lee, 1995, Abdennour and Lee, 1996, Alturki and Abdennour, 1999, Abdennour, 2000, Moon and Lee, 2003]. While these methods are effective, the following problems makes the advanced control methods listed above seldom applied in practice:

(1) The design methods are not generic in that for each unit a model should be identified and a controller should be designed. This process is called ‘controller design’. However, for control engineers, ‘controller tuning’ is probably more preferred. For example, it is well-known that many industrial processes can be modeled as first-order-plus-deadtime (FOPDT) models and thus PID controllers can be tuned for such processes to achieve satisfactory performance. Though boiler-turbine units are different, they show similar dynamics and thus a similar control structure can be used for a class of boiler-turbine units. In other words, controllers can be tuned instead of designed for a class of units.
(2) The control algorithms are not easy to implement and maintain. It is well-known that the controllers designed by advanced control methods are generally complex. Though distributed control systems (DCS) are widely used in the power generation units, high-order controllers are still not easy to implement. Moreover, the anti-windup techniques for high-order controllers are far more difficult for PID controllers [Goodwin et al., 2000]. Finally, the control engineers are not familiar with the structure of the complex controllers and thus the cost of maintenance is high.
(3) The performance of a single controller cannot be guaranteed for wide-range load variations. Nowadays, the units at the power plants need to follow the demand by the dispatch. As the capacity of boilers increases, the nonlinearity of the boiler-turbine units increases, which makes the linearized models at the

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operating points vary, and a single (possibly robust) linear controller may not meet the control objectives for the desired operating range.

For the first two problems, Tan et al. [2004a] proposed a tuning method for coal-fired boiler-turbine units. The control structure is of PID type thus easy to implement, and the tuning procedure is not difficult. However, the controller is tuned at the nominal operating point, thus may not achieve the desired performance for wide-range load variations.

Gain scheduling control [Chen and Shamma, 2004] may solve the third problem, however gain scheduling needs to have a detailed nonlinear model or complete knowledge of the operating points, which is not practical. Moreover, the cost of implementing a gain scheduling controller is high.

Another possible method to solve the third problem is to use multi-model control [Tan et al., 2004b]. The method divides the operating range into several 'linear' range, and designs linear controllers at each local operating point, and combines them into a multi-model controller. The idea is simple and the simulation results shown that the method is effective.

A still simpler method is to 'avoid' the nonlinearity of a unit by carefully choosing the operating points [Tan et al., 2005]. The idea is that some of the operating points are seldom met in practice so even if the linear controller may not work well under such operating points it can still get good global performance as long as the controller does not enter such an operating range.

In this paper, we will report our practice on the design of a linear controller for the No.4 unit located at Dalate Power Plant in Inner Mongolia, China. A nonlinear model is derived first and the nonlinearity of the model is analyzed. It is noted that by carefully choosing the operating points a linear controller can achieve wide-range performance. Simulation results and field tests show that the designed controller works well for a reasonable range of operating points.

2. NONLINEAR BOILER-TURBINE MODEL

Detailed models on boiler-turbine units are rare in the open literature, but some simplified models can be found, e.g., Astrom and Eklund [1972]. The well-known control-oriented model of a 160MW boiler-turbine unit is reported in Bell and Astrom [1987], which has been studied for almost two decades. A simple nonlinear model for low and medium size boilers is reported in Cheres [1990], and it was shown that the model captures the essential nonlinearities that exist in a unit. However, this model is not verified for units with large size boilers. Nowadays as the size of boilers becomes larger and larger, new models are needed.

In Tian et al. [2004] a nonlinear unit model is proposed. The model is built for subcritical units (throttle pressure between 15.7 MPa and 19.6 MPa, main steam temperature between 535°C and 565°C) with pulverized-coal-fired, naturally-circulated drum boilers. These units are widely used in China. The nonlinear model takes the following form:

- Mill dynamics:

$$K_f \frac{dD_Q}{dt} = -D_Q + e^{-\tau s} B$$

- Energy balance for boiler:

$$C_B \frac{dP_D}{dt} = -K_3 P_T \mu + K_1 D_Q$$

- Energy balance for turbine:

$$K_t \frac{dN}{dt} = -N + K_3 P_T \mu$$

- Pressure drop between drum and throttle pressure:

$$P_T = P_D - K_2 (K_1 D_Q)^{1.5}$$

The model contains 5 variables (B, μ, P_T, N, and P_D) and 7 parameters (K_1, K_2, K_3, τ, K_f, C_B, and K_t). All the symbols are described in Table 1.

Table 1. Nomenclature

Parameters	Description
B	Boiler firing rate (t/h)
μ	Throttle valve position (%)
N	Megawatt output (MW)
P_T	Throttle pressure (MPa)
P_D	Drum pressure (MPa)
C_B	Boiler storage constant
K_1	Constant related to boiler firing rate and megawatt output
K_2	Superheater friction drop coefficient
K_3	Constant related to throttle valve and megawatt output
K_t	Time constant of the turbine (s)
K_f	Time constant of the mill (s)
τ	Pure time delay of the mill (s)

For the No.4 unit at Dalate Power Plant, the parameters are identified as:

$$\begin{aligned} K_1 = 6.313, K_2 = 0.000138, K_3 = 0.2334, P_D = 18.59, \\ K_t = 16, C_B = 2100, \tau = 60, K_f = 145 \end{aligned} \tag{1}$$

Five typical operating points are chosen to verify the accuracy of the statics of the model, the data are shown in Table 2. It is shown that the static errors are less than ±2%.

To verify the accuracy of the dynamics of the model, at operating point where the megawatt output N is 246MW and the throttle pressure P_T is 15.8MPa, decrease the fuel command B from 39t/h to 36.35t/h, the outputs of the model and the measured outputs of the unit are shown in Figure 1. At the specific operating point, the error between the model and the unit is small. Similar tests are done for other operating points. For brevity the figures are omitted.

3. NONLINEARITY ANALYSIS

It is generally accepted that a boiler-turbine unit is a highly nonlinear and strongly coupled complex system. However, there is no definite quantification of the complexity of a unit. For example, how nonlinear is it? can a *linear* controller be used to cover the whole operating range? These are fundamental issues in the control system design for a boiler-turbine unit.

One way to approach this problem is to study the nonlinearity of a unit. The nonlinearity measure attracted

Table 2. Verification of the statics of the model

Operating Points		Outputs of the unit		Outputs of the model		error	
B(t/h)	μ(%)	N(MW)	P_T(MPa)	N'(MW)	P'_T(MPa)	$(N'-N)/N$	$(P'_T-P_T)/P_T$
52.27	79.60	328.86	17.61	329.98	17.76	0.3	0.9
40.26	70.66	249.16	15.29	250.01	15.16	0.3	-0.9
28.40	56.29	168.52	12.38	165	12.56	-2.0	1.5
34.00	66.58	227.3	14.69	229.73	14.81	1.1	0.8
46.96	75.37	295.56	16.79	296.75	16.85	0.4	0.4

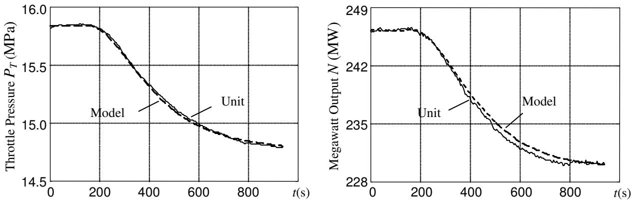

Fig. 1. Comparison of the outputs of the model and those of the unit

much attention in the past years, and several definitions and computation methods were proposed, see the survey paper [Nikolaou and Misra, 2003] and the references cited. Roughly speaking, a nonlinearity measure can be regarded as the 'distance' between a nonlinear system and a class of feasible linear systems [Nikolaou, 1993, Nikolaou and Misra, 2003]. A larger distance means that the system is 'more' nonlinear; and in this case, a linear control may not achieve good global performance.

In this section we will adopt the 'distance' measure proposed in Tan et al. [2005] to analyze the nonlinearity of the Dalate No.4 unit, and select appropriate operating points for wide-range operation accordingly.

Given a nonlinear system G, the distance measure proposed in Tan et al. [2005] is given by

$$v_g := \sup_{r_0} \delta(L_{r_0}(G), L), \qquad (2)$$

where $L_{r_0}(G)$ is the linearization of the nonlinear system G at operating point r_0, and $\delta(\cdot,\cdot)$ denotes the gap between two linear systems, and takes value between 0 and 1.

Consider the following operating point:

$$N = 330, P_T = 17.76, B = 52.27, \mu = 79.61,$$
$$P_D = 18.59, D_Q = 52.27 \qquad (3)$$

Figure 2 shows the distance from the model at the nominal operating point to the models at other operating points (which are determined from the throttle pressure P_T and the megawatt output N). It can be seen that some distance is larger than 0.8, thus we see that the unit is strongly nonlinear, and one single linear controller may not achieve the desired performance in the whole operating range.

However, it does not mean that we have to use a nonlinear controller to achieve the desired performance [Tan et al., 2005]. In fact, the plot of v_g shows us how to 'avoid' the nonlinear dynamics due to the operating point change. From the plots, we see that the same amount of megawatt output can be obtained with different throttle pressures, and the distances from these operating points to the

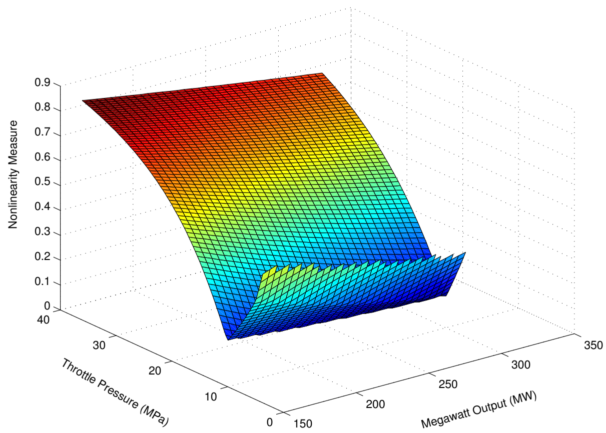

Fig. 2. Distances between the nominal model and the models at other operating points

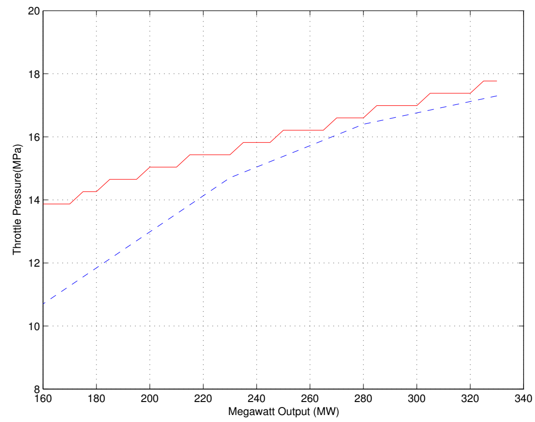

Fig. 3. Comparing operating points

nominal point are quite different, and have a minimal value. It shows that if we operate the unit in the load range from 150MW to 330MW then the unit will show little nonlinearity if the throttle pressure setpoints are varied accordingly. This kind of operation is called *sliding pressure operation* compared with *fixed pressure operation* where the throttle pressure is kept to a fixed value. The curve that lines up the corresponding points between the desired megawatt output and the desired throttle pressure is called a *sliding pressure curve*.

This sliding pressure curve is chosen according to the controllability of the unit. In practice, despite controllability, economics is the main impact of choosing operating points. We compare the chosen sliding pressure curve with that used in practical unit operation in Figure 3. They achieve good agreement from 230MW (70% load) to 330MW (100% load), which means that the chosen operating points are feasible from both the control point of view and the economic point of view.

4. CONTROLLER DESIGN

The previous section shows that a single linear controller might achieve the desired performance in the specific load range, so our next step is to design a linear controller at the nominal operating point. Many methods can be used in this step, however, considering the simplicity in structure and ease in tuning and maintenance, we will adopt the method proposed in Tan et al. [2004a]. The method is based on the following model which is linearized at a certain operating point of the unit:

$$\begin{bmatrix} \Delta N \\ \Delta P_T \end{bmatrix} = \begin{bmatrix} \dfrac{m_{11}(\alpha T_2 s+1)}{(T_1 s+1)(T_0 s+1)(T_2 s+1)} & \dfrac{m_{12}s(\alpha T_2 s+1)}{(T_0 s+1)(T_2 s+1)} \\ \dfrac{m_{21}}{(T_1 s+1)(T_0 s+1)} & -\dfrac{m_{22}(T_b s+1)}{T_0 s+1} \end{bmatrix} \times \begin{bmatrix} \Delta B \\ \Delta \mu \end{bmatrix} \tag{4}$$

where

$$m_{11} := k_1 k_2,\ m_{12} := \frac{P_T C_B k_1}{\mu}, \quad m_{21} := \frac{k_1}{\mu},\ m_{22} := \frac{P_T}{\mu} \tag{5}$$

and all the parameters can be easily found from the step response data at the operating point.

By Tan et al. [2004a], a candidate for the decoupler of the model (4) can be chosen as:

$$W(s) = \begin{bmatrix} \dfrac{T_1 s+1}{k_1 s} & 0 \\ 0 & \dfrac{1}{s} \end{bmatrix} \begin{bmatrix} T_b s+1 & C_B s \\ \dfrac{1}{P_T} & -\dfrac{\mu}{P_T} \end{bmatrix} \begin{bmatrix} \dfrac{1}{k_2} & 0 \\ 0 & 1 \end{bmatrix} = \begin{bmatrix} \dfrac{(T_1 s+1)(T_b s+1)}{k_1 k_2 s} & \dfrac{(T_1 s+1)C_B}{k_1} \\ \dfrac{1}{k_2 P_T s} & -\dfrac{\mu}{P_T s} \end{bmatrix} \tag{6}$$

and two PD controllers for the diagonal elements of the decoupled system are tuned to improve the dynamic responses. The final coordinated controller will be of the form:

$$K(s) = \begin{bmatrix} \dfrac{(T_1 s+1)(T_b s+1)}{k_1 k_2 s} & \dfrac{(T_1 s+1)C_B}{k_1} \\ \dfrac{1}{k_2 P_T s} & -\dfrac{\mu}{P_T s} \end{bmatrix} \begin{bmatrix} \mathrm{PD}_1 & 0 \\ 0 & \mathrm{PD}_2 \end{bmatrix} = \begin{bmatrix} \dfrac{(T_1+1)(T_b+1)}{m_{11}s} & \dfrac{(T_1 s+1)m_{12}}{m_{11}m_{22}} \\ \dfrac{m_{21}}{m_{11}m_{22}s} & -\dfrac{1}{m_{22}s} \end{bmatrix} \begin{bmatrix} \mathrm{PD}_1 & 0 \\ 0 & \mathrm{PD}_2 \end{bmatrix} \tag{7}$$

To ensure that each elements of the final controller can be realized with a PID structure, the second-order polynomial $(T_1 s+1)(T_b s+1)$ is approximated with a first-order one $(T_1+T_b)s+1$, which is possible as long as $T_1 T_b$ 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

$$K_c(s) = \begin{bmatrix} \dfrac{(T_1+T_b)s+1}{m_{11}s} & \dfrac{m_{12}}{m_{11}m_{22}} \\ \dfrac{m_{21}}{m_{11}m_{22}s} & -\dfrac{1}{m_{22}s} \end{bmatrix} \begin{bmatrix} \mathrm{PD}_1 & 0 \\ 0 & \mathrm{PD}_2 \end{bmatrix} \tag{8}$$

We note that though the model adopted in this paper is not the same as that used in Tan et al. [2004a], however, the step responses of the model near an operating point takes the typical form as shown in Tan et al. [2004a], so we can in fact tune the PID controller for the Dalate No.4 unit.

For the No.4 unit at Dalate Power Plant, at the nominal operating point we have

$$m_{11} = 0.1699, m_{12} = 0.1116, m_{21} = 6.3130, m_{22} = 234.30, \quad T_b \approx 0, T_1 = 144.57, T_0 = 56.52 \tag{9}$$

The final controller is tuned as:

$$K_c(s) = \begin{bmatrix} 332.6 & 22.9 + 0.1584/s \\ -8.963/s & 0.2412/s \end{bmatrix} \times \begin{bmatrix} 0.002 + 0.1s & 0 \\ 0 & 0.005 \end{bmatrix} \tag{10}$$

5. SIMULATION AND FIELD TEST RESULTS

To test the performance of the control system, simulations are done by connecting the linear controller (10) with the nonlinear model. The load variation range is from 50% to 100% of the maximum load (in practice, a co-ordinated control system is only required to work within the range from 70% to 100% of the maximum load). The load variation rate is 9MW/min, and the throttle pressure setpoint is changed according to the chosen sliding pressure curve, with rate less than 0.36MPa/min. The boiler firing rate and the throttle valve opening are limited by

$$0 < B < 100(t/h), 0 < \mu < 100(\%) \quad |dB/dt| < 0.55/s, |d\mu/dt| < 0.5/s \tag{11}$$

The simulation results are shown in Figure 4, where the dotted lines are the setpoints and the solid are the outputs of the unit. It can be seen that the controller can follow the load demand in a wide range.

The controller was implemented in the No.4 unit at Dalate Power Plant and field tests were done to test the performance of the controller. Figure 5 shows the main parameters of the unit under fixed pressure operation mode. The load demand was decreased from 310.0MW to 288.2MW with a rate of 8MW/min, and after the system becomes steady, increased from 288.2MW to 315.0MW with the same rate. The throttle pressure setpoint was fixed at 17.25MPa. (Note: Because several curves are put together, the vertical axis is not labeled. Instead, the upper bound and the lower bound for each parameter are denoted beside the parameter with 'H' representing the upper bound and 'L' for the lower bound. It is the same with the figures below)

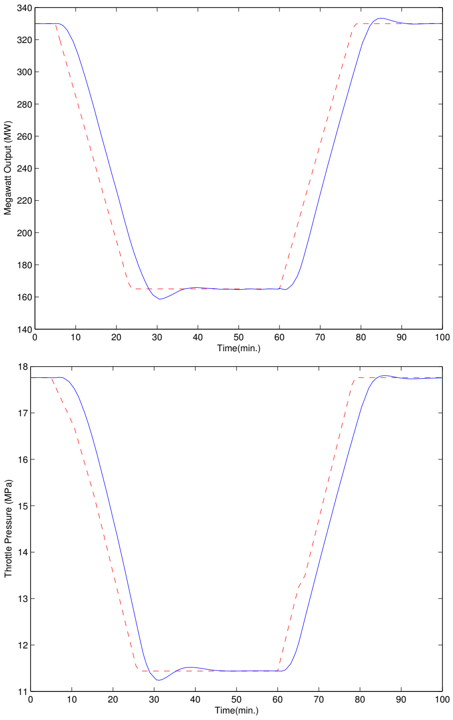

Fig. 4. Simulation results of wide-range operation

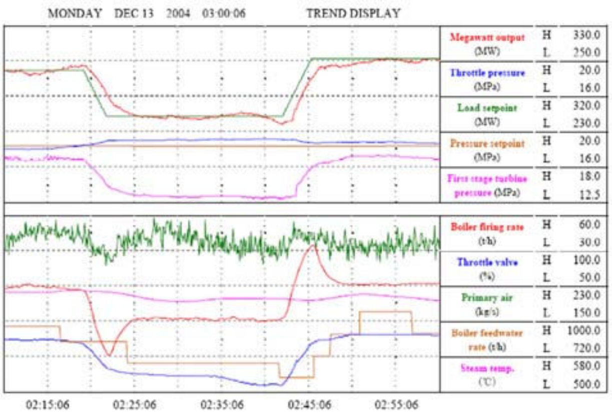

Fig. 5. Main parameters under fixed pressure operation mode

Figure 6 shows the main parameters of the unit under sliding pressure operation mode. The load demand was decreased from 308.2MW to 285.1MW with a rate of 3MW/min, and after the system becomes steady, increased from 285.1MW to 315.0MW with a rate of 6.9MW/min. The throttle pressure was varied according to the selected sliding pressure curve. The megawatt output follows the load demand swiftly and accurately, and other parameters are steady.

The above tests shows that the linear controller works well when the load demand is high (90% of the maximum load) in both fixed pressure operation mode and sliding pressure operation mode. To show that the controller can work in a wide range of load variations, we need to test the performance when the load demand is low. This is especially important, since the linear controller is tuned at high load. Figure 7 shows the main parameters of the unit

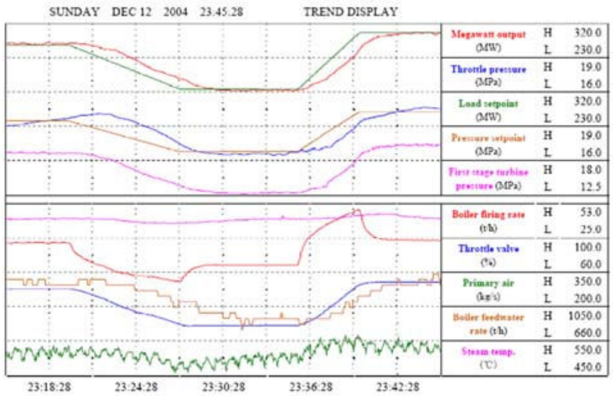

Fig. 6. Main parameters under sliding pressure operation mode

under the sliding pressure operation mode when the load demand is low. The load demand is first increased from 238.0MW to 268.0MW with a rate of 7.5MW/min, and after the system becomes stable, decreased from 268.0MW to 247.1MW and to 233.5MW continuously. The throttle pressure setpoints are changed according to the sliding pressure curve. It is shown that the controller also works well when the load demand is low (70% of the maximum load), thus the control system may work in a wide range of load variations.

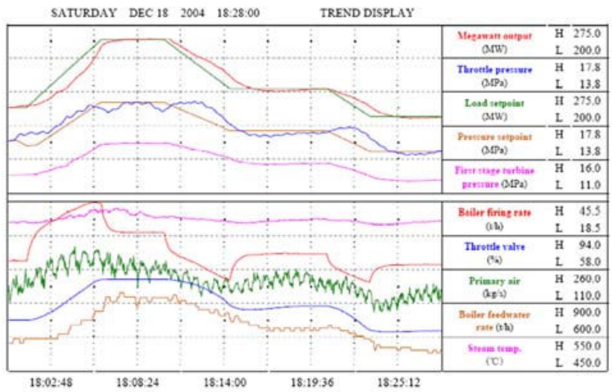

Fig. 7. Main parameters under sliding pressure operation mode: low load

Besides the tracking performance, the disturbance rejection ability of a control system is also very important. Figure 8 shows the main parameters of the unit under soot blowing. The soot blowing process can be regarded as a large disturbance affecting the boiler firing rate B. It is shown that the throttle pressure and the megawatt output change slightly despite the input disturbance.

From the field tests, we see that the designed linear controller has a fast load tracking performance and works fine under both the sliding pressure mode and the fixed pressure mode and may work in a reasonable range of load variations (70% to 90% of the maximum load). It also has a good disturbance rejection ability. The simulation results show that the controller can also work under the range of 50%-70% of the maximum load. However, due to the strict safety restriction in the power plant, we could not test the performance within this range of load in field.

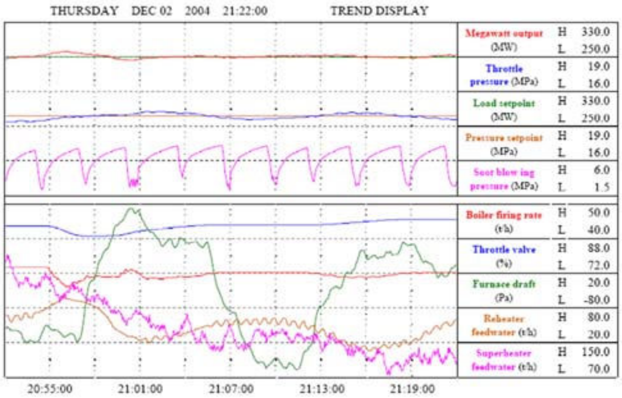

Fig. 8. Main parameters under soot blowing

6. CONCLUSION

Linear control of a boiler-turbine unit was discussed in this paper. The nonlinearity of the unit was analyzed based on the nonlinear model of the unit, and appropriate operating points were selected so that the linear controller could achieve wide-range performance. Simulation and experimental results at the No.4 Unit at the Dalate Power Plant show that the linear controller can achieve the desired performance under the specific range of load variations.

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