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DEB-oriented Modelling and Control of Coal-Fired Power Plant

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Abstract: Direct-energy-balance (DEB) based coordinated control strategy is critical in achieving good load following and steam pressure stability in modern power plant. Dealing with the adverse impact of perturbation in coal quality, however, is still a challenge to be resolved. In this paper, a 300MW drum-boiler unit model is first established and verified by experimental data, for the purpose of DEB and calorific value flexibility study. Unlike conventional measurement correction methods, the heat value variation is considered here as a part of the internal disturbance, which is estimated and compensated in an improved active disturbance rejection control (ADRC) structure. The proposed control strategy brings a novel design concept for a critical problem of heat variation in any coal-fired power plant. The simulation results demonstrate that the dynamic performance of main steam pressure can be significantly improved.

1. INTRODUCTION

The coal-fired power plant has achieved a rapid development in recent years and will undoubtedly be the main power generation unit for a long time. Increasingly strict demand on load and frequency regulations (Edlund et al., 2008), however, is put forward to ensure the safety of power grid. Generally, the main requirements for load control system are listed as follows.

- Electric power output can be adjusted rapidly as demand by automatic dispatch system (ADS). The regulation rate in China is usually 1.5%-2% full load per minute.
- Main steam pressure can be maintained in a limited range (usually $\pm$ 0.4MPa) despite variations of the load.
- The output power and main steam pressure must be maintained well while the heat value of the coal varies.

In modern power plant, direct energy balance (DEB) based coordinated control strategy can fulfil the first two goals well. The boiler follows turbine (BFT) mode is usually adopted in DEB, in which the governor valve is responsible for tracking the grid demand rapidly. At the same time, some existing feedforward strategy in DEB can guarantee the throttle pressure within the bounds. So the urgent obstacle facing field engineers is the third requirement.

To reach this goal, we should first develop a simple and suitable model for DEB control research. The classical model proposed by (Bell and Astrom, 1987) which representing a 160 MW oil fired power plant was widely accepted as a real plant and a number of control strategies were researched based on this model ((Lee et al., 2008), (Lu et al., 2010) and (Wu et al., 2013)). However, this model differs from modern power plant in (1) the time delay neglected due to the fast oil pumping process while the coal conveying process needs to cost quite a long time; (2) the heat value of oil stayed relatively steady compared with raw coal; (3) the controlled pressure in oil fired plant is drum pressure while main steam pressure is more preferred in coal-fired plant. In addition, some parameters needed by DEB cannot be provided by Bell-Astrom model.

In conventional DEB structure, the solution to the variation of heat value is usually dependent on the measurement methods. The traditional way was to adopt the calorific value correcting method. The results, however, were not reliable due to the poor calculation accuracy. The modern on-line measurement is under research but the associated high cost limited its applications.

Active disturbance rejection control (ADRC) strategy (Han, 2009) received more attention recently (Gao 2013), which was applied successfully in the Parker Parflex hose extrusion facility (Zheng and Gao, 2012) and ALSTOM gasifier (Huang et al., 2013). These applications demonstrate its enormous potential for process control.

In this study, the perturbation of heat value and other unknown dynamics were regarded as the disturbance which is then estimated using the improved extended state observer (ESO) and compensated in real time. Simulation results demonstrate the superiority of the proposed scheme over the conventional method.

2. A SIMPLIFIED PLANT MODEL FOR DEB DESIGN

2.1 Plant Description

The unit of a drum-boiler 300MW power plant in Guangdong Province, China, is selected for the modelling study. In this section, we build a nonlinear model based on the first law of thermodynamics. A schematic view of a coal-fired power plant is shown in Fig.1. The number of parameters was reduced significantly through some assumptions and simplifications so that the parameters can be calculated simply.

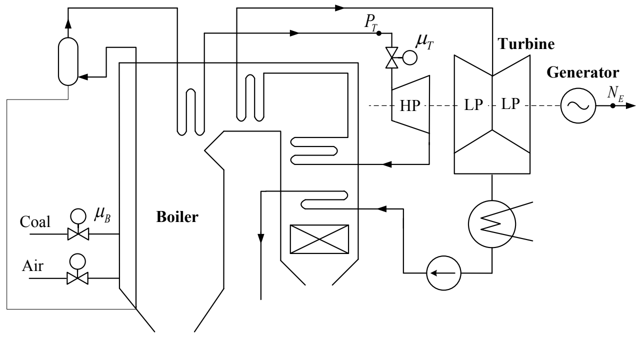

Fig.1. A schematic view of a coal-fired power plant

The system dynamics can be roughly separated into two different time scales. The dynamics governing the coal and steam flows are relatively fast, whereas dynamics from heat transfers are much slower. In this paper, we ignored the slow processes in steam and water circuit as no temperature or enthalpy parameters are needed in the DEB control system. However, the slow characteristics in the combustion process still remained to reflect the fundamental feature of coal-steam process, which is also the original difficulty for DEB control and coal quality flexibility.

2.2 Coal-Steam Transformation Channel

The performance of coal-steam channel has a significant impact on the overall dynamic response of coal-fired power plants. Thus, the channel model should reflect the specific features of large inertia and time-delay in the coal pulverizing, combustion and heat release processes.

The mass balance for the pulverizer is

$$q_p^i - q_p^o = \frac{dM}{dt} \tag{1}$$

where q_p^i and q_p^o represent the mass flow of coal moving into and out of the pulverizer, respectively. M is the coal storage mass.

With operating experience, the mass flow of pulverized coal can be considered proportional to the mill load.

$$q_p^o = \frac{1}{c_0} M \tag{2}$$

where c_0 is a time constant which can be identified.

In principal, the pure delay resulted from the conveyor belt, τ_1, and primary air pipe, τ_2, which can be expressed as:

$$q_p^i = u_B e^{-\tau_1 s} \tag{3}$$

$$q_f = q_p^o e^{-\tau_2 s} \tag{4}$$

where u_B is the boiler demand, q_f is the mass flow of coal blowing into the burner.

To write energy balance equations, let V_f denotes the volume of the furnace, Q_r denotes the radiant energy released, Q_{net}^{ad} the lower heating value (LHV) and η the combustion efficiency. Furthermore, let subscripts a, s, g, f and w refer to the inlet air, steam, flue gas, furnace and water wall, respectively. D, c, m, and T is the mass flow, specific heat capacity, mass and Kelvin temperature.

The energy balance for the furnace is

$$V_f c_g \frac{dT_g}{dt} = q_f Q_{net}^{ad} \eta + D_a c_a T_a - D_g c_g T_g - Q_r \tag{5}$$

The energy balance equation for water-wall is

$$m_w c_w \frac{dT_w}{dt} = Q_r - Q \tag{6}$$

where Q is the heat transferred from water-wall to the flowing steam inside.

In addition, two heat transfer equations should be included to ensure the closure of the equations.

$$Q_r = K_1 (T_g^4 - T_w^4) \tag{7}$$

$$Q = K_2 A_w (T_w - T_s) \tag{8}$$

where K_1 and K_2 is the radiation and convection heat transfer coefficient, respectively.

Considering the water and steam in the drum and risers are both in the saturation condition, the heat-mass transformation equation lists below.

$$Q \approx D_s r_s \tag{9}$$

where r_s is the latent heat of vaporization of water. Finally, combining the Eq. (1)-(9), we can calculate the real-time steam production rate, D_s, according to boiler demand, B.

2.3 Steam-Pressure Transformation Channel

In the proposed model, the boiler steam system was divided into two parts as depicted in Fig. 1, which could be considered as two single-phase pipes. The conservation law of mass is listed below.

$$\frac{\partial D}{\partial t} + A \frac{\partial \rho}{\partial x} = 0 \tag{10}$$

Here A is the cross-sectional area, and both of the mass flow in the pipe, D, and steam density, ρ, are the function of x-axis position and time. In addition, the resistance formula for the pipe is

$$\frac{\partial \rho}{\partial x} + p_d = 0 \tag{11}$$

The friction loss per unit length p_d can be calculated based on Darcy-Weisbach equation.

$$p_d = \xi_d \frac{D^2}{\rho} \tag{12}$$

So far, the closure of the equations in the steam-pressure channel was also achieved.

2.4 Pressure-Power Transformation Channel

The pressure of the governing stage, p_1, can be expressed as:

$$p_1 = k_1 \mu_T p_T \tag{13}$$

where p_T is the main steam pressure, μ_T is the position of turbine valve actuators, and k_1 is the proportional coefficient.

Let superscript '$\sim$' denote rated operating condition, T_{th} refer to the steam temperature in the throttle. Assuming the exhaust pressure, p_n, is much less than p_1, Flugel-based model of inlet steam mass flow, D_T, can be simplified as

$$D_T = \sqrt{\frac{p_1^2 - p_n^2}{\tilde{p}_1^2 - \tilde{p}_n^2}} \sqrt{\frac{T_{th}}{\tilde{T}_{th}}} \tilde{D}_T = kp_1 \sqrt{1 - (\frac{p_1}{p_n})^2} \approx k_2 p_1 \tag{14}$$

where, k_2 is a static gain parameter varying with operating point.

Let subscript r and e denote reheater and economizer, h refer to the enthalpy, φ turbine efficiency, H_c denote the heat released in the condenser. Based on energy conservation, the output power can be expressed as

$$Ne = \varphi(D_T(h_1 - h_e) - H_c + D_r(h_r^o - h_r^i)) \tag{15}$$

Actually, the heat transfer amount in the reheater and condenser is usually proportional to the current load power according to engineering experiences. So, a good approximation of Ne is

$$Ne = \varphi(D_T(h_T - h_e)) + \beta Ne \tag{16}$$

Simplify the expression by combining like terms:

$$Ne = k_3 D_T \tag{17}$$

where, $k_3 = \frac{\varphi(h_T - h_e)}{1-\beta}$, which is also a static parameter depending on operating point.

For simplicity, the dynamic characteristics of turbine can be described by three equivalent first order links, considering the storage capacity or inertia of nozzle chamber and reheater (de Mello (1991)). Thus, the Eq. (15), (16) can be respectively rewritten as:

$$p_1 + c_1 \dot{p}_1 = k_1 \mu_T p_T \tag{18}$$

$$D_T + c_2 \dot{D}_T = k_2 p_1 \tag{19}$$

With identified time-varying parameters k_i, the critical operation parameters of turbine can be easily calculated from the previous channel and input variables.

3. THE STATE SPACE MODEL

3.1 Further simplifications

The model above can capture the gross behaviour of the plant well. The model does, however, have two serious deficiencies, which make it not practical for system synthesis. The first is there are too many intermediate variables existing in the equations, which will increase the accumulative errors; the other is the large computational cost due to strong nonlinearity and partial differential equations.

Additional simplifications based on linearization and lumped parameter method (LPM) can be made if we are only interested in the coordinated control level such as DEB. Many unimportant intermediate variables are summarized as a time-varying item which can be identified through experimental data.

Due to the page limit, this part is omitted, which is considered to be presented in an extended version.

3.2 The affine nonlinear state-space model

There are many alternative choices of state variables to obtain a control-oriented model. In this section, we will build a model with six states, $x = \left[q_f \ D_s \ p_b \ p_T \ p_1 \ D_T \right]^T$, which gives insight into the key physical mechanisms that affect the dynamic behaviour of the power plant remarkably. By defining control input $u = \left[u_B(t-\tau) \quad \mu_T \right]^T$ and system output $y = \left[p_T \ Ne \right]^T$, the affine nonlinear state-space model can be expressed as

$$\dot{x} = f(x) + \mathrm{g}(x)u \tag{20}$$

$$y = h(x) \tag{21}$$

where

$$f(x)=\begin{bmatrix} -\frac{1}{c_0}x_1 \\ \frac{1.37}{c_5}x_3^{0.6}(x_1-x_2) \\ \frac{1}{c_6}(x_2-19.4x_4^{1.3}\sqrt{x_3-x_4}) \\ \frac{1}{c_7}(19.4x_4^{1.3}\sqrt{x_3-x_4}-x_6) \\ -\frac{1}{c_1}x_5 \\ \frac{1}{c_2}(74.86x_5-x_6) \end{bmatrix} \quad g(x)=\begin{bmatrix} \frac{1}{c_0} & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0.0084x_4 \\ 0 & 0 \\ 0 & 0 \end{bmatrix}$$

$$h(x)=\begin{bmatrix} x_4 & 0.31x_6 \end{bmatrix}^T$$

3.3 Parameter identification and model validation

In the state-space model proposed above, there are six inertia constants c_0, c_1, c_2, c_5, c_6, c_7 and time-delay τ to be identified. In this section, genetic algorithm (GA) was adopted to optimize these parameters by minimizing the error between the model output and experimental data. Another group of test data was applied to validate the high precision of the identified model, as shown in Fig. 2.

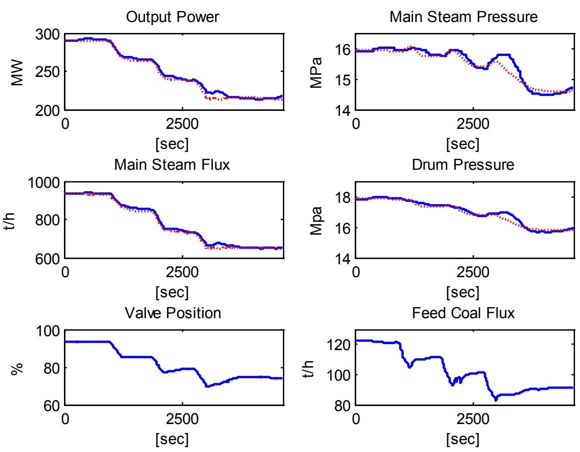

Fig. 2. Model validation (—— Test data; ---- Model output)

4. A MODIFIED DEB STATEGY BASED ON ADRC

4.1 Analysis of the conventional DEB structure

Enhanced stability requirements for coal-fired power plant in the past thirty years have led to a rapid progress in coordinated control systems, wherein the direct energy balance control strategy was most widely used. It balances the heat released from the boiler, ($Q_m = p_1 + C_b \frac{dp_b}{dt}$), with the energy demanded by the turbine-generator, ($Q_r = p_1 \frac{p_r}{p_T}$).

Here, p_r is the reference of main steam pressure, and C_b is the thermal storage coefficient. Under this structure, the dynamic regulation of main steam pressure mainly relies on the feedforward action while PI controller is responsible for eliminating static error. As long time-delay exists in the pressure loop, a weak PI controller is preferred to avoid frequent adjustment which is harmful for coal mill and consumes extra energy. However, the weak PI may fail in disturbance rejection since the unknown disturbances can only be rejected through feedback loop. So, the primary drawback of the traditional DEB strategy is the slow characteristics in disturbance rejection.

4.2 Active disturbance rejection control

4.2.1 Basic principles

Active disturbance rejection controller (ADRC) was originally proposed by (Han, 1999), aiming to design a novel control strategy which was independent on accurate plant model. The central idea of ADRC was to treat the nonlinear, coupling and disturbances in the plant as an extended state, which would be actively estimated and compensated for in real time.

Assume the process G_p can be modelled as a general first order plant:

$$\dot{y} = g(t, y, \ddot{y}, \cdots, \omega) + bu \tag{22}$$

where b is the plant parameter and g represents the high order, nonlinear, coupling, disturbances, etc. in the plant. Define $f = g + (b - b_0)u$ and then we can get

$$\dot{y} = f + b_0 u \tag{23}$$

Define $x_1 = y$ as a state variable and $x_2 = f$ as an extended state variable. On the premise that x_1 is measurable and f is differentiable, (23) can be written in the canonical state space form as:

$$\begin{cases} \dot{x}_1 = x_2 + b_0 u_1, \\ \dot{x}_2 = h. \end{cases} \tag{24}$$
$$y = x_1.$$

To estimate f, design an extended state observer (ESO) for plant expressed in (24) as follows:

$$\begin{cases} \dot{z}_1 = z_2 + \beta_1(x_1 - z_1) + b_0 u_1 \\ \dot{z}_2 = \beta_2(x_1 - z_1) \end{cases} \tag{25}$$

where β_1, β_2, b_0 are the observer parameters. When ESO is accurately tuned, z_1, z_2 will track y, f, respectively.

With the estimated extended state z_2, the control law is constructed as:

$$u = \frac{(u_0 - z_2)}{b_0} \tag{26}$$

Combining (23) with (26) to get a simplified plant as follows:

$$\dot{y} = f + b_0 u \approx z_2 + b_0 \frac{u_0 - z_2}{b_0} = u_0 \tag{27}$$

Till now, the extended state f named 'generalized disturbance' is compensated after being estimated by ESO.

Design a proportional controller for the reduced plant in (27) as follows:

$$u_0 = k_p(r - y) \tag{28}$$

where r is the reference input. Combine (27)-(28) to obtain the closed-loop dynamic equation:

$$\dot{y} + k_p y = k_p r \tag{29}$$

Conducting Laplace Transformation, we can get

$$G_{cl}(s) = \frac{k_p}{s + k_p} \tag{30}$$

Finally, the derivation process above can be illustrated as

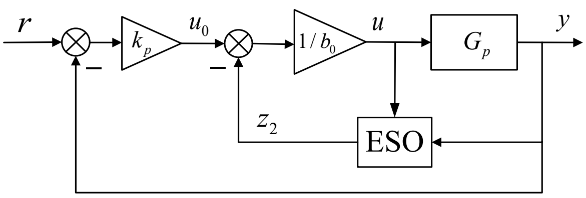

Fig.3. Diagram of ADRC

4.2.2 Tuning rules for ADRC parameters

To simplify the parameter tuning procedure, (Gao. 2003) proposed a practical method based on bandwidth-parameterization. In this approach, k_p, β_1, β_2 are certain functions of the controller or observer bandwidth as follows:

$$k_p = \omega_c, \beta_1 = 2\omega_o, \beta_2 = \omega_o^2 \tag{31}$$

Thus the ADRC parameters decreased to 3, which are ω_o, ω_c, b_0.

In order to achieve the desired control performance, the following experiences would be helpful in further tuning.

- The larger the ω_o is, the stronger tracking ability the ESO has. But a large ω_o results in the sensitivity to noises.
- A large ω_c or a small b_0 causes the strong control action, which leads to fast response but also overshoot and fluctuation at the same time.

Thus simply tuning of ω_o, ω_c, b_0 is required to obtain a satisfactory set of ADRC parameters.

4.3 Modified framework of DEB based on ADRC

In this section, a modified framework is proposed, which regards coal quality perturbation as internal disturbance. Based on an improved ESO with input-delay proposed by (Zhao and Gao, 2013), the internal disturbance occurring in the time-delay process can be captured synchronously. However, the estimated total disturbance $\hat{d}$, includes not only the internal disturbance, but also external disturbance and slight fluctuation, which would result in invalid control effort. As is known, the external disturbance is mainly from valve action, whose dynamic characteristic is much faster than boiler. Based on the different frequency band of internal and external disturbances, a strong filter after ESO is introduced here to separate the external disturbances and unavoidable fluctuation from the estimated total disturbance. The schematic of the framework is illustrated below.

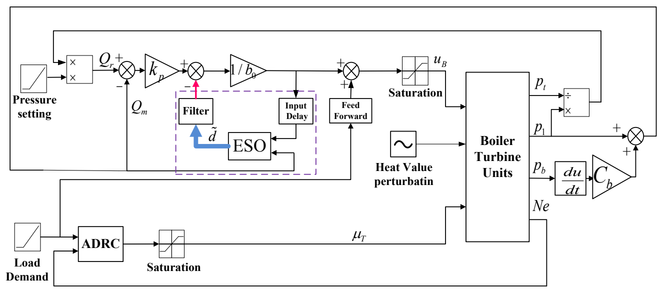

Fig.4. The schematic of modified DEB structure

A Butterworth filter was adopted in the structure, which was often referred as a maximally flat magnitude filter.

4.4 Simulation results

The input delay was assigned equal to the identified time delay in section 3.3. The controllers' parameters were set below.

Table 1(a) Controller parameters

Control Loop	Parameters of conventional DEB
Feedback Loop1	$P_1 = 2, I_1 = 0.02$
Feedback Loop2	$P_2 = 0.4, I_2 = 0.1$
Feed forward Loop	$Kp_{f1} = 0.423, Kd_{f1} = 100, T_1 = 0.855$

Table 1(b) Controller parameters

Control Loop	Parameters of modified DEB
Feedback Loop1	$\omega_{o1} = 3.0, \omega_{c1} = 2.24, b_{01} = 1.0, fc = 0.005$
Feedback Loop2	$\omega_{o2} = 1.0, \omega_{c2} = 1.0, b_{02} = 10.0$
Feed forward Loop	$Kp_{f2} = 0.62, Kd_{f2} = 80, T_2 = 0.855$
Cut-off frequency	$fc = 0.005$

During the simulation, the load demand is reduced by 40MW at t = 2000s with a speed of -6MW/min; After the output power and the main steam pressure reached stable, the heat value (k_4) reduced at t =7000s. The simulation results were shown in Fig. 5.

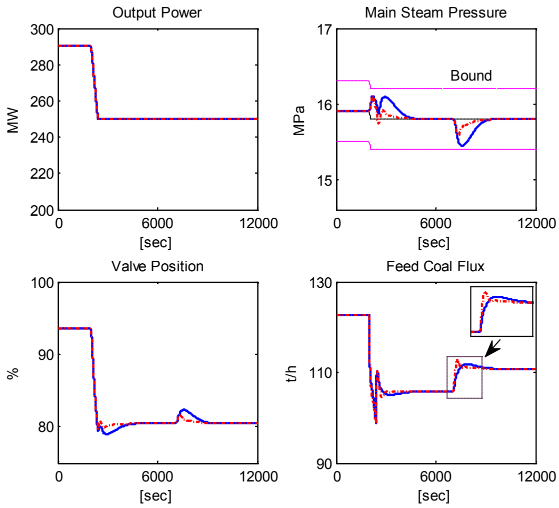

Fig. 5. The comparison of both structures
(—— Conventional; ---- Modified; ---- Reference)

The simulation results show the prefect the performance of both structures in load tracking. Less control effort, however, shows the advancement of ADRC controller. Main steam pressure during the load down process could be dragged back more quickly under the modified framework.

5. CONCLUSION

A state space model was built under the DEB framework for the purpose of control design. The comparison with experimental data demonstrates a high accuracy of this model. More importantly, the model adopts a novel conception of the problem where the heat value variation, a critical problem in coal-fired power plant, is treated as a part of disturbance, to be estimated and rejected. This leads to a more effective control strategy based on ADRC where the weakness of the existing measurement methods is overcome. In particular it is shown that the method proposed can be feasibly implemented with a small but key revision under the existing framework.

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