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# Optimization on Loading Up Control Method for Marine Supercharged Boiler Installation

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**Abstract.** With the intricate dynamic and thermodynamic coupling relationship among the equipment of supercharged boiler installation, its dynamic response is not the same as that of atmospheric pressure boiler in the process of loading up. Specific control method is required to keep the boiler loading up steadily and safely. On the basis of characteristics analysis on load-up process of supercharged boiler installation, the mechanism simulation model on a type of marine supercharged boiler was built, and its load-up control method was studied. In addition, the maximum uniform opening velocity of fuel flow rate control valve during loading up was obtained via simulation experiments. Besides, a control method for loading up supercharged boiler rapidly was put forward. The results can provide reference for the design of the actual monitoring system and the parameters setting of regulating valve.

**Keywords:** supercharged boiler installation, loading up, control method, coefficient of excess air, simulation.

## 1. Introduction

Because of the advantages of smaller volume and higher efficiency with normal pressure boiler, the supercharged boiler installation (SBI) is more and more widely used in the field of energy power [1]. Compared with normal pressure boiler, it has stronger coupling relationship among each equipment compositions. In order to ensure the safety, stability and economic operation of SBI, it is necessary to study the dynamic matching relationship between SBI, especially the matching relationship during the loading up of boiler [2]. Because of the mechanical inertia of rotor of turbo charging uint during the loading up of SBI, it is difficult to accurately grasp which control unit to regulate and with what speed or amplitude to regulate. During the loading up of boiler, the general approach is to ensure safety and stability and operate in the safety range given by the manufacturer, which will usually leave a large margin. Thus, it is difficult to make maximum use of its loading up capacity in practice, and it takes more time in the face of emergencies, such as setting sail with an emergency start.

In recent years, some experts and scholars have done some research on the modeling and matching operation of the SBI, and some results have been achieved [3-9]. Zhu Yong established the dynamic mathematical model of combustion and evaporation system of boiler, and conducted some simulation experiments. The dynamic responses of drum pressure, water level and superheated steam temperature of the supercharged boiler in different load disturbance were obtained [10]. Based on improved characteristic calculation method of turbo compressor, the stable and dynamic simulation model of turbo compressor with higher precision was established, and the acceleration process was simulated and analyzed by Zhao Donglai [11]. The flow characteristic of turbo compressor during acceleration and deceleration was studied based on fluid network, and the power balance point in different atmospheric temperature and the influence the variety

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of coefficient of excess air (CEA) has on turbo compressor were analyzed by Fang Tongyi [12]. These studies mainly focus on the analysis of the characteristics of the supercharged boiler and the turbo compressor, without further analysis and research on the control strategy of the SBI.

A type of marine SBI was taken as study subject, and based on the analysis of its characteristics of loading up, the mechanism simulation model of a type of marine supercharged boiler was structured. The CEA was studied during the load process through the research of simulation experiment. By adjusting the loading rate or amplitude of boiler, the CEA was kept in the fluctuation range in which the boiler can combust stably, and the minimum time of loading up under different initial conditions was got. The experimental results can guide setting the parameters of the actual monitoring system and provide support for the safe operation of this type of SBI.

## 2. Mechanism Model of SBI

A type of marine SBI was taken as study subject in this paper. According to its equipment composition and working medium flow direction, the marine SBI was divided into five modules of boiler furnace, turbo compressor, air duct system, air interlayer and flue system by using modular modeling method.

### 2.1. Boiler furnace

Furnace is a place where the mixture of fuel and air combustion heat release. Because of the compact structure of SBI, the boiler furnace can be considered with lumped parameter during combustion, and the energy conservation equation of combustion is

$$0.5m_y C_{py}\, d\overline{T}_y / dt = W_r H_r (1-k_b) - (W_r + W_k)(\overline{T}_y - T_0) - Q_F, \tag{1}$$

where $m_y$ is the quality of furnace gas; $C_{py}$ is the specific heat capacity at constant pressure of flue gas; $\overline{T}_y$ is the flue gas average temperature; $W_r$ and $W_k$ are the injection quantity of fuel and air; $H_r$ is the chemical exergy of fuel; $k_b$ is the heat transfer coefficient of furnace; $T_0$ is the environmental temperature of cabin; $Q_F$ is the radiant heat per unit time.

The calculation formula for the quality of flue gas in furnace is

$$m_y = P_b V_b / R_g \overline{T}, \tag{2}$$

where $P_b$ is the pressure of furnace; $V_b$ is the furnace volume; $R_g$ is the state parameters of ideal gas.

The formula for calculating the radiant heat transfer is

$$Q_F = \sigma A_{sl} (\overline{T}_y^4 - \varepsilon_{sl} T_{sl}^4), \tag{3}$$

where $\sigma$ is the Boltzmann's constant; $A_{sl}$ is the effective radiation area of water wall; $T_{sl}$ is the wall mean temperature for water wall; $\varepsilon_{sl}$ is the gray scale of the water wall and $\varepsilon_{sl} = 0.9$.

The calculation formula for the CEA in furnace is

$$\alpha = W_k / (13.865 W_r). \tag{4}$$

### 2.2. Turbo compressor

Turbo compressor can be divided into four modules of compressor, flue gas turbine, auxiliary steam turbine and generator rotor.

#### 2.2.1 Compressor

With the power mainly provided by flue gas turbine and auxiliary steam turbine, compressor compresses air to a certain pressure and temperature into boiler furnace to help combustion.

The formula for the power of compressor is

$$N_c = G_k C_{Pk} T_{k0} [\pi_c^{(K_k - 1)/K_k} - 1] / \eta_c, \tag{5}$$

where $G_k$ is the mass flow rate of air through compressor; $C_{Pk}$ and $T_{k0}$ are the specific heat capacity at constant pressure and inlet temperature of the air; $\pi_c$ is the compression ratio of compressor.

The calculation formula for internal efficiency of compressor $\eta_c$ is

$$\eta_c = 1 - (\eta_f + \eta_\theta + \eta_y + \eta_w). \tag{6}$$

where $\eta_f$ is the energy loss of compressor in the working process includes the friction loss of impeller, the sector loss $\eta_0$, the air leakage $\eta_y$ and the blast loss $\eta_w$ [13].

#### 2.2.2 Flue gas turbine

Flue gas turbine is the main power source of the compressor. Because the gas in high temperature can also be considered as the ideal gas, the calculation formula for output power of gas turbine is

$$N_{gt} = G_y C_{Py} T_{y0}[1-\varepsilon_{gt}^{(K_g-1)/K_g}]\eta_{gt}, \tag{7}$$

where $T_{y0}$ is the inlet temperature of flue gas; $G_y$ and $C_{Py}$ are the mass flow rate and the specific heat capacity at constant pressure of flue gas; $\varepsilon_{gt}$ is the expansion ratio of flue gas; $K_g = 1.35$ is the adiabatic coefficient of flue gas; $\eta_{gt}$ is the internal efficiency of flue gas turbine.

#### 2.2.3 Unit rotor

According to the energy conservation equations, the kinetic equation of the rotor is

$$a_a = 30(N_{st} + N_{gt} - N_c - N_{loss})/(\pi J_a n_a), \tag{8}$$

where $N_{st}$ is the output power of gas turbine; $N_{gt}$ is the output power of auxiliary turbine; $N_c$ is the power consumption of compressor and $N_{loss}$ is the power loss of rotor.

The calculation formula for rotor loss power is

$$N_{loss} = 0.01(N_{st} + N_{gt}) + 500. \tag{9}$$

According to Newton's second law,

$$dn_a^2/dt = 60[0.99(N_{st} + N_{gt}) - N_c - 500]/(\pi J_a). \tag{10}$$

## 2.3. Air interlayer

The mass flow rate inlet and outlet the air interlayer is mainly composed of air mass flow rate inlet $G_{jca1}$, air inlet of furnace $G_{bk}$ and air leakage $G_{jcl}$, which can be obtained by mass conservation equation,

$$K_{NPk} V_{jc}\, dP_{jc}/dt = G_{jca1} - G_{bk} - G_{jcl}, \tag{11}$$

where $P_{jc}$ is the air pressure in the air layer; $K_{NPk} = \partial\rho_k/\partial P_k$ is the coefficient of air compression and $V_{jc}$ is the air volume.

The air mass flow rate inlet and the air mass flow rate inlet of furnace are

$$\begin{cases} G_{jca1} = \xi_{c2jc}\sqrt{P_{c2} - P_{jc}} \\ G_{bk} = \xi_{jc2b}\sqrt{P_{jc} - P_b} \end{cases}, \tag{12}$$

where $G_{jca1}$ is the air mass flow rate inlet; $G_{bk}$ is the air mass flow rate inlet of furnace; $\xi_{c2jc}$ and $\xi_{jc2b}$ are the resistance coefficient of the inlet duct of air interlayer and the air distributor of boiler.

## 2.4. Air duct system

Air duct system consists of three pressure nodes, namely the atmosphere, the inlet of compressor and the outlet of compressor, and two flow branches, namely the branch from atmosphere to compressor inlet and the branch from compressor outlet to air interlayer. Where the parameters of the atmospheric joint are given, the parameters of the inlet of compressor and the outlet of compressor are calculated by iterative calculation with the fluid network model as shown in Eqs.(24).

$$\begin{cases} P_{c1} = P_0 - (G_k/\xi_{a2c})^2 \\ K_{NPa} V_{cpn}\, dP_{c2}/dt = G_k - \xi_{c2jc}\sqrt{P_{c2} - P_{jc}} \end{cases}, \tag{13}$$

where $\xi_{a2c}$ and $\xi_{c2jc}$ are the resistance coefficient of air duct from the atmosphere to the air inlet and outlet of compressor; $V_{cpn}$ is the volume of the outlet duct of compressor; $K_{NPa}$ is the compression coefficient of air.

## 2.5. Flue system

Flue system consists of six pressure nodes, namely the outlet of convective evaporation tubes, the outlet of superheater, the outlet of economizer, the inlet of flue gas turbine , the outlet of flue gas turbine and the atmosphere, and six flow branches. Similar to the air duct system, the pressure of nodes varies with the change of the flow rate of each branch except for the atmospheric node, and the calculation is based on the iterative calculation with the fluid network model,

$$\begin{cases} P_{\mathrm{gt2}} = P_0 + (G_{\mathrm{y}} / \xi_{\mathrm{g2a}})^2 \\ P_{\mathrm{gt1}} = P_{\mathrm{jj}} - (G_{\mathrm{geg}} / \xi_{\mathrm{ge}})^2 \\ P_{\mathrm{jj}} = P_{\mathrm{gr}} - (G_{\mathrm{jjg}} / \xi_{\mathrm{jj}})^2 \\ P_{\mathrm{gr}} = P_{\mathrm{dl}} - (G_{\mathrm{grg}} / \xi_{\mathrm{gr}})^2 \\ P_{\mathrm{dl}} = P_{\mathrm{b}} - (G_{\mathrm{dlg}} / \xi_{\mathrm{dl}})^2 \\ K_{\mathrm{NPy}} V_{\mathrm{ge}}\, dP_{\mathrm{gt1}} / dt = G_{\mathrm{geg}} - \xi_{\mathrm{g2a}} \sqrt{P_{\mathrm{gt2}} - P_0} \\ K_{\mathrm{NPy}} V_{\mathrm{jj}}\, dP_{\mathrm{jj}} / dt = G_{\mathrm{jjg}} - G_{\mathrm{geg}} \\ K_{\mathrm{NPy}} V_{\mathrm{gr}}\, dP_{\mathrm{gr}} / dt = G_{\mathrm{grg}} - G_{\mathrm{jjg}} \\ K_{\mathrm{NPy}} V_{\mathrm{dl}}\, dP_{\mathrm{dl}} / dt = G_{\mathrm{dlg}} - G_{\mathrm{grg}} \\ K_{\mathrm{NPy}} V_{\mathrm{b}}\, dP_{\mathrm{b}} / dt = G_{\mathrm{jca2}} + G_{\mathrm{boil}} - G_{\mathrm{dlg}} \end{cases}, \tag{14}$$

where $\xi_{\mathrm{g2a}}$, $\xi_{\mathrm{ge}}$, $\xi_{\mathrm{jj}}$, $\xi_{\mathrm{gr}}$ and $\xi_{\mathrm{dl}}$ are the resistance coefficient of chimney, flue gas purification device, economizer, superheater and convective evaporation tubes; $P_{\mathrm{gt2}}$, $P_{\mathrm{gt1}}$, $P_{\mathrm{jj}}$, $P_{\mathrm{gr}}$ and $P_{\mathrm{dl}}$ are the gas pressure of the outlet of flue gas turbine, the inlet of flue gas turbine, the outlet of economizer, the outlet of superheater and the outlet of convective evaporation tubes; $G_{\mathrm{geg}}$, $G_{\mathrm{jjg}}$, $G_{\mathrm{grg}}$, $G_{\mathrm{dlg}}$, $G_{\mathrm{jca2}}$ and $G_{\mathrm{boil}}$ are the flue gas flow rate of flue gas purification device, economizer, superheater, convective evaporation tubes, the air inlet of furnace and the amount of fuel injection; $V_{\mathrm{ge}}$, $V_{\mathrm{jj}}$, $V_{\mathrm{gr}}$, $V_{\mathrm{dl}}$ and $V_{\mathrm{b}}$ are the volume of flue gas circulation parts of flue gas purification device, economizer, superheater, convective evaporation tubes and furnace.

Based on the above model and the existing research results [4, 5, 9, 10], the whole system model of a marine SBI is built, and its topological structure diagram is shown in Fig. 1.

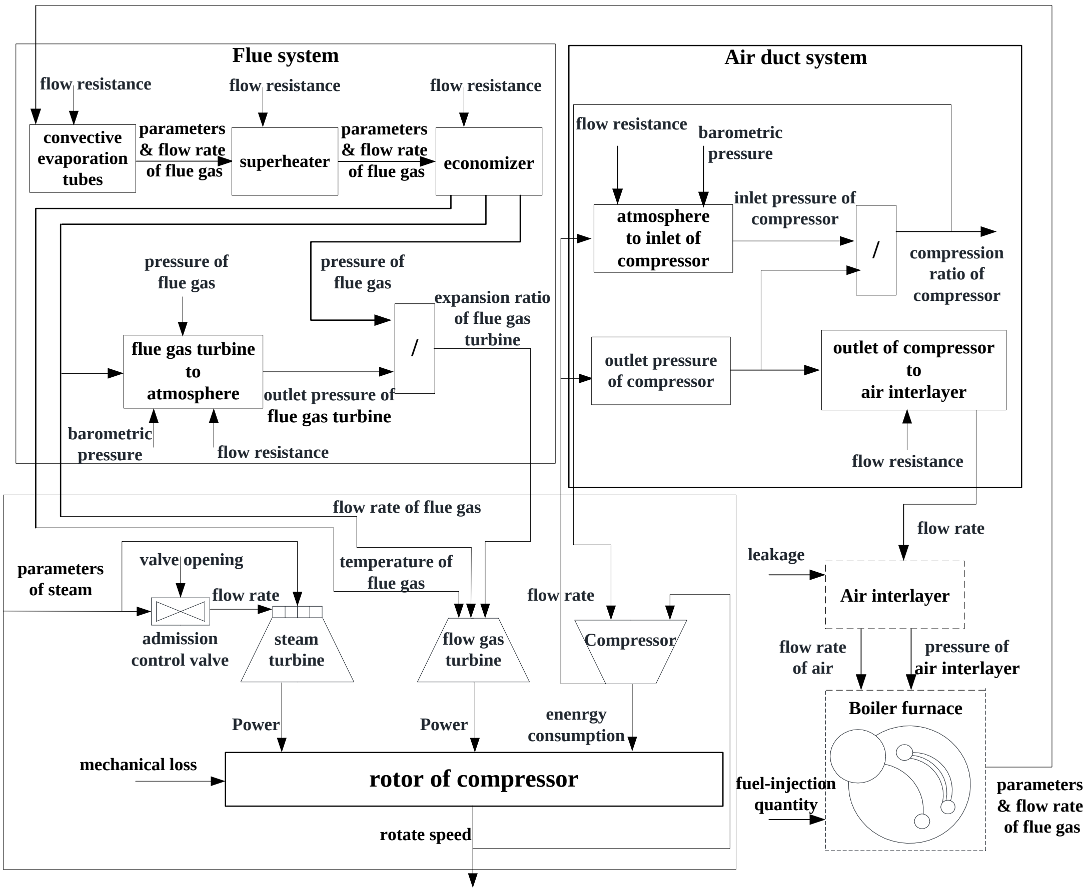


Fig. 1: Topological structure diagram of SBI model

# 3. Simulation Experiment and Result Analysis

If the CEA is too large or too small, it will cause the boiler to extinguish. In order to get the conclusion from the experimental result directly, the boundary value of the CEA is set in the simulation experiment. The maximum and minimum boundary values reflect the maximum and minimum inlet air volume that can be allowed under the condition of ensuring the combustion stability of boiler (without extinction). Because the boundary value of CEA is related to the structure of boiler, according to the practical operation data of the boiler, in the experiment the maximum CEA is 2. 3, and the minimum is 0. 8. It is assumed that the boiler combustions stably when the CEA is in the range of 0. 8~2. 3.

## 3.1. Simulation experiment of loading up with uniform velocity

It is assumed that the boiler is in the stable operation state after ignition with the minimum load, and the fuel flow rate control valve angle is 0 degrees. The initial state time T=0 s. In T=2 s, open the fuel flow rate control valve to increase fuel injection quantity with different uniform velocities, to make the boiler load increasing gradually from the minimum load to full load, where the fuel flow rate control valve angle is 315 degrees. The experimental results are shown in Fig. 2, and each line represents a curve of CEA with different valve opening velocities. The valve opening velocities from the bottom to top are 3.15 degrees/sec, 1.58 degrees/sec, 1.38degrees/sec and 1 degrees/sec respectively.

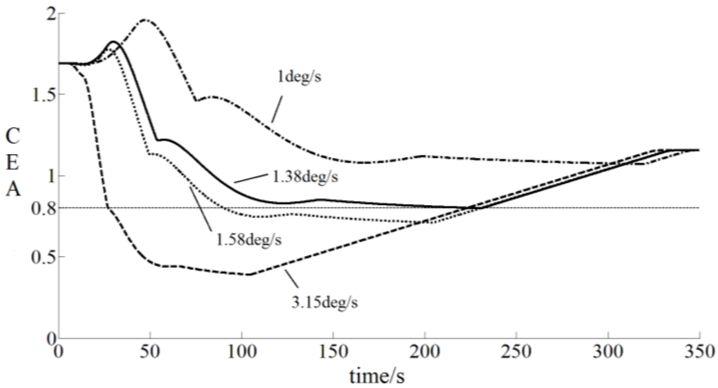


Fig. 2: Curve of CEA with different valve opening velocities

As Fig. 2 shows, due to the rise in fuel injection quantity, the flow rate of flue gas increases. However, the air inlet flow rate can not respond immediately to a change in boiler load and the CEA decreases because of the mechanical inertia of turbo compressor. The rate of decline is proportional to the opening speed of fuel flow rate control valve, and the time to reach the highest point is inversely proportional to the opening speed of fuel flow rate control valve. When the valve opening speed is 1.38 degrees/sec, the minimum CEA is 0.8, and the transition time is 338.2 s. Thus, as to this type of supercharged boiler, the opening speed of fuel flow rate control valve should not exceed 1. 38 degrees/sec.

## 3.2. Control strategy research on loading up rapidly

In the process of ship maneuvering, we are more concerned about how to make the boiler up to the target load in the shortest time, as well as to ensure the premise of safe operation. On the basis of experiment on loading up with uniform velocity, the control strategy of loading up rapidly of the booster boiler is further studied. In order to analyze the degree of influence the auxiliary turbine has on the process of lifting load, the auxiliary steam turbine is put into operation before the load process. The experimental results are shown in Fig.3, in which the different curves show different turbine start times, which are 5 seconds, 2 seconds and 0 seconds in turn from top to bottom.

By comparing Fig. 2 and Fig. 3, it can be defined that whether the auxiliary steam turbine is put into operation in the process of lifting load has great influence on the combustion of the boiler. The peak value of the CEA is proportional to the leads of starting the auxiliary steam turbine, the greater the leads, the greater the peak value is. Put the auxiliary steam turbine into work in parallel with the start of lifting load, and the valve opening velocity can be increased to 2. 23 degrees /s, besides the stable time can be reduced to 224. 2 s, whose reduction rate is 32. 9%, compared with not putting the auxiliary steam turbine into work. It is the addition power of auxiliary turbine that makes the rotate speed of compressor increased, and then the air flow rate into the furnace increases, thus the loading up velocity can be accelerated.

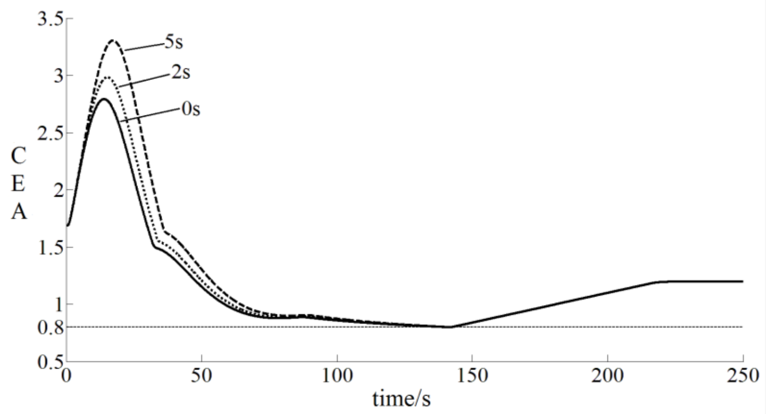


Fig. 3: Curve of CEA with prestarting the auxiliary turbine

If the auxiliary steam turbine is put into operation in advance in the process of loading up, the air flow rate into the furnace will increase suddenly in a short time, and the peak value of the CEA will exceed the maximum boundary value 2. 3. To this end, we will delay the start time of the turbine, and the experimental results are shown in Fig. 4, in which the different curves show different steam turbine delay start time, which are 0 seconds, 2 seconds and 5 seconds in turn from top to bottom.

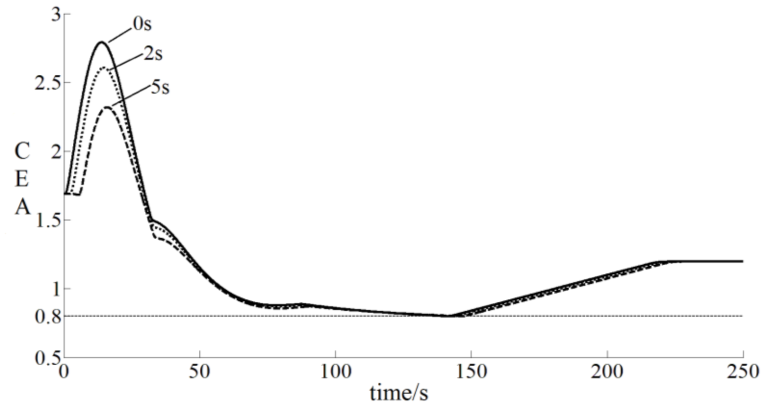


Fig. 4: Curve of CEA with delayed starting the auxiliary turbine

As Fig. 4 shows, compared with the control strategy of prestarting the auxiliary steam turbine, the one of delayed starting auxiliary steam turbine can greatly reduce the peak of CEA during the loading process under the premise that the whole loading time is almost the same (in fact slightly extended). When the auxiliary steam turbine is put into work in T=5 s, the peak value of CEA reaches the maximum boundary value of 2. 3. Thus the conclusion can be drawn that in order to ensure the stable combustion of the boiler, the auxiliary steam turbine should be put into operation 5 seconds after the loading up starts.

Comparison of Fig. 2, Fig. 3 and Fig. 4 shows that whether to and when to put the auxiliary steam turbine into work both have an impact on the velocity of loading up. First of all, to turn on the auxiliary steam turbine can greatly reduce the load time. Secondly, the earlier to put the auxiliary steam turbine into work, the larger the peak value of CEA is, and the shorter the load time is.

Through the simulation experiment on loading up rapidly, it is shown that the segmented control strategy is slightly better than the uniform loading up strategy, while whether to and when to turn on the auxiliary steam turbine is more effective to shorten the time it takes to stabilize after loading up.

## 4. Conclusion

The loading up characteristics of SBI are analyzed, and the mechanism modeling of a certain type of marine SBI is built with the modular modeling method. Finally, the loading up control strategies of the boiled are studied through simulation experiments, and some qualitative and quantitative conclusions are drown as follows.

(1) The maximum velocity to open the fuel flow rate control valve uniformly during the loading up process of this type of supercharged boiler is 1.38 degree / s, and the transition time for stabilization is 334. 3 s.

(2) Whether to and when to turn the auxiliary steam turbine into work are concerned with the loading up time, and the earlier to start it, the shorter the loading up time is, and the larger the peak value of CEA is.

The above conclusions can provide reference for the design of the actual monitoring system and the parameters setting of the regulating valve, and can also provide support for the logic and algorithm research on regulation, control and protection.

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