Experiment Study and Industrial Application of Slotted Bluff-Body Burner Applied to Deep Peak Regulation

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ABSTRACT

During the deep peak shaving period, the boiler needs to operate at a lower load, so higher requirements were set for the boiler's stable combustion. In order to better evaluate and compare the stable ignition capacity of bluff-body burners and slotted bluff-body burners, guidance and calculation support for the design of boiler deep peak shaving were needed. This study adopted the reflux heating chain ignition analysis method to study the differences in the steady combustion mechanism between the bluff-body burner and the slotted bluff-body burner. A thermal combustion experiment was conducted in employing the single angle coal powder combustion furnace. The experimental results were compared against the theoretical analysis results. In addition, a study was conducted on the application of slotted bluff-body burners in the deep peak shaving of a 330MW unit power plant boiler. The results showed that as long as the small slot flow can catch fire, the mixed temperature of the reflux fluid of the slotted bluff-body burner will be higher than that of the bluff body burner. This will enhance its steady combustion ability. The combustion test results were found to be consistent with the analysis results, indicating that when the slotted bluff-body burner is used on the 330MW unit, the boiler combustion is in good condition. In this case, it has a stable combustion capacity of 66MW (20% economic continuous rating) without oil injection at low loads. This study revealed the advantages of the slotted bluff-body burner in terms of its steady combustion mechanism compared to traditional bluff-body burners. It verified the feasibility for boiler deep peak shaving in practical applications. This is of great significance for improving the flexibility of coal-fired power generation units, enhancing the flexibility of the power system, and promoting carbon reduction.

KEYWORDS

Bluff Body Burner, Boiler Deep Peak Shaving, Ignition Analysis, Reflux Heating Chain, Slotted Bluff-Body Burner, Steady Combustion Mechanism

INTRODUCTION

With the large-scale deployment of intermittent renewable energy, represented by wind and solar energy, the demands for flexibility in the power system are continuously increasing (Dai, H. et al.,

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2016; Johansson, T. et al., 2012), China relies on coal as its primary energy source (Qi, Y. et al., 2016). Thus, the implementation of a flexible transformation of the thermal power has become the most realistic and feasible choice to improve the power system flexibility (Chen, H. et al., 2021; Xiao, D. et al., 2014; Gong, S. et al., 2017; Ding, Y. et al., 2014). On this basis, in-depth research on the peak shaving technology of large-scale coal-fired units has become necessary. Grid peak shaving involves many aspects such as boilers, turbines, generators, auxiliary equipment, and transmission systems (Gu, Y. et al., 2016; Gao, Z. et al., 2021; Xue, Y. et al., 2019; Manojkumar, R. et al., 2022; Wang, J. et al., 2021). In terms of boilers, the stable combustion under low-load is a key issue, and the main solution is the development of stable combustion technology (Cheng, H. et al., 2021). Researchers have developed stable combustion technologies, including bluff bodies, ship types, pre-combustion chambers, large differential speeds, asymmetric jets, reverse blowing, and coal concentration (Zhang, H. et al., 2007; Tsumura, T. et al., 2003; Zeng, G. et al., 2017; Hu, F. et al., 2022; Wang, J. et al., 2009; Chayalakshmi, C. et al., 2009). Among them, the bluff-body stable combustion technology integrates a bluff body at the nozzle of the burner to form a recirculation zone at the nozzle outlet. This allows the fuel to ignite and burn stably under conditions of high-speed airflow. During combustion, the reflux behind the bluff body heats the fuel-air mainstream at the root of the air mass by entraining a high-temperature flue gas, thereby accelerating the exchange of momentum, mass, and energy. This is conducive to flame stability (Zhang, L. et al., 2011). However, the unburned fuel flow can only enter the recirculation zone through the boundary layer. This prevents the beneficial effect of the high-temperature recirculation zone behind the bluff body from being fully harnessed. In comparison, the slotted bluff-body stable combustion method sets a gap in the middle of the bluff body to directly introduce a small amount of fuel flow into the high-temperature recirculation zone. In this case, this fuel can be ignited first in a favorable environment before igniting the mainstream. This allows for a staged ignition mechanism in the recirculation zone and further improves the stability of combustion. At the same time, it does not destroy the basic structure of the recirculation zone behind the bluff body. (Du, Y. et al., 2006; Liao, Y. et al., 2022; Yan, Y. et al., 2019). However, there is a lack of in-depth analysis on why the slotted bluff-body combustion has good stability.

This study addressed the question by analyzing the thermal balance equation. Also, practical application research was conducted on a 330MW unit, and the application effect of slotted bluff-body burners in deep peak shaving of boilers was verified. The research results showed that in practical applications, the slotted bluff-body burner exhibits a good low-load stable combustion ability without oil injection, proving its feasibility and practicality in actual power systems. The innovative aspects of this study are characterized by the following: (1) the reflux heating chain ignition analysis method was used to construct the thermal balance equation, explaining the differences in the steady combustion mechanisms between the bluff-body burner and the slotted bluff-body burner; (2) the difference between the two burners in steady combustion was evaluated and highlighted through an experimental study; (3) the slotted bluff-body burner was applied in a practical application and improved the boiler's ability to achieve deep peak shaving, reducing the minimum steady combustion load of the boiler from the original 45% ECR to 20% ECR.

RESEARCH METHODS

Comparative Analysis of Theoretical Foundations

According to the traditional recirculation zone fire theory, a high-temperature flue gas that has already ignited flows back to the root of the main airflow in the recirculation zone to mix and heat the main flow. This causes the gas to reach ignition temperature and combust. When stable combustion is achieved, a closed reflux heating ignition chain is formed.

As shown in Figure 1, the unburned mainstream in the bluff-body flame stabilizer gathers a portion of the fresh fuel-air flow near the recirculation zone. At location 1, it mixes with the hot gas in the recirculation zone. Then, it flows to location 2 to start combustion. After that, the fluid burns while entering the recirculation zone. The reaction ends at location 3, forming a closed ignition chain loop. The mixing, heating, and intense reaction mainly occur in the shear layer near the intersection of the main flow and the reflux. This region has a large gradient of velocity, mass, and temperature along the radial direction. However, the radial temperature difference near the central axis of the recirculation zone of the bluff body (the lower edge of the ignition chain loop) and near the central axis of the main flow (the upper edge of the ignition chain) is relatively small. Therefore, it can be assumed that except for the fresh unburned fuel-air flow and the high-temperature flue gas flow that flows out and enters the reflux ignition chain loop through the boundaries of the main flow and reflux, there is no other thermal and mass exchange process with the external environment in the ignition chain system.

Based on the above analysis, assuming that the specific heat capacity of the material flow in the ignition chain loop remains constant, the reflux heating ignition chain theory could be used to compare and analyze the flame stability of the bluff-body and slotted bluff-body burners.

The ignition chain loop of the bluff body is represented in Figure 2. Assuming that the mass flow rate of the fuel gas flow main stream is denoted by unit 1, T_0 is the initial temperature of the fuel gas flow, T_m is the mixed temperature, R is the mass flow ratio of the reflux flue gas and fuel gas flow, which represents the reflux rate, and Q_c is the heat released per unit of fuel combustion. In the process, the total mass flow of the loop remains constant due to the closed nature of the reflux ignition chain loop. Under stable combustion conditions, the proportion of the unburned fuel gas flow mixed with the reflux flue gas is also fixed. Therefore, the fluid temperature after mixing can be calculated using the heat balance equation.

Equations (1) and (2) present the heat balance equations for nodes m and f, respectively:

Figure 1. Schematic diagram of bluff-body reflux combustion

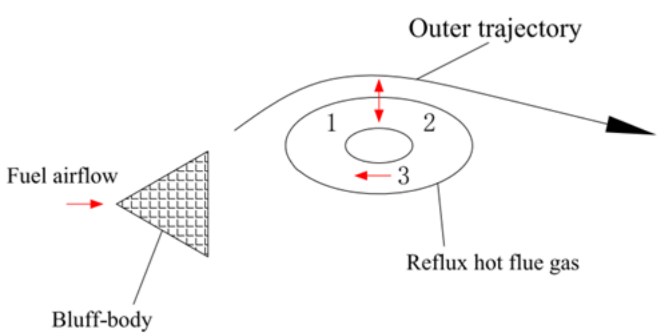
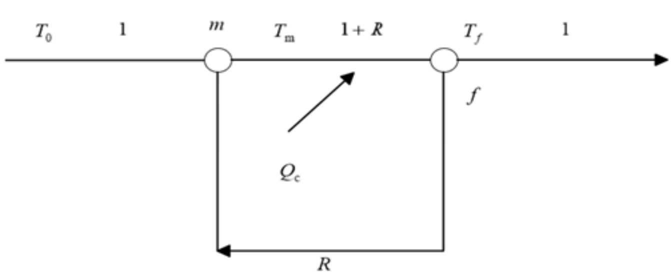


Figure 2. Ignition chain loop of bluff-body burner



$$ST_0c_p + RT_fc_p = (1+R)T_mc_p \quad (1)$$
$$\left(1+R\right)T_mc_p + Q_c = (1+R)T_fc_p \quad (2)$$

Based on equations 1 and 2, the temperature after mixing the fuel gas flow can be obtained, as shown in Equation (3).

$$\Delta T_m = T_m - T_0 = \frac{Q_c}{c_p} \times \frac{R}{1+R} \quad (3)$$

When the heat release per unit of fuel combustion, Q_c, and the specific heat capacity, c_p, remain constant, the higher the reflux rate, R, the higher the temperature of the fuel gas flow after mixing. In this case, the flame is more likely to be stable. In theory, when $R \to \infty$, the reflux heat becomes equal to Q_c. It is worth noting here that in practice, the reflux rate cannot be greater than 1 and is generally less than 0.5, which greatly limits the mixing temperature of the fuel gas flow.

In addition, this model is based on the traditional recirculation zone reflux theory, which explains the flame stabilizing effect of the bluff body through the theory of reflux heating. Within the slotted bluff-body combustion stabilizer, there are gaps in the middle of the bluff body that allow a small amount of fuel gas flow to enter the recirculation zone. Experiments have shown that an appropriate gap flow will not destroy the low-speed reflux region behind the bluff body (Yan, Y. et al., 2019). In the high-temperature and low-speed reflux environment of the recirculation zone, the gap is extremely easy to ignite. Thus, heat transfer through reflux still exists in this area.

The schematics of the reflux combustion with the slotted bluff body and the ignition chain loop are shown in Figures 3 and 4, respectively.

In the process, a portion of the fuel gas flow is diverted from the mainstream to enter the reflux area through the bluff body. This proportion of the fuel flow to the mainstream is set to S. After mixing with the reflux flue gas, the resulting temperature is T_{mk}, where the flue gas temperature after combustion is T_{fk}. The heat balance equations for nodes m', mk, and fk are as follows:

$$ST_0c_p + RT_{fk}c_p = (S+R)T_{m'}c_p \quad (4)$$
$$(1-S)T_0c_p + (S+R)T_{m'}c_p + SQ_c = (1+R)T_{mk}c_p \quad (5)$$
$$(1+R)T_{mk}c_p + (1-S)Q_c = (1+R)T_{fk}c_p \quad (6)$$

Figure 3. Schematic diagram of slotted bluff-body reflux combustion

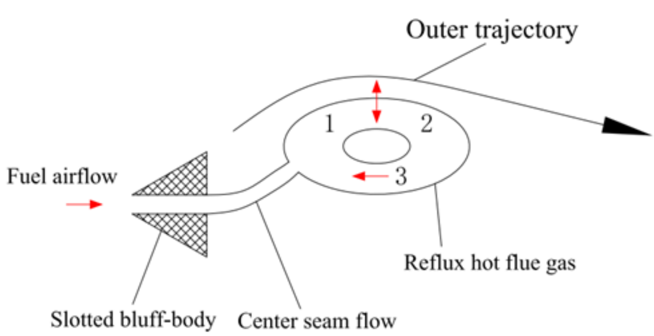
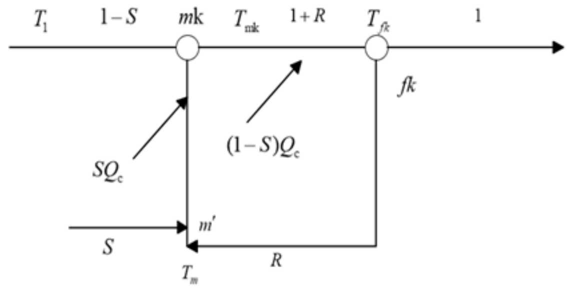


Figure 4. Ignition chain loop of slotted bluff-body burner



By combining equations (4), (5), and (6), the mixed temperature of fuel gas flow in the slotted bluff body can be calculated:

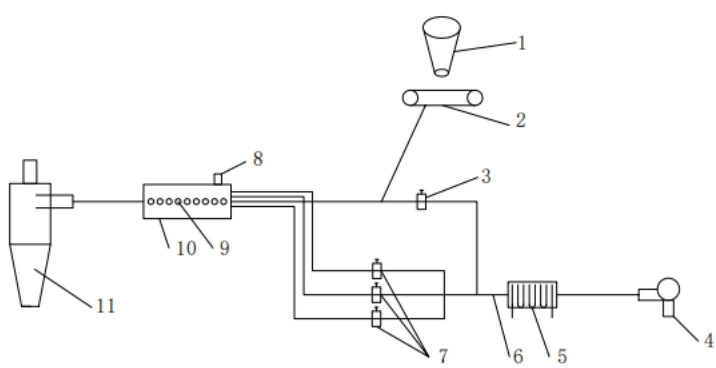
$$\Delta T_{mk} = T_{mk} - T_0 = \frac{Q_c}{c_p} \times \frac{R}{1+R}(1+\frac{S}{R}) \quad (7)$$

It is noted that the greater the temperature difference of the fuel after combustion, ΔT_{mk}, the greater the heat provided, indicating conditions more conducive to combustion. It can be seen from equation 7 that ΔT_{mk} is related to the unit fuel heat release, Q_c, the physical property parameter, c_p, the reflux rate, R, and the slot flow rate, S.

Comparative Analysis of Experimental Research

To further verify the stable combustion characteristics of the slotted bluff-body burner, a comparative analysis was conducted on a single-cornered cylindrical horizontal furnace, shown in Figure 5, between the bluff-body burner and the slotted bluff-body burner using Loping coal (as detailed in Table 1). The following parameters were considered: primary air velocity v_1 =25 m/s, secondary air velocity v_2 =45 m/s, primary air ratio of 20%, ratio of slotted bluff-body width to channel width $b / B = 0.5$, and ratio of slot width to bluff body width $\delta / b = 0.10$

Figure 5. Schematic diagram of single-cornered cylindrical horizontal furnace



1- Coal Powder Silo, 2- Feeder, 3- Primary Air Regulating Damper, 4- Fan, 5- Electric Heating Air Preheater, 6- Supply Air Duct, 7- Secondary Air Regulating Damper (3 in number), 8- Furnace Observation Hole, 9- Testing Hole, 10- Single-corner Furnace Combustion Chamber, 11- Dust Collection Device

EVALUATION OF INDUSTRIAL APPLICATION EFFECTS

A 330MW thermal power plant was considered for investigation in this study. The plant has a subcritical pressure primary intermediate reheat natural circulation boiler with a steam generation capacity of 1180t/h. The boiler operates under subcritical conditions, along with natural circulation, single steam drum, single furnace balanced ventilation, four-corner tangential firing, primary intermediate reheat, balanced ventilation, swinging burner temperature control, tight-sealing enclosure, full steel frame (main and auxiliary double frame) Π-type steam drum boiler, solid waste disposal, and pulverized coal furnace. The air preheater employs a three-compartment rotary (Koch) preheater. The flue gas is also equipped with desulfurization and denitrification devices. Five medium-speed mills are installed in the plant. The boiler's main parameters are listed in Table 2.

The original boiler is arranged with four-corner tangential firing and vertical rich and lean bluff body burners, as shown in Figure 6 (a). Additionally, a "louver" style vertical coal powder concentrator is used in the primary air duct. The burner box is divided into several air chambers by partition plates. Nozzles are arranged at the outlet of each air chamber, and the secondary air damper plate is placed at the inlet of the air chamber. A micro-oil ignition system is integrated in the primary air chamber of layer A for cold starting and stable combustion at low loads. In the current modification, the existing coal powder burners (A, B, and C layers) are replaced with slotted bluff-body burners. Considering that the modification of the burner will not affect its load-bearing capacity, it is not advisable to set a very large blunt body. On this basis, the slot width accounts for 12.9% of the edge width of the blunt body, as shown in Figure 6 (b).

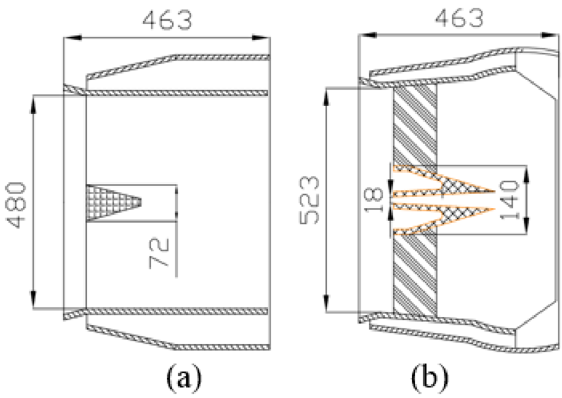
Table 1. Coal quality analysis

Proximate Analysis (wt %, as Received Basis)				Qar.net (kJ/kg)
Volatile Matter	Moisture	Fixed Carbon	Ash	
12.80	1.46	33.64	52.10	13820

Table 2. Boiler design parameters

Name	Unit	Boiler Maximum Continuous Rating	Turbine Heat Acceptance
Superheated steam flow	t/h	1180	1034.6
Superheater outlet steam pressure	MPa	17.5	17.3
Superheater outlet steam temperature	K	814	814
Reheat steam flow rate	t/h	991	875.8
Reheater inlet steam pressure	MPa	4.18	3.695
Reheater outlet steam pressure	MPa	3.96	3.502
Reheater inlet steam temperature	K	613	601
Reheater outlet steam temperature	K	814	814
Economizer inlet feed water temperature	K	551	543

Figure 6. Burner schematic diagram: (a) the original burner before modification, (b) the modified burner after modification



RESULTS AND ANALYSIS

Theoretical Aspect

By comparing equations 3 and 7 regarding the temperature of the mixed fluid after reflux through the bluff body and the slotted bluff body, respectively, it can be concluded that under identical conditions, the temperature of the mixed fluid after recirculation through the slotted bluff body is higher than that after reflux through the bluff body by a factor of S / R, provided that the small slot flow can ignite (R>0). In this regard, previous experimental studies showed that the reflux rate R for both the bluff body and the slotted bluff body is generally between 10% and 20% (Xue, F. et al., 1996). Additionally, the optimal slot flow rate S for the slotted bluff-body is also between 10% and 20%. Therefore, it can be predicted that the temperature of the mixed fluid after passing through the slotted bluff body will be about twice that of the bluff body. This can fully explain the better stability of the slotted bluff-body burner compared to the bluff-body burner.

Moreover, the effects of the reflux rate R and the slot flow rate S on the mixed fluid temperature ΔT_m, as calculated by equations 3 and 7, are shown in Figure 7. It is shown that as long as the slot size of the slotted bluff body does not affect the reflux rate, increasing the width of the slot can deliver more fuel into the recirculation zone. Additionally, increasing the width of the slot provides more combustion heat and further increases the mixed fluid temperature. This greatly facilitates the ignition of the mainstream.

Theoretical Aspect

After 30 minutes of oil-coal co-combustion, the fuel oil was turned off. The slotted bluff-body burner was found to maintain stable combustion, while the bluff-body burner had to reduce the primary air flow rate to 70% of the original level to achieve stable combustion. Figure 8 shows the axial temperature distribution at different air ratios for the two burners.

The temperature behind the slotted bluff-body burner near the nozzle was significantly higher than that behind the tangentially-fired burner. However, later in the process, the temperature behind the tangentially-fired burner increased because of the lower primary air flow rate.

Figures 9 and 10 present a comparison of the slotted bluff-body burner and the bluff-body burner under the same primary air flow rate. As shown in Figure 9, due to the injection of primary air, the temperature at the center of the slotted bluff-body burner in the upper part of the combustion chamber was found to be lower than that of the bluff body but a higher temperature was measured around it. As X increased, the temperature of each part behind the slotted bluff-body burner was

Figure 7. Relationship between reflux rate and mixed fluid temperature

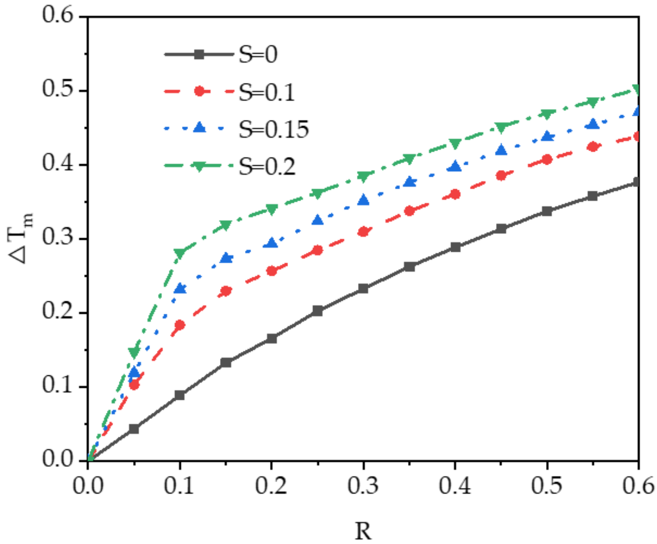
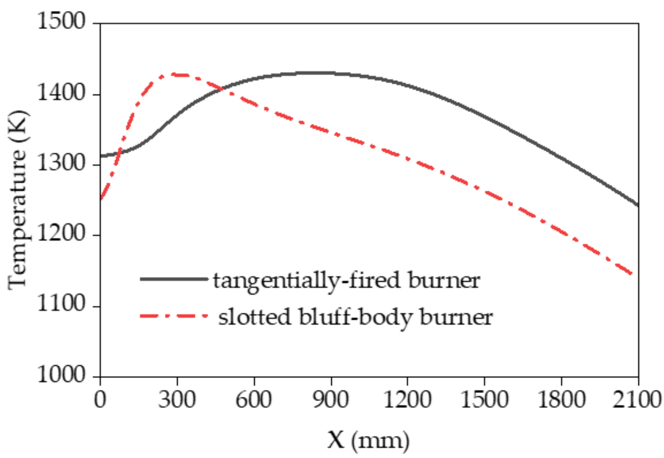


Figure 8. Axial temperature distribution of slotted bluff-body burner and tangentially-fired burner



higher than that of the bluff-body burner. Then, at X = 185mm, the temperature at the center of the slotted bluff-body burner was shown to increase sharply from 973K to 1423K. This indicates that ignition occurred in the recirculation zone. As shown in Figure 9, the flue gas temperature behind the slotted bluff-body burner after ignition was in the range of 343-403K higher than that behind the bluff-body burner along the entire axis of the combustion chamber.

Figure 9. Radial temperature distribution diagram of slotted bluff-body and bluff-body burner at axial distances X=55mm and X=185mm of combustion chamber head

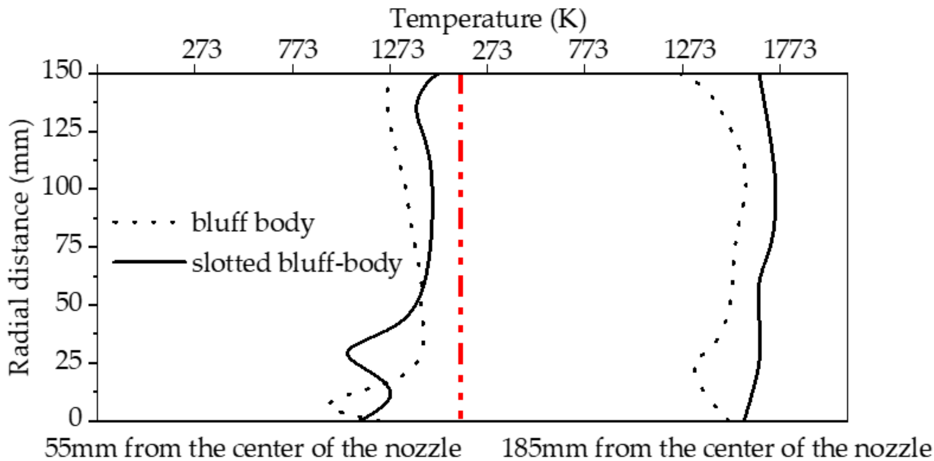
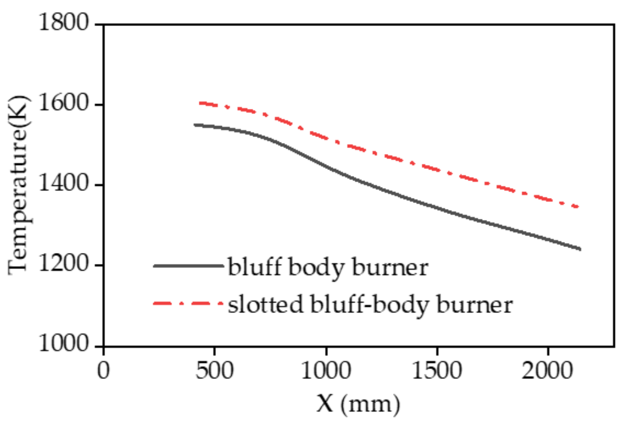


Figure 10. Axial temperature distribution of slotted bluff-body burner and bluff-body burner



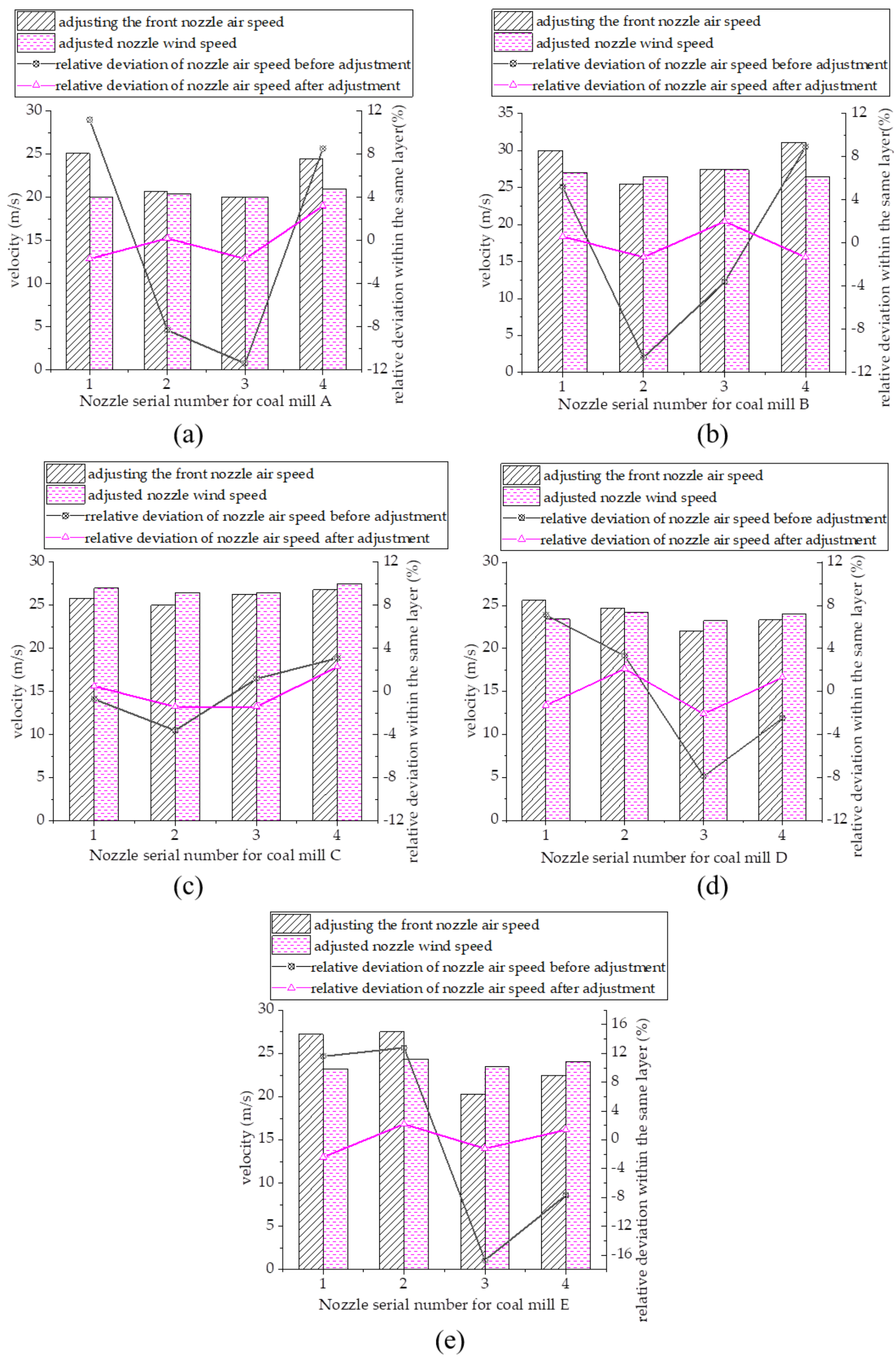
INDUSTRIAL APPLICATION SITUATION

Cold State Test of Boiler

During the experiment, measurements were collected for the wind speeds of the five layers of burner nozzles in the furnace. The corresponding layers of burner nozzles for the five coal mills A, B, C, D, and E were shown to have a certain deviation in the wind speed. After adjusting the contraction holes on the powder outlet pipes of each coal mill, the wind speed deviation of each layer of the burner nozzles was adjusted within the range of ±5%. The detailed data for each coal mill is shown in Figure 11.

*Nozzle of Each Layer of the Coal Mill: (a) coal mill A, (b) coal mill B, (c) coal mill C, (d) coal mill D, (e) coal mill E*It is shown in Figure 11 that before the primary air was leveled, the air speed deviation of the nozzle corresponding to the same coal mill was relatively high. In particular, the deviations of the three coal mills A, B, and E are the highest. The relatively high deviation of the primary air speed leads to the disturbance of the furnace combustion. In addition, there will be a

Figure 11. Air speed and relative deviation before and after adjusting corresponding burner



deflection of the tangential circle in the furnace and a burning flame scouring the wall surface of the water-cooling wall. After leveling, it is shown that the wind speed deviation of the same layer was within ±5%, which meets operational requirements.

The spark tracing test for the air dynamic field inside the furnace was conducted under the condition that the primary air had been leveled and the primary and secondary air speeds had met the conditions for cold state simulation. The spark tracer was mainly arranged at the primary air burner nozzle on the A, D, and E layers at the four corners of the furnace. Additionally, it was arranged on the secondary air nozzle on the AB, CD, and E2 layers. The flow of the sparks in the furnace was captured by a top-mounted monitoring camera to visually reflect the airflow trajectory of the primary and secondary air as well as the overall airflow in the furnace. The results of the cold dynamic field spark tracing test are shown in Figures 12 and 13.

As shown in Figure 12, the primary air-cooled tangential circle maintained good fullness in the furnace. In this case, the weakened rigidity of the primary air jet and the jet brushing the wall due to the change in the burner type presented no problems. Additionally, it can be seen from Figure 13 that the secondary air tangential circle was centered and maintained a good rotation. Thus, it was not affected by the change of the burner from the vertical pulverized coal concentrator of the "louver" type to the slotted bluff-body burner. Overall, the main findings of the fireworks tracing test in the aerodynamic field of the furnace are as follows: (1) the airflow in the primary combustion zone moved counterclockwise, where the center of the imaginary tangent circle of the burner is located at the center of the furnace, without any significant deviation, and (2) the shape of the primary and secondary air tangent circles was good. The overall size of the secondary air tangent circle is larger than that of the primary air tangent circle. This can form "wind-package coal" and is beneficial for controlling the slagging of the furnace (Kang & Xian, 2020; Zhao & Zhi, 2011).

Boiler Heat Operation Status

In this section, a deep peak shaving experiment was conducted to verify the effectiveness of the slotted bluff-body burner in the deep peak shaving of power plant boilers.

During the stable combustion test of the boiler, considering electric loads between 115MW and 66MW, the micro-oil ignition system, small oil combustion support system, and large oil gun system were not put into operation. Thus, the boiler was only relying on the slotted blunt-body burner itself to stabilize the combustion. Based on the results obtained, the stability of the coal flame detection in the operation layer was good, and the analog signal was greater than 90%. From the on-site observation

Figure 12. Sectional view of cold primary air flow circle

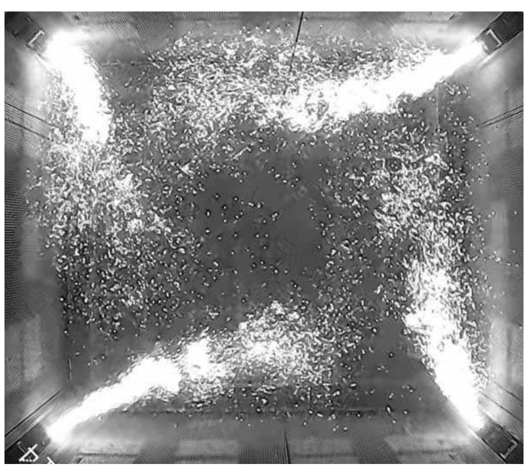


Figure 13. Sectional view of cold secondary air flow circle

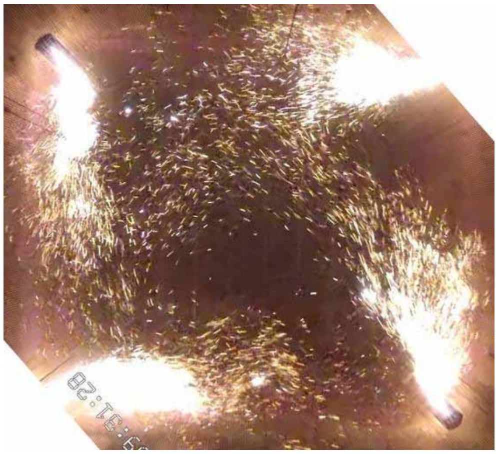


Table 3. Main parameters of boiler during deep peak shaving

Test Progress	Time	Current Load (MW)	Running the Coal Mill	Total Coal Quantity (t/h)	Operating Oxygen Content (%)	Lower part of Platen Superheater Temperature (°C)	Reduction Zone Temperature (°C)	Average Temperature of Burner Area (°C)
Start	September 16, 2022 12:13	115	ABC mill	62.15	6.12	719	1136	1057
Process 1	September 16, 2022 13:17	99	BC mill	55.25	6.48	684	1110	1011
Process 2	September 16, 2022 15:04	82.5	BC mill	49.19	8.01	657	1096	953
Process 3	September 16, 2022 16:26	66	BC mill	43.22	9.54	622	958	894
End	September 16, 2022 18:40	66	BC mill	43.12	9.50	628	961	898

of the fire hole, the flame was relatively bright, with no flickering. This indicated that the boiler is in good combustion condition and has a stable combustion capacity of 66MW (20% economic continuous rating) without oil input at low load in the furnace.

CONCLUSION

The main purpose of this study was to adopt the reflux heating chain ignition analysis method to develop the thermal balance equation, analyze the stable combustion characteristics of the slotted bluff-body burner, and conduct a thermal combustion test and theoretical results comparison on a single corner coal-pulverized combustion furnace. Based on the theoretical and experimental analysis results obtained, a stable combustion slotted bluff-body burner suitable for deep peak shaving under the 330MW-class unit was designed. Using this burner, the stable combustion effect was studied. The main conclusions of this work can be summarized as follows:

(1) In the ignition chain system, besides the fresh and unburned fuel air flow entering the reflux ignition loop through the main flow and reflux boundaries and the high-temperature burned gas flow flowing out, there was no other heat exchange indicated with the external environment.
(2) Increasing the middle slot width of the slotted bluff body appropriately without affecting the reflux rate allowed for delivering more fuel to the recirculation zone, increasing the mixing temperature. This was conducive to main flow ignition and allowed for a more stable combustion effect.
(3) Through the reflux heating chain ignition analysis and the experimental research on the single corner furnace stable combustion, it was confirmed that the mixing temperature of the slotted bluff-body burner is higher than that of the traditional bluff-body burner. This explains why the stable combustion effect of the slotted bluff-body burner is better.
(4) When using a slotted bluff-body burner in a 330MW-class unit boiler, the air dynamic field performed well and had excellent thermally stable combustion characteristics, achieving a 66MW (20% economic continuous rating) low-load stable combustion without oil injection.
(5) The slotted bluff-body burner had good combustion stability. It also promoted the combustion of pulverized coal where the temperature near the nozzle was high. This may cause overheating and damage to the burner nozzle. In real engineering applications, a nozzle with a high-temperature resistant material should be selected to improve its service life.

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