Design of a partially premixed burner for biogas-fired wall-mounted boiler

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Abstract

Biogas is deemed as one of the most promising renewable fuels emerging in the past several decades. At present, biogas is mainly applied to cooking for substituting fossil fuels. However, biogas is rarely reported to be used for space heating. In this paper, we designed a biogas-fired partially premixed burner for gas-fired wall-mounted boilers, which has the heat input of 25 kW and can be used for space heating and water heater tank heating. The reference gas was assumed as 60% of methane (CH₄) and 40% of CO₂, and the burner port area was designed as 4744.5 mm² according to the calculation. Then, experiments were conducted to study the performance of this burner. Jet diameters were modified in order to investigate their effects on heat input, primary air ratio and flame stability of the burner. The results indicate that the burner shows superior performance when the jet diameter is 2.0 mm. Meanwhile, the primary air ratio (α') satisfies the design requirement. Moreover, a gas-fired wall-mounted boiler with this burner was examined to investigate the influence of different biogas composition on exhaust emissions and thermal efficiency. The results show that the concentration of CO in the flue gas meets the national standard when CH₄ varied from 40% to 60% in the biogas. However, the thermal efficiency of the biogas-fired wall-mounted boiler reduces greatly with the increase of CH₄ percentage.

Keywords: biogas; partially premixed burner; wall-mounted boiler; jet diameter

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1. INTRODUCTION

Due to the depletion of fossil fuels and the increase of pollutant emissions, finding environmentally friendly alternative fuels is taken into account to guarantee the reliable energy supply and global health [1]. Biomass fuels may be an alternative solution to the future shortage of fossil fuels, and it is possible for many countries to produce energy from domestic renewable resources [2]. In China, many regions are short of clean energy, such as natural gas (NG), since it is too costly to install NG pipeline networks especially in rural areas [3,4]. However, almost all gas-fired boilers are designed for NG, meaning they are rarely employed in the rural locations away from NG pipe networks. Fortunately, the biogas has been widely used as an important source of rural renewable energy with the financial support of the Chinese government at present [5]. Because of adequate biogas resource (landfills [6], agricultural waste and other sources of biomass), designing biogas-fired wall-mounted boilers has a long-term significance in those areas. Not only can the boilers supply heating and hot water but also can heat the biogas fermentation tanks to improve the biogas outcomes in winters.

Biogas is mainly composed of 50–70% (by volume) methane (CH₄) and 25–40% (by volume) carbon dioxide (CO₂) [7, 8]. Due to the high CO₂ content, the combustion characteristics of the biogas are inferior to those of NG, which is almost all CH₄, resulting in reduced burning velocity, lower flame temperature and narrower range of flame stability [9, 10].

For the aforementioned shortcomings, the flame characteristics of biogas have been studied and some methods to avoid blowout, lifting and yellow tipping were proposed. Unlike other fuel gases, flashback is not prone to occur in biogas flame due to decreased burning velocity, but lifting and blowout are the most concerned conditions of flame instability [11]. With an increase of the primary air ratio, the lifting limits decreased, while the yellow tipping
porous media combustion allows operation in ultra-lean combustion, temperature uniformity and low pollutant formation mode could contribute to fuel consumption reduction, stability of operation; for instance, adopting new techniques such as flameless combustion method of biogas utilization is to alter the way of biogas consumption. While the initial 10% hydrogen addition to the biogas led to a tremendous boost to the blowout limits, the extent of blowout limit enhancement diminished with a continuous increase from 10% to 25%. Another aspect that the blowout limits, the extent of blowout limit enhancement diminished with a continuous increase from 10% to 25%. Another aspect that needs to be considered in the operation of biogas is the wider jet diameters required when biogas is applied to household heating boilers, it is required to blend the biogas with other gases or use flameless techniques, which is an expensive and complicated operation. Biogas cannot be combusted stably in the conventional NG burners for the biogas due to the higher content of CO. Partially premixed burner (PPB) is the dominant part in household gas boiler, but rare studies on the modifications of PPBs have been reported. When biogas is burnt in PPBs, the wider jet diameters are required since the biogas has a lower calorific value—partly because of the lower CH4 content.

This paper focused on modifying an NG-fired burner into a biogas-fired one, using mixed gases of CH4 and CO2 to simulate biogas. The PPB that was applied to gas-fired wall-mounted boilers was studied and the jet diameter and burner port area of this burner were determined to produce the rated heat input (25 kW). In order to optimize the jet diameter, we changed the jet diameters and verified their effects on primary air ratio, heat input and flame shape. The effects of different biogas components on the thermal efficiency and emissions of NOx and carbon monoxide (CO) were investigated as well.

2. DESIGN OF THE BIOGAS BURNER

In China, PPBs are extensively applied to all kinds of domestic gas appliances (gas stoves, gas water heaters, gas-fired wall-mounted boilers, etc.). PPB is also one of the most promising gas appliances operated in the atmosphere and has lots of advantages [20, 21], including significantly reducing pollutant emissions (e.g. CO and NOx), improving flame temperature and shortening flame length. As shown in Figure 1, it is a typical PPB of the gas-fired wall-mounted boiler that consists of 15 bar burners. The intervals between bar burners must be appropriate to ignite easily and obtain secondary air. Once the flammable mixture nearest to the igniter (mid-point position in Figure 1) starts burning, the flame will be rapidly propagated from port to port over the entire burner until a proper flame is anchored and stabilized at each bar burner. Figure 2 shows a bar burner that mainly comprises the ejector device, premixing chamber and burner head. The sizes of the jet diameter of the ejeccor and burner port are the most important parts for PPBs’ design [20], which control the characteristics of the partially premixed flame. Therefore, only modifications to the jet diameter and burner port areas were carried out.

2.1. Jet diameter of the ejector

The core size of an ejector is the jet diameter because it directly influences the amount of primary air, which determines the characteristic of the flame. Generally, it is crucial for a proper operation that the proportion of primary air should be held constantly at all burning states. In the paper, an appropriate jet diameter was provided for the conversion of gas-fired wall-mounted boilers commonly used for NG to these for biogas. Therefore, a series of jet diameters was designed by trial and error at rated pressure of 1600 Pa in the experiments. To verify the correctness of the experiment results, the jet diameters were compared with the ones obtained by calculation using Equation (1), which was extensively applied to define the jet diameter of the PPB for the type of fuel [22]. Biogas was regarded as ideal gas for its relatively low pressure (<2000 Pa) in calculation.

\[
d_0 = \left(\frac{1}{0.0036 \mu}\right)^\frac{1}{2} \left(\frac{S}{P_g}\right)^\frac{1}{3},
\]

where \(d_0\) is the jet diameter (mm), \(l\) is the gas flow rate of single jet of ejector (m’/h), \(\mu\) is the coefficient of discharge for the jet, \(s\) is the relative density of gas and \(P_g\) is the gas pressure at the jet (kPa).

2.2. Burner ports

In order to obtain stable flame and uniform thermal intensity of burner port (q), lifting problem arising from the combustion of biogas must be taken into account in designing the burner port.
The way of avoiding lifting is to choose a bigger burner port than a principal one. Additionally, the distance between burner ports determines the characteristics of the ignition and flame. The strong interactions between the flame of burner ports can promote flame stability.

Figure 2b and c shows the burner port area that was optimized to prevent flame lifting due to lower burning rate when utilizing biogas in the conventional burner. As shown in Figure 2b, it is noteworthy that there are five additional burner ports of 2.0-mm diameter drilled on either side of every bar burner, and the total areas (A) of burner ports are increased to 4744.5 mm$^2$. The additional burner ports decrease the pressure of gas mixture inside the bar burner, which results in a decreased exit velocity of the gas mixture. These ports make the ignition more continuous and flame retention more effectively [20]. Therefore, the thermal intensity of modified burner port is 5.28 W/mm$^2$ calculated by Equation (2). This value is laid in the middle of stable flame zone as plotted by Dai [12]. This value also meets the design requirement of the biogas burner [22], which makes the combustion state preferably being within the range of stability region.

$$q = \frac{Q}{A}, \quad (2)$$

where $q$ is thermal intensity (kW/mm$^2$), $Q$ is heat input of gas-fired wall-mounted boiler (kW) and $A$ is total burner port area (mm$^2$).

3. EXPERIMENT AND METHODOLOGY

3.1. Experimental biogas in the tests
In the experiment of optimizing the jet diameter, biogas was prepared by blending pure CH$_4$ (60% by volume) and CO$_2$ (40% by volume) to simulate the biogas supplied typically in practice. To investigate the influence of different biogas components on emissions and thermal efficiency of the biogas-fired wall-mounted boiler, five groups of the gas mixture with different properties are shown in Table 1. Component experiments were conducted on the jet diameters of 2.0 mm. Pure gases of CH$_4$ and CO$_2$ were taken from corresponding high-pressure cylinders. All gases were of commercial grade with over 99% purity to avoid any change of the mixture composition during the experiments.

### Table 1. Components and properties of biogas.

| Group | CH$_4$ (%) | CO$_2$ (%) | $H_2$ (MJ/Nm$^3$) | Relative density |
|-------|------------|------------|-------------------|-----------------|
| 1     | 40         | 60         | 14.361            | 1.393           |
| 2     | 50         | 50         | 17.951            | 1.042           |
| 3     | 60         | 40         | 21.541            | 0.945           |
| 4     | 70         | 30         | 25.131            | 0.847           |
| 5     | 80         | 20         | 28.722            | 0.750           |

3.2. Experimental apparatus
Figure 3 schematically shows the gas blending system and the experimental apparatus for testing PBPBs. NG and CO$_2$ were controlled by the regulating needle valves. After the flow rates were measured by gas flow metres, the gases entered a wet storage tank alternately. When the gases in the tank reached the expectation, they were stirred by the stirrer for at least 30 minutes to make the components mixed. A U-gauge was used to measure the biogas pressure before PBPBs. Finally, biogas and primary air were mixed evenly in the premixing chamber of PBPBs and the mixture supplied to the burner head. Additionally, there was a small hole drilled on the premixing chamber, where a gas sampling probe would be inserted in. The probe was connected to a 1-m stainless steel pipe through which the gas samples were delivered to a gas chromatography, and the O$_2$, CO$_2$, CH$_4$ and N$_2$ were analysed.

The measuring range and accuracy of the instruments used in the experiment are listed in Table 2. The test platform of the whole experiments was set up, as shown in Figure 4.

3.3. Experimental procedure
The experiment aimed to investigate the impacts of various jet diameters and biogas composition on the primary air ratios, heat inputs and flame shapes. The operational parameters for the current experiments were listed in Table 3. Biogas pressure was typically 1600 Pa and the burner port area was 4744.5 mm$^2$. During the experiments, the designed operating condition was achieved by regulating the jet diameter of the ejector. When the
flame temperature remained stable, it reached the thermal/chemical equilibrium. Then, measurements were carried out at each test individually.

4. RESULTS AND DISCUSSION

4.1. Experiments of the jet diameter

The jet diameter is one of the most important design parameters of PPBs, and it has great influences on the heat input, primary air ratio and flame shape. By comparing the corresponding heat input, primary air ratio as well as flame shape for different jet diameters listed in Table 3, the optimal jet diameter can be obtained. Additionally, the correctness of Equation (1) can be also properly validated.

4.1.1. Effects of jet diameter on heat input

Heat input is the essential assessment criterion for most of domestic gas appliances, and it reveals the ability of heat release of a burner. The jet diameter has a significant impact on gas flow rate that affects the heat input. Generally, the predicted heat input ($Q_0$) at standard state (101.325 kPa, 288.15 K) can be calculated from the Equation (3) derived from Equation (1). In fact, under the restriction of experimental conditions, the experiment process cannot be conducted in standard state, so Equation (4) can be obtained after correction.

$$Q_0 = \frac{L_0 H_1}{3600} = \frac{15 \times 0.0036 \mu d_0^2 H_1 \left( \frac{P_g}{P_o} \right)^{1/2}}{3600},$$

$$Q_1 = \frac{L_1 H_1}{3600} \times \frac{P_g + P_a}{P_0} \times \frac{T_0}{T_1}. \tag{4}$$

where $Q_0$ and $Q_1$ is heat input at standard state (kW) and actual heat input (kW), respectively; $H_1$ is lower heating value (MJ/m$^3$); $P_g$, $P_a$ and $P_0$ are biogas pressure (kPa), atmospheric pressure (kPa) and standard atmospheric pressure (101.325 kPa), respectively; $T_0$ and $T_1$ are standard temperature (288.15 K) and actual biogas temperature (K), respectively; and $L_0$ and $L_1$ is theoretical and actual biogas flow rate (m$^3$/h), respectively. Since there are 15 bar burners, the total gas flow rate should be 15 times the single flow rate.

The comparison of measured and calculated heat inputs is shown in Figure 5. The predicted results coincided with the experimental data well, indicating that Equation (3) could calculate the heat input relatively accurately. The reason why there was relative deviation between the predicted and measured heat inputs might be the imprecision of jet processing or the interference of some experimental apparatus. Furthermore, the heat input increased with the increase of jet diameter because the fuel flow rate rose along with the increase of the jet diameter. When the jet diameters were bigger than 1.9 mm, the heat input could meet the design requirement (25 kW).

4.1.2. Effects of jet diameter on primary air ratio

The primary air ratio ($\alpha'$) is a crucial design parameter for PPBs. The amount of primary air added to the gas before the flame varies with the design of burner, but it is usually around 50–75% of the
Table 3. Operational parameters for the experiment.

| Case | Biogas pressure (Pa) | Total burner port area (mm²) | Jet diameter (mm) |
|------|---------------------|-----------------------------|------------------|
|      | 1600                | 4744.5                      | 1.5              |
|      |                     |                             | 1.6              |
|      |                     |                             | 1.7              |
|      |                     |                             | 1.8              |
|      |                     |                             | 1.9              |
|      |                     |                             | 2.0              |
|      |                     |                             | 2.1              |
|      |                     |                             | 2.2              |
|      |                     |                             | 2.3              |

Figure 5. The comparisons between the computed and measured heat input for different jet diameters.

Figure 6. Comparison between calculated and measured primary air ratio (α′) under different jet diameters.

total air requirement [23]. The low value of α′ means that the gas needs to be supplied more secondary air for combustion and results in the long flames and burning imperfectly. Meanwhile, poor combustion generates poisonous CO and carbon particles that show up as yellow flashes in the flame. The high value of α′ explains that the gas can burn under the primary air, so the flames are much shorter. However, larger primary aeration is inadvisable, as the flashback can occur. The predicted primary air ratio can be calculated by Equation (5), entrainment ratio is given by Priggs [22] in Equation (6). In order to validate the predicted α′ value calculated by Equation (5), the measured α′ value can be obtained by Equation (7) based on the data from the experiment. The corresponding experimental results are shown in Figure 6.

\[
\alpha' = \frac{rs}{V_0}. \tag{5}
\]

\[
r = \sqrt{s \left( \frac{d_t}{d_0} - 1 \right)}. \tag{6}
\]

\[
\alpha' = \frac{(21 - O_2)}{(O_2 V_0)}, \tag{7}
\]

where α′ is primary air ratio; r is entrainment ratio; V_0 is theoretical air volume required (m³/h); d_0 and d_t is jet diameter (mm) and throat diameter (mm), respectively; and O_2 is O_2 molar fraction in gas (%). In addition, 21 is assumed to be the 21% O_2 of air.

The primary air ratio was inversely proportional to jet diameter—primary air ratio increased with decreasing jet diameter, as shown in Figure 6. The narrower jet diameter decreases the biogas flow rate at the jet, causing that the less amount of primary air will be injected. Besides, the air required by biogas combustion also decreases with biogas flow rate reduction. The total air requirement is not proportional to the primary air entrainment and changes more than primary air, so the primary air ratio increased with narrower jet diameter. Figure 6 indicated that the predicted values calculated by Equation (5) were basically accurate and in consistency with the experimental data, especially when the jet diameter was between 1.7 and 2.1 mm. As mentioned previously, when the value of α′ was between 0.5 and 0.75, the partly premixed flame was relatively stabilized. Hence, the values of jet diameter ranging from 1.8 to 2.2 mm could meet the design requirement of PPBs.

4.1.3. Effects of jet diameter on flame shape

Stable flame means that no flashback, yellow tipping and lifting are observed. The flame becomes stabilized and attaches to the burner rim when the burning velocity and flow speed of the flame are matched with each other. For a PPB, it usually runs under small excess of air to avoid the danger caused by rich flame. If too much air is supplied, the flame cools off, thus prolonging the working time and increasing the fuel demand.

In the experiments, flame shapes in different jet diameters were observed under rated conditions where the biogas supply pressure was maintained at 1600 Pa and two components of pure
biogas were 60% CH$_4$ and 40% CO$_2$. The flame heights increased with the increase of the jet diameters and the colour of ‘outer mantle’ in the flame became yellower. It can be interpreted that the hydrocarbon in the flame is converted into aldehyde and ethyl alcohol inside the reaction zone of flame because the primary air ratio decreases with increasing jet diameter especially when the primary air ratio remains higher than 0.7 ($\alpha' > 0.7$). So, the outer mantle is nearly transparent (Figure 7a–c). On the contrary, when the primary air ratio remains lower than 0.7 ($\alpha' < 0.7$), the hydrocarbon is converted into carbon particles inside the reaction zone because of lower temperature. The outer flame would become bright (Figure 7d–i). Lower primary air ratio causes the shortage of air required for biogas burning, resulting in forming richer and longer flames, which damage the chamber walls and increase emissions. Particularly, when the jet diameter was more than 2.1 mm, the yellow tipping was fairly distinct, as shown in Figure 7g–i. If the jet diameter was <1.9 mm, the heat input cannot satisfy the design requirement (25 kW). Therefore, the optional jet diameter was between 1.9 mm and 2.1 mm.

4.2. Performance experiments of biogas-fired wall-mounted boiler with the designed burner

The composition of biogas varies with the change of external conditions, so the biogas-fired wall-mounted boiler is required to have a good adaptability to different composition of biogas. As biogas composition changes, the combustion characteristics of PPBs will change. Therefore, the PPB was installed in a biogas-fired wall-mounted boiler and tested its exhaust emissions and thermal efficiency. The burner ($d_0 = 2.0$ mm) was chosen as the test burner and the primary air ratio was 1.7 based on the recommendation above.

4.2.1. Emissions

CO and NO$_x$ are often regarded as the important indices of pollutant emissions. The effects of different biogas composition on the variation of CO and NO$_x$ emissions were investigated in this section. According to Equations (8) and (9), the converted CO and NO$_x$ contents can be calculated from the original data to the data where the excess air coefficient ($\alpha$) is equal to 1 and the calculation were presented in Figure 8.

$$CO = CO' \times \frac{21}{21 - O_2},$$  \hspace{1cm} (8)

$$NO_x = NO'_x \times \frac{21}{21 - O_2},$$  \hspace{1cm} (9)

where CO and NO$_x$ are CO content and NO$_x$ content (ppm) when $\alpha$ is 1, respectively; $CO'$ and $NO'_x$ are measured CO content and measured NO$_x$ content (ppm) in the experiment; and $O_2$ is O$_2$. 

Figure 7. Flame status under different jet diameters.
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4.2.2. Thermal efficiency

It was illustrated in Figure 9 that the thermal efficiency of the biogas-fired wall-mounted boiler decreased with the increasing of CH4. When CH4 in biogas was higher than 60%, the thermal efficiency changed dramatically and was lower than the standard value (84%). When the size of the burner remains unchanged, the CH4 increases and the primary air ratio decreases, resulting in poor combustion condition and increased heat loss from chemical incomplete combustion, thereby reducing the thermal efficiency.

5. CONCLUSION

The main aim of the research was to investigate the effects of jet diameter on heat input, primary air ratio and flame shape, as well as the influence of different biogas composition on emission and thermal efficiency. From theoretical analysis, the appropriate burner port area was defined as 4744.5mm² and the jet diameters were set up on this basis to conduct the experiments. The major conclusions are as follows.

- The optimal jet diameter experiments were carried out with the mixed gas of 60% CH4 and 40% CO2 as the standard gas. In order to meet the design requirement of heat input.
(25 kW), the jet diameter should be more than 1.9 mm. When the jet diameter ranged from 1.8 mm to 2.2 mm, the primary air ratio \( (\alpha') \) satisfied the design requirement of ejecting ability. However, when the jet diameter was more than 2.1 mm, the combustion characteristic became worse. Generally, the heat input should have a little margin, therefore the optimized jet diameters should be 2.0 or 2.1 mm. And these results coincided with calculated values fairly well.

- From the whole experiments of the biogas-fired wall-mounted boiler, it was concluded that the content of CO met the national standard when CH\(_4\) percentage was within 40–60%. However, with the increase of CH\(_4\) content, the thermal efficiency of the biogas-fired wall-mounted boiler fluctuated greatly.

CONFLICTS OF INTEREST

The authors declare no conflict of interest.

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