Can hydrogen enriched biogas be used as domestic fuel? - Part I – Thermal Characteristics of Blended Biogas/H₂ Impinging Flames

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ABSTRACT

Biogas is a renewable energy source widely produced by breakdowns of organic matters in natural environment and industry. However, it is not yet an ideal replacement of fossil fuels because its high CO₂ content would deteriorate its thermal performance. To upgrade biogas for possible domestic application, hydrogen enrichment is proposed by adding high-grade hydrogen (H₂) to biogas in order to improve its flammability and heating value, and reduce pollutant emission. However, most previous studies on blended Biogas/H₂ focus on analysing the effects of H₂ fraction and nozzle-to-plate distance on the heat flux profile and flame temperature. No comprehensive study has ever demonstrated the influence of the Reynolds number and equivalence ratio under a wide operating range. In this study, a test rig was built to investigate the effects of the Reynolds number and equivalence ratio on heat flux and thermal efficiency of blended biogas/H₂ impinging flame. The blended biogas/H₂ consisted of 80% biogas and 20% H₂ addition in volume. Biogas was artificially made by 60% CH₄ and 40% CO₂ (BG₆₀). The Reynolds number ranges from 300 to 1500 and equivalence ratio ranges from 1 to 3. A comparative study was also conducted between pure biogas (BG₆₀) and biogas with 20% H₂ enrichment.

KEYWORDS Blended biogas/H₂; thermal characteristic; heat flux; thermal efficiency; Reynolds number; equivalence ratio

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1. Introduction

Great efforts are being made to search for alternative energy sources to traditional fossil fuels under the rising energy crisis in recent decades (Barreto, 2018). Biogas is a promising fuel as it is renewable and widely available in natural environment and industry (sewage treatment, landfill) (Weiland, 2010). Biogas, mainly consists of 40% to 70% of methane (CH₄) and 30% to 60% of carbon dioxide (CO₂), is an excellent hydrocarbon fuel and highly combustible. However, the largest obstacle to its domestic application is the high CO₂ content which deteriorates its thermal performance, including stability, flammability and heating value (Angelidaki et al., 2018). To upgrade the thermal performance of biogas for the possible domestic applications, many attempts have been made by adding a higher-grade fuel to biogas. Mixing liquefied petroleum gas (LPG) and biogas is one commonly used solution. Higher heating value can be found in the mixture compared with pure biogas (Dai et al., 2011). An addition of propane and hydrogen to biogas is proven to increase its burning velocity (Cardona and Amell, 2013).

Increasing attention has been paid in recent decades to upgrading biogas by hydrogen enrichment, i.e. adding high-grade hydrogen (H₂) to biogas in order to improve its flammability and heating value, and reduce pollutant emission (Chen and Zheng, 2011). As a clean and high calorific gaseous energy source, the hydrogen itself has received intensive research attention. Extensive theoretical and experimental studies were conducted on hydrogen/air flames. Pareja et al (2010) presented both experimental

investigations and numerical simulations on laminar burning velocities of H₂/air mixtures by using the angle method with equivalence ratio ranging from 0.8 to 3.0. A further study was conducted to investigate the stability of H₂/air premixed flames under a wide range of equivalence ratios (Pareja et al., 2011). An opposite trend of burning velocity and pressure relationship can be observed under the fuel-lean condition and fuel-rich condition. Although hydrogen shows superior properties in pollution emission and calorific value, the safety issue is a major concern of researchers because of its high tendency of leakage, wide flammability limits and small ignition energy in air (Butler et al., 2009).

Research studies on thermal characteristics of hydrogen enriched flames (premixed flame and diffusion flame) have been widely reported. Zhen et al (2012) reported a comparative study on H₂/LPG and H₂/CH₄ in terms of pollutant emission and impingement heat transfer characteristics. The results show that, with an increase in H₂, the laminar burning speed, flame temperature and NO_x emission increase accordingly, but the CO emission is reduced. Besides, the longer CH₄ flame appearance than LPG flame's is observed, a wider wall jet layer for integrated heat transfer rate improvement results. Zhen et al (2016) experimentally investigated stoichiometric mixed biogas/H₂ flame. They pointed out that the heat transfer was dominantly affected by flame temperature. Moreover, the longer flame appearance facilitated heat transfer rate only under a large nozzle-to-plate distance. Wei et al (2016a) investigated the influences of equivalence ratio (0.8, 1.0, 1.2), H₂ (10% - 50%) and CO₂ on the heat release

characteristics of premixed laminar biogas/H₂ flame by numerical simulations (Wei et al., 2016a). Wei et al (2016b) also investigated the influences of H₂ enrichment and CO₂ volumetric fractions on heat flux and flame temperature of laminar biogas/H₂ flame. The total heat transfer rate was enhanced by 20% to 30% due to hydrogen addition and the H₂ enrichment fraction was optimised to be 30%.

Though different important areas of blended/biogas/H₂ thermal performances have been studied, most of them focus on analysing the effects of H₂ fraction and nozzle-to-plate distance of impinging flame. The influences of the Reynolds number and equivalence ratio on heat flux and thermal efficiency under a wide operating range are rarely reported. Therefore, it serves as a motivation to conduct this study to fill the research gap. In this study, a test rig was built to intensively investigate how the Reynolds number (from 300 to 1500) and equivalence ratio (from 1 to 3) affected the heat flux and thermal efficiency of blended biogas/H₂ impinging flame. A study was also conducted to compare pure biogas (BG60) and biogas with 20% H₂ enrichment.

2. Methodology

2.1. Experimental test rig

The experimental test rig consisted of three high-pressure air cylinders (CH₄, CO₂ and H₂), an air compressor, a Bunsen burner, an impingement copper plate, and a data acquisition system, as shown in Figure 1. CH₄, CO₂ and H₂ were supplied from respective cylinders and air supplied by the air compressor. The nozzle type burner was a cylindrical tube with outer diameter of 11.11 mm, inner diameter of 8.93 mm and length of 250 mm. A screen mesh was used to prevent the surrounding flow from disturbing the flame. The burner length was designed to be long enough to ensure full development of the flow at the nozzle outlet. The burner was placed under the impingement copper plate (600 mm x 600 mm x 12 mm) equipped with a water-cooling system. The 40°C-cooling-water was recirculated by a Julabo fc1600s cooler. A mixing chamber was installed before the burner in order to pre-mix the supplied fuels and air.

The data acquisition system included a temperature sensor for plate temperature measurement and a heat flux sensor was mounted at the centre of the copper plate. The plate temperature was measured by Type B thermocouple with accuracy of $\pm 0.5\%$. The heat flux sensor with dimensions of 2 mm x 2 mm (Vatell Corporation, Model HFM-6D/H) was coated with zynolyte which had emissivity of 0.94. A NIST traceable calibration certificate was provided for the heat flux sensor with the accuracy of $\pm 3\%$. The voltage signal output of the sensor was amplified by an AMP-6 amplifier and recorded by a data acquirer.

Figure 1. Systematic diagram of experimental test rig.

In the experiment, the gas flow rate could be controlled by an individual valve and monitored by flow meters and pressure gauges. The biogas (primary fuel) used in the experiment was artificially made by mixing methane (CH₄) and carbon dioxide (CO₂). Hydrogen (H₂) was added to the biogas as a secondary fuel. The hydrogen content in the mixture is defined as (Hu and Zhang, 2019):

$$\alpha = \frac{V_{H_2}}{V_{CH_4} + V_{CO_2}}, \quad (1)$$

where V_{H_2} , V_{CH_4} and V_{CO_2} are the volumes of H₂, CH₄ and CO₂ respectively. All the fuels are commercial gases with high purity. The methane contained at least 99% of CH₄. The hydrogen contained 99.999% H₂, less than 3 ppm moisture and less than 5 ppm O₂. The carbon dioxide consisted of at least 99.9% CO₂, less than 1 ppm CO, 10 ppm moisture and 5 ppm oil.

The Reynolds number is one of the key parameters affecting the thermal characteristic of flame (Baigmohammadi et al., 2015), which is calculated based on the fuel-air mixture at room temperature.

$$Re = \frac{\rho_{mix} V_{exit} d}{\mu_{mix}}, \quad (2)$$

where, ρ_{mix} is the density of fuel-air mixture, kg/m³; V_{exit} is the exit flow velocity from the burner, m/s; d is the inner diameter of the nozzle, m; μ_{mix} is the dynamic viscosity of gaseous mixture, N·s/m².

The equivalence ratio (Φ) of fuel/air mixture, another parameter to be studied influencing thermal characteristic of the flame, is defined as:

$$\Phi = \frac{V_{fuel}/V_{air}}{(V_{fuel}/V_{air})_{st}}, \quad (3)$$

where, V_{fuel} and V_{air} are the volumetric flow rates of fuel and air respectively. $(V_{fuel}/V_{air})_{st}$ is the flow rate ratio under stoichiometric condition.

The thermal characteristic of blended biogas/ H_2 flames was investigated for heat flux and thermal efficiency. The heat flux distribution along the radial direction away from the stagnation point was measured by the heat flux sensor. The thermal efficiency was evaluated by boiling water in stable operating condition. An aluminium pot with diameter of 220 mm, placed above the centre of the pot, was used for boiling 1 litre of water. A Type B thermocouple was immersed into water for water temperature measurement. Recording the time duration was required to heat the water from 30°C to 50°C by a timer. During the heating process, stirring was essential to maintaining even temperature distribution inside the boiler. Thermal efficiency, representing the overall thermal energy utilisation of biogas/ H_2 flames, is calculated as (Hou and Ko, 2004):

$$\eta_{th} = \frac{mc_p(t_2 - t_1)}{V_c \times LHV} \times 100\%. \quad (4)$$

where m is the water mass in the boiler, kg; c_p is the specific heat of water, kJ/kg·°C; t_1 and t_2 are the initial water temperature and final water temperature in the boiler respectively, °C; V_c is the volume of fuel burned, Nm³; LHV is the lower heating value of the fuel, kJ/Nm³.

2.2. Parameters setting

To comprehensively investigate the effects of the Reynolds number, equivalence ratio, and hydrogen addition on the heat flux and thermal efficiency of blended biogas/ H_2 flame, the experiments were conducted under a wide range of Reynolds numbers from 300 to 1500 (laminar flow) with an increment of 200 and equivalence ratios from stoichiometry condition $\Phi = 1$ to $\Phi = 3$ with an interval of 0.5. The biogas used in the experiment consisted of 60% CH_4 and 40% CO_2 (BG₆₀). The hydrogen addition was set constant at 20% H_2 addition and 80% BG₆₀ in blended biogas/ H_2 . The nozzle-to-plate distance ratio (H/D) was fixed at 5, considering the largest heat flux at the stagnation point under the middle values of both equivalence ratio and the Reynolds number within the experimental ranges (i.e. $\Phi = 1.5$ and $Re = 750$). The effect of hydrogen addition was evaluated by comparing the heat flux and thermal efficiency between BG₆₀ and BG₆₀ with 20% H_2 addition.

3. Results and discussion

The influences of equivalence ratio (Φ), the Reynolds number (Re) and H_2 enrichment of impinging blended biogas/ H_2 flame on heat flux and thermal efficiency were comprehensively analysed as follows.

3.1. Effects of equivalence ratio

Figure 2 shows the effects of equivalence ratio on heat flux distribution of blended biogas/ H_2 impinging flame under different Reynolds numbers. The heat flux generally increases with the growth of equivalence ratio at the stagnation point under $Re = 300$. The heat flux increased from 72.6 kW/m² to 86.3 kW/m² when Φ increased from 1 to 2.5. Then the stagnation heat flux declined to 75.9 kW/m² as Φ further increased to 3. The reason can be attributed to the change of flame height. Under $Re = 300$, open flame formed at $\Phi = 1$ to 2 while the flame touched the impingement plate slightly at $\Phi = 2.5$ and further extended radially at $\Phi = 3$. The increasing stagnation heat flux was found when the conical flame front moved closer to the impingement plate when Φ increased from 1 to 2. However, the flame began to stretch radially on the impingement plate with the cool core that remained at the stagnation point. Therefore, the peak heat flux shifted from the stagnation point towards a larger radial distance under $\Phi = 2.5$ and 3 scenarios. In these two cases, the peak heat flux appeared at around 5.8mm radial distance from the stagnation point.

As Re increased to 700, the peak local heat flux was obtained at stagnation point only when $\Phi = 1$ and $\Phi = 1.5$ while it shifted to 12 mm, 24 mm and 24 mm away from the stagnation point when $\Phi = 2$, 2.5 and 3 respectively. The trend that indicated a larger jet area and radial stretching were achieved under higher equivalence ratio. When Re further increased to 1100, the appearance of peak local heat flux at stagnation point only took place when $\Phi = 1$. The peak heat flux shifted radially outward when $\Phi \geq 1.5$. In general, the larger the equivalence ratio, the further the distance between peak heat flux point and stagnation point. At $Re = 1500$, stagnation heat flux was no longer the largest among the radial distance, which is attributed to the longer flame front owing to the increase in the Reynolds number. No data was recorded at $\Phi = 1$ under $Re = 1500$ because of non-stability of blended biogas/ H_2 flame.

Figure 3 shows the effects of equivalence ratio on thermal efficiency of blended biogas/ H_2 flame under different Reynolds numbers. The thermal efficiency ranged from 56% to 80% in wide operating condition. It was found that the thermal efficiency trend affected by the equivalence ratio behaved differently under different Reynolds number ranges. Under low Reynolds number ($Re = 300$, $Re = 500$ and $Re = 700$), thermal efficiency generally increased with the growth of the Reynolds number. At $Re = 300$, the thermal efficiency increased from 56% to 68% when Φ improved from 1 to 2.5. However, under high Reynolds number ($Re = 900$, $Re = 1100$ and $Re = 1300$), thermal efficiency generally decreased with the increase in the Reynolds number. At $Re = 1300$, the thermal efficiency dropped from 80% to 68% when Φ improved from 1 to 3.

The trend can be explained as follows. The flame length was shorter at low Reynolds number. the high temperature conical flame front could be brought closer towards the boiler, achieving higher heat flux contact.

Figure 2. Effects of equivalence ratio on heat flux distribution.

At high Reynolds numbers, however, the flame was long enough for the high temperature conical flame front to collide with the boiler even under the low equivalence ratios. A further increase in equivalence ratio would convert the open flame into impinging flame with the inner cool core pushed upward to the boiler. The inner cool core that consisted of unburned fuels was of lower temperature and led to lower heat flux. The highest thermal efficiency was found at $Re = 1300$ and $\Phi = 1.5$ while the lowest thermal efficiency was found at $Re = 300$ and $\Phi = 1$, indicating that the conical flame front was the closest to the boiler in the former scenario while it was far away from the boiler in the latter scenario.

Figure 3. Effects of equivalence ratio on thermal efficiency.

3.2. Effects of Reynolds number

Figure 4 shows the effects of the Reynolds number on heat flux distribution of blended biogas/ H_2 flame under two equivalence ratios ($\Phi = 1$ and $\Phi = 2.5$). The local heat flux increases with the increase in the Reynolds number not only at the stagnation region but also at the wall jet region. At $\Phi = 1$, the peak heat flux appeared at the stagnation point under all Reynolds numbers. The highest heat flux was 135 kW/m^2 at $Re = 1100$ (stable flame was not available at $Re = 1300$ and $Re = 1500$) and it decreased to 73 kW/m^2 at $Re = 300$. Hargrave et al (1987) point out that heat flux profiles mainly depend on mean velocity and flame temperature, with the maximum heat transfer rate taking place within or close to the flame reaction zone. It explains why the Reynolds number reflects the flame velocity and plays a significant role in the heat flux profile.

Figure 4. Effects of the Reynolds number on heat flux distribution.

In general, the peak heat flux tends to shift towards a larger radial distance at higher Reynolds number. At $\Phi = 2.5$, the peak heat flux of 92 kW/m^2 appeared at the radial distance of 6 mm under $Re = 300$. The heat flux peaked at the radial distance of 18 mm under $Re = 500$ and 24 mm under $Re = 700$. This trend can be explained by the changes of flame structure. At $Re = 300$, the cold core had yet to touch the plate; therefore, the maximum heat flux took place near the central radial distance. When Re increased to 500, the cold core impinged on the plate and extended along the radial direction. Thus, the heat flux at the central radial distance dropped but reached the peak point at 18 mm away from the central location. When the Reynolds number further increased, the diameter of the core at stagnation region increased, making the maximum heat flux shifts outward. However, the peak value shift became less significant when the Reynolds number exceeded 900. All peak heat flux appeared at around 24 mm radial distance under $Re = 900$, $Re = 1100$, $Re = 1300$ and $Re = 1500$.

Figure 5. Effects of Reynolds number on thermal efficiency.

Figure 5 shows the effects of the Reynolds number on thermal efficiency of blended biogas/ H_2 flame under various equivalence ratios. In general, the thermal efficiency increases with the growth of the Reynolds number. The thermal efficiency at $\Phi = 1$ was relatively low especially under low Reynolds numbers ($Re = 300$, $Re = 500$ and $Re = 700$). It is due to the fact that the flame height was too short to collide with the boiler directly under these operating conditions. The thermal energy released by the flame heated the surrounding air first and then the air heated the boiler by

convection and conduction. Thermal efficiency increased from 56% to 68% when the Reynolds number improves from 300 to 1100 at $\Phi = 1$. The trend was similar at $\Phi = 1.5$, but it became less significant when Φ was higher than 2. The thermal efficiency could be improved by 10% on average if the Reynolds number increased from 300 to 1500 at $\Phi = 2$, $\Phi = 2.5$ and $\Phi = 3$.

3.3. Effects of hydrogen addition

Figure 6 shows the effects of hydrogen addition on heat flux at the stagnation point ($\Phi = 1.5$) of blended biogas/ H_2 flame under various Reynolds numbers. The stagnation heat flux of BG_{60} with 20% H_2 addition was higher than that of BG_{60} (pure biogas) under all Reynolds numbers. It is understandable as the heating value of H_2 was much higher than that of biogas. With the Reynolds number increased from 300 to 900, the stagnation heat flux of BG_{60} first increased and peaked at 95 kW/m^2 at $Re = 500$ and then dropped to 18 kW/m^2 at $Re = 900$. The blended biogas/ H_2 shows a similar trend but the stagnation heat flux peaked at $Re = 700$, slightly delays than the peak point observed in BG_{60} at $Re = 500$. The heat flux at stagnation point significantly drops at $Re = 700$ (BG_{60}), and $Re = 900$ (BG_{60} with 20% of H_2) can be attributed to the change of flame appearance, as shown in Figure 7. The flame length increased with the Reynolds number. The flame length of BG_{60} was longer than that of blended biogas/ H_2 under the same Reynolds number due to the faster burning speed of H_2 (Hargrave et al., 1987). At $Re = 700$, the inner cool core of the BG_{60} flame already collided with the plate, resulting in lower heat flux, while the flame of blended biogas/ H_2 had not touched the plate. In summary, by adding 20% H_2 , the stagnation heat flux of impinging biogas/ H_2 flame could be improved by 11% compared with BG_{60} under flame front heating condition.

Figure 6. Effects of hydrogen addition on heat flux at the stagnation point ($\Phi = 1.5$).

Figure 7. Images of pure biogas (0% of H_2) and hydrogen enriched biogas (20% H_2).

Figure 8 shows the effects of hydrogen addition on thermal efficiency of blended biogas/ H_2 flame under various Reynolds numbers. In general, the thermal efficiency decreased by adding H_2 to biogas. The thermal efficiency of BG_{60} ranged from 68% to 79% with an average value of 71.7%, while the thermal efficiency of blended biogas/ H_2 ranged from 63% to 72% with an average value of 67.2% (thermal efficiency decreased by 6.3%). The slight decrease in thermal efficiency may be attributed to the larger heat loss of blended biogas/ H_2 flame. Figure 7 shows the cool core flame height decreased by adding H_2 while the area of high temperature flame front is enlarged. The heat transfer was driven by the temperature difference between the two regions. Therefore, more heat loss would take place between the flame and surround air by convection and radiation of blended biogas/ H_2 flame. In summary, by adding 20% of H_2 to pure biogas will deteriorate thermal energy efficiency slightly by 6.3% on average.

Figure 8. Effects of hydrogen addition on thermal efficiency.

4. Conclusion

The thermal characteristics of blended biogas/ H_2 impinging flames was experimentally investigated. The influences of equivalence ratio (Φ), the Reynolds number

(Re) and H_2 enrichment of impinging blended biogas/ H_2 flame on heat flux and thermal efficiency were analysed. The Re varied from 300 to 1500 within laminar flame region and Φ varied from 1 to 3, whereas the nozzle-to-plate distance was fixed at 5. A constant ratio of 20% H_2 and 80% BG_{60} in volume was used in blended biogas/ H_2 . Some important findings are summarised as follows:

- (1) The flame length increases with the growth of equivalence ratio. Under low Reynolds number ($Re = 300$), the stagnation heat flux generally increases with the equivalence ratio because its short flame height has yet to collide with the plate. Under higher Reynolds numbers, the stagnation heat flux decreases with the equivalence ratio and the peak heat flux shifts away from the central axis because the cool core of the flame impinges on the plate directly.
- (2) The thermal efficiency trend affected by the equivalence ratio behaves differently under different Reynolds number ranges. Under low Reynolds numbers ($Re = 300$, $Re = 500$ and $Re = 700$), thermal efficiency generally increases with the growth of Reynolds number. However, under high Reynolds numbers ($Re = 900$, $Re = 1100$ and $Re = 1300$), thermal efficiency generally decreases with an increase in the Reynolds number.
- (3) Increments of the Reynolds number will result in a larger flame length. With an increase in the Reynolds number, larger local heat flux can be observed not only at the stagnation region but also at the wall jet region. However, the peak heat flux will deviate towards a larger radial distance at higher Reynolds numbers.
- (4) Higher Reynolds number contributes to larger thermal efficiency. The thermal efficiency can be improved by 10% on average if the Reynolds number increases from 300 to 1500 at $\Phi = 2$, $\Phi = 2.5$ and $\Phi = 3$.
- (5) By adding 20% H_2 , the stagnation heat flux of impinging biogas/ H_2 flame increases by 11% compared with BG_{60} under flame front heating condition. By adding H_2 to BG_{60} will deteriorate thermal energy efficiency slightly by 6.3% on average.

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