Burner Geometry Effect on the Stability of a Swirling Premixed Biogas Flame

by

Abdullah Mustafa Al-Abbasi

A Thesis submitted to the Faculty of Graduate Studies of

The University of Manitoba

in partial fulfillment of the requirements of the degree of

MASTER OF SCIENCE

Department of Mechanical Engineering
University of Manitoba
Winnipeg, Manitoba
Canada

Copyrights © 2019 by Abdullah Al-Abbasi
ABSTRACT

Biogas is a renewable fuel which can be used in numerous engineering applications such as heating, combined heat and power (CHP) generation and transportation. Biogas is a low calorific value gaseous fuel with low burning velocity, both of which make the burning and combustion stability of biogas a real challenge. This drawback of biogas combustion, especially under high flow velocity applications, restrains its industrial implementation. The present thesis aims to experimentally investigate the effect of burner exit geometry on the stability of a premixed biogas flame. The test burner consists of a central bluff-body surrounded by an annulus through which a swirling premixed biogas stream discharges into an atmospheric combustion chamber. Two different burner geometries were tested; a 43.5 mm long cylindrical or conical with an angle of 15 degrees, with the bluff-body was either flushed with the exit of the burner or recessed. Swirl strength was varied by changing the swirl generator’s vanes angle (25-degree or 60-degree, which represent a swirl number, $S$, of 0.39 or 1.16, respectively). The biogas surrogate composition is kept constant (75% CH$_4$ and 25% of CO$_2$).

The results showed that attached biogas flames were observed at all test conditions explored in this thesis. The results revealed that the conical burner promotes higher flame divergence than the cylindrical one, and hence becomes in direct contact with the burner inner wall. The experimental results also showed that recessing the center body, when using a cylindrical burner geometry, enhances flame stability. Whereas, the opposite scenario happens when using a conical burner geometry. The effect of burner configuration (cylindrical or conical) on premixed biogas flames stability is found more significant at high swirl number.
ACKNOWLEDGEMENTS

I would first like to express my sincere gratitude to my advisor Prof. Madjid Birouk for his guidance and support. The door to Prof. Madjid Birouk office was always open whenever I run into an issue or had questions regarding my research/writing. His patience and motivation to allow this project to be my own work, however steered me in the right the direction whenever he thought I needed it.

I would like to thank the lab technicians of the Mechanical Engineering Department Sviatoslaw Karnaoukh. In addition, I would like to acknowledge the financial, academic and technical support of the University of Manitoba. Natural Sciences and Engineering Research Council (NSERC) support is highly acknowledged. I would also like to thank my parents for their continuous support through out the hard times to achieve this remarkable stage. Additionally, my brother and sister who stood by my side along this journey.
DEDICATION

I would like to dedicate this project to my family for their continuous support, love and motivation.
# TABLE OF CONTENTS

Chapter 1 INTRODUCTION AND OBJECTIVES

1.1 Introduction, Applications and Challenges: ......................................................... 1

1.2 Motivations and Objectives: .................................................................................. 4

1.3 Thesis Outline ........................................................................................................ 5

Chapter 2 BACKGROUND AND LITERATURE REVIEW

2.1 Combustion challenges of biogas fuels ................................................................... 6

2.2 Combustion challenges of biogas fuel .................................................................... 6

2.3 Premixed Flame Stabilization Techniques ............................................................... 7

  2.3.1 Flame Stabilization ......................................................................................... 7

  2.3.2 Non-blending techniques ............................................................................... 10

  2.3.3 Blending techniques ...................................................................................... 17

2.4 Literature on The Effect of Non-Blended Techniques on Premixed Biogas Flames .. 19

Chapter 3 EXPERIMENTAL SETUP AND TEST CONDITIONS

3.1 Experimental Setup ................................................................................................. 21

  3.1.1 Burner ............................................................................................................ 21

  3.1.2 Measurements Technique ............................................................................ 25

3.2 Test Conditions ...................................................................................................... 29

Chapter 4 RESULTS AND DISCUSSION

4.1 Introduction ............................................................................................................. 32

4.2 Flame Stability Limits Map .................................................................................. 32

4.3 PIV Measurements and Discussion of Biogas Flame Stability Limits ................. 34
4.3.1 PIV Measurement of Flow Field Characteristics ........................................................... 34

4.3.2 Effect of Flow Rate (Reynolds Number) and Heat Release (Equivalence Ratio) on recirculation zone ......................................................................................................................... 41

Chapter 5 CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK .................. 45
5.1 Concluding remarks ........................................................................................................... 45

5.2 Recommendations for future work .................................................................................... 47

REFERENCE ................................................................................................................................. 49

Appendix A Isothermal Flow Field of Various Flow Rates and Swirl Strength ......................... 65
Appendix B Theoretical and Calculated Swirl Strength ............................................................. 73
List of Figures

Figure 2.1. Potential effects of combustion instabilities in a diffusion jet combustor: a) combustor after combustion instabilities. b) new burner assembly [52] ................................................................. 10

Figure 2.2. Schematic of bluff-body for flame stabilization [88] ................................................................. 11

Figure 2.3. Streamlines and flow structure of a typical high swirling flow [50]. ............................................. 15

Figure 2.4. Multi-hole nozzle employed in [79]. .......................................................................................... 17

Figure 3.1 Schematic diagram of the experimental setup. (A) Gas cylinder. (B) Valves. (C) Regulator. (D) Needle valve. (E) Matheson and Brooks flowmeters. (F) CH4-CO2 gas manifold. (G) Air flowmeter. (H) CH4-CO2-Air manifold. (I) Bypass line. (J) Seeder. (K) Distribution manifold. (L) Burner. (M) Exhaust. (N) Compressed air ............................................................................................................. 23

Figure 3.2 Schematic of the burner ............................................................................................................. 24

Figure 3.3. Axial swirl generator ................................................................................................................. 25

Figure 3.4. Schematic diagram of PIV setup [78] ......................................................................................... 27

Figure 3.5 Schematic diagram of LDV setup [82] ....................................................................................... 29

Figure 3.6. Schematic of burner exit geometry (a) Reference case (Ref); b) tube extension with center body (Ext-CB); c) cone extension with center body (Cone-CB); d) tube extension with no center body (Ext-recess); and e) cone extension with no center body (Cone-recess) ......................................................................................... 31

Figure 4.1. Stability map (Φ – Re) of a premixed biogas flame for (a) S=0.39 and (b) S=1.16 .............. 34

Figure 4.2. Mean velocity contour maps and streamlines of (a-e) isothermal flow at 500 LPM and S=0.39, and (f-j) reacting flow at 450 LPM and S=0.39, ......................................................................................... 37

Figure 4.3. The corresponding biogas flame images of reacting flow cases at 450 LPM and S=0.39 for a) Reference case (Ref); b) tube extension with center body (Ext-CB); c) cone extension with center body (Cone-CB); d) tube extension with no center body (Ext-recess); and e) cone extension with no center body (Cone-recess). ......................................................................................... 37

Figure 4.4. Flow mean velocity vectors (a) velocity field vectors and b) zoomed section of the central region) ................................................................................................................................. 38

Figure 4.5. Mean velocity contour maps of (a-e) isothermal turbulent kinetic energy cases at 500 LPM and S=0.39, and (f-j) reacting turbulent kinetic energy cases at 450 LPM and S=0.39 .................. 40

Figure 4.6a. diagram of equivalence ratio vs. Re number at S=0.39 for different burner configurations (Cone-CB (Case III) & Ext-recess (Case IV)) ......................................................................................... 42
Figure 4.6b. Mean velocity vectors and the corresponding streamlines of the different burner configurations (Cone-CB (Case III) & Ext-recess (Case IV)) represented by points 1 to 5 in diagram of Figure 4.6a. ................................................................. 44

List of Tables

Table 2.1: Potential hazards associated with combustion instability [51] ............................................. 9

Table 3.1: Test conditions .................................................................................................................. 30
NOMENCLATURE

English Symbols:

GHG
Green House Gases

CO$_2$
Carbon dioxide

CH$_4$
Methane

CHP
Combined heat and power

CO
Carbon monoxide

NOx
Nitric oxide and nitrogen dioxide

S
Swirl number

ṁ
mass flow rates

O$_2$
Oxygen gas

D
Nozzle diameter

N$_2$
Nitrogen gas

H$_2$
Hydrogen gas

$u'$
The rms fluctuation velocities in the axial direction

$v'$
The rms fluctuation velocities in the radial direction
\( q' \)  
\text{Turbulent Kinetic Energy}

\( Da \)  
\text{Damkohler number}

\( t_{flow} \)  
\text{Characteristic flow timescale}

\( t_{chem} \)  
\text{Characteristic chemical kinetic timescale}

\( S_L \)  
\text{Laminar flame speed}

\( g_v \)  
\text{Velocity gradient}

\( d_q \)  
\text{Quenching distance}

\( U_{blowoff} \)  
\text{Blowoff velocity}

\( P \)  
\text{Pressure of the gas}

\( d \)  
\text{Diameter of bluff-body}

\( G_{ang} \)  
\text{The axial flux of the circumferential momentum}

\( G_x \)  
\text{The axial flux of the axial momentum}

\( U \)  
\text{Axial velocity}

\( W \)  
\text{Tangential velocity}

\( \text{N1, N5, and N6} \)  
\text{Single-hole fuel nozzles}

\( \text{N2, N3, N4} \)  
\text{Multi-hole fuel nozzles}

\( \text{OH} \)  
\text{OH radical}
K  Kalvin

$S_T$  Turbulent flame speed

$\text{TiO}_2$  Titanium dioxide

**Subscripts and abbreviations:**

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>LCVG</td>
<td>Low calorific value gas</td>
</tr>
<tr>
<td>LSB</td>
<td>Low Swirl Burner</td>
</tr>
<tr>
<td>LBO</td>
<td>Lean blowout limits</td>
</tr>
<tr>
<td>STP</td>
<td>Temperature and Pressure</td>
</tr>
<tr>
<td>IRZ</td>
<td>Internal Recirculation Zone</td>
</tr>
<tr>
<td>CRZ</td>
<td>Corner Recirculation Zone</td>
</tr>
<tr>
<td>PSIG</td>
<td>Pound per Square Inch Gauge (pressure unit)</td>
</tr>
<tr>
<td>LPM</td>
<td>Litter per minute (flow rate unit)</td>
</tr>
<tr>
<td>Hz</td>
<td>Hertz (frequency unit)</td>
</tr>
<tr>
<td>PIV</td>
<td>Particle Image Velocimetry</td>
</tr>
<tr>
<td>m/s</td>
<td>Meter per second (velocity unit)</td>
</tr>
<tr>
<td>mm</td>
<td>Milimiter</td>
</tr>
<tr>
<td>mJ</td>
<td>Milijoule (energy unit)</td>
</tr>
<tr>
<td>Symbol</td>
<td>Definition</td>
</tr>
<tr>
<td>--------</td>
<td>------------</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>CIVB</td>
<td>Combustion-Induced Vortex Breakdown</td>
</tr>
<tr>
<td>PVC</td>
<td>Precessing vortex core</td>
</tr>
</tbody>
</table>

**Greek symbol:**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>β</td>
<td>Discharge angle</td>
</tr>
<tr>
<td>Φ</td>
<td>Equivalence ratio</td>
</tr>
<tr>
<td>Da</td>
<td>Damkohler number</td>
</tr>
<tr>
<td>α</td>
<td>Thermal diffusivity</td>
</tr>
<tr>
<td>α₁</td>
<td>Turbulent thermal diffusivity</td>
</tr>
<tr>
<td>ρₘ and ρᵤ</td>
<td>The densities of the burned and unburned gas</td>
</tr>
</tbody>
</table>
Chapter 1

INTRODUCTION AND OBJECTIVES

1.1 Introduction, Applications and Challenges:

The demand for high energy coupled with concerns over environmental issues have motivated researchers to develop renewable and sustainable sources of energy. Biogas is a promising renewable, feasible and biodegradable source of energy which can replace fossil fuels in heating systems, cooking (stove) and stationary gas turbines [1,5]. Biogas can be produced through organic wastes treatment of agricultural products, domestic sewage and landfills, rendering it an environmentally friendly approach of producing biofuels [1]. This approach lowers Greenhouse Gases (GHGs) emissions, reduces the release of carbon dioxide (CO\textsubscript{2}) and averts emissions of methane (CH\textsubscript{4}) to the atmosphere, complying with the national and international emissions regulations for reducing CO\textsubscript{2} and other GHGs [1,5]. Biogas is composed of up to 80\% CH\textsubscript{4} and 20-60\% CO\textsubscript{2} with small amounts of other substances such as hydrogen sulphide, moisture and nitrogen [2–4].

Biogas can be potentially used in various practical applications such as heating, combined heat and power (CHP) generation and transportation fuel (when upgraded to biomethane) [5–7]. The utilization of biogas has been classified into two categories depending on methane concentration [6]. Firstly, desulphurization dehydration process yields biogas (60-70\% CH\textsubscript{4}) which can be used in boilers, stationary gas turbines, CHP and fuel cells [6]. Secondly, upgrading biogas to
biomethane (97-99% CH₄) allows it to be either compressed, or liquefied, for vehicles and consumer’s applications or utilized in natural gas pipeline system, as well as gas turbines, boilers and CHP [6]. However, combustion challenges resulting from low heating value, low flame temperature and low flame stability of biogas restrain its direct industrial implementation [8-9]. Low biogas flame stability emerges from low flame propagation velocity and high gas velocity required to achieve high energy release rate. Moreover, the low volumetric energy content of biogas leads to combustion instabilities and flame blowouts [9-11]. Chomiak [12] reviewed the problems of LCVG, and reported that flammability limits are a strong function of temperature which depends on the concentration and composition of the fuel. Very lean mixture or insufficient chemical reaction time can lead to blowout [13-14]. The stage of developing renewable fuels to replace fossil fuels while using existing on-site gas turbines is still in progress and remains a challenge [15, 35]. Thus, further research is needed to better understand biogas combustion and consequently develop strategies to improve flame stability [35].

In practical applications, diffusion flames are known for their safety and control over energy release rate [16] as well as their stability over a wide range of operating parameters [17]. However, relatively higher pollutant emissions, such as NOx and CO, of diffusion biogas flames compared to their counterparts’ premixed biogas flames made them a less attractive approach for biogas combustion [16,18]. In order to minimize pollutants emissions, industrial systems operating in lean premixed flames conditions are close to extinction [19-21]. Such operation conditions are effective in decreasing fuel consumption [22, 35] due to enhanced mixing of fuel/air [17]. However, several drawbacks severely affect the stability of lean premixed flames such as autoignition, flashback, increase of CO emissions and combustion instabilities [22]. According to [23] the low flame
temperature of lean combustion associated with small heat release intensity is one of the main causes of the aforementioned drawbacks. Consequently, one non-blending technique approach to alleviate stability concerns can be achieved by implementing a swirl [5,7,14,15,24-27], altering burner geometry [27-30] or/and using a confinement [16,31-34]. In this approach, premixed biogas stability can be enhanced via controlling the flow characteristics. It was reported that increasing the inlet temperature and swirl number can reduce the lean blowout equivalence ratio for a specific swirl number and inlet temperature [5]. Lean blowout limits can be enhanced with increasing swirl number due to an increase in flame stretch [5]. Several studies (e.g., [6, 7]) investigated the stability of premixed biogas flames (up to 30% CO2) using a gas turbine type burner [6, 7]. These studies investigated mainly the effect of burner geometry using a center body (bluff body) and reported an enhanced internal recirculation zone as swirl number increased, which led to an enhancement in biogas flame stability. Another study attributed the improvement of biogas flame stability to the reduction of reagents turbulent mixing time as swirl number increased from 0.69 to 1.35 [8]. It was reported that increasing the quarl angle (confinement) and swirl number significantly influence the flame liftoff, flame topology, blowout and stability of premixed flames [9]. The reason for that is the reduction in the velocity and increase in mixing layer thickness, which make the flow velocity relatively comparable to the flame propagation speed [10]. The structure and stabilization of low calorific value gaseous fuels in conical burner was also studied in [11] where it was reported that the flame stability improved due to vorticity structure inside the cone [12]. However, excessive cone (quarl) angle allowed more ambient air entrainment which led to elongate and narrow the stabilization core [13]. Another approach of enhancing the biogas flame stability can be achieved by blending (blending techniques) with higher calorific values fuels, such as hydrogen (e.g., [14]). This approach can produce higher heating value, laminar flame speed and
heat release, and consequently expands stability limits and allows leaner flames compared to pure biogas flames [14]. However, there exist several issues associated with blending techniques, such as increased NOx emissions, high tendency of flashback, storage risks, and capital and operational costs [12, 15]. The present study/thesis investigates the effect of burner geometry and swirl strength on the stability of non-blended premixed biogas flame with more focus on the lean blowoff limits. These approaches will be discussed in further details in the next chapter, literature review.

1.2 Motivations and Objectives:

The effect of non-blending techniques, such as swirl, nozzle geometry and confinement, on the stability of non-premixed biogas flame has been widely investigated. However, the impact of these techniques on premixed biogas flames is less abundant compared to diffusion flames. Recent studies (e.g., [36-43]) focused on enhancing the stability of premixed biogas flames by blending it with higher calorific value gases, while attempts to stabilize premixed biogas flames via non-blending techniques remain a challenge [35,44]. Therefore, further research using non-blending techniques is needed in order to help develop premixed biogas combustion technology.

The present study aims to experimentally examine the effect of different burner configurations (such as changing the outer tube geometry, swirl number and swirl generator position) on the flow field and stability of premixed biogas flame. However, only a single composition of biogas surrogate (75% CH₄ and 25% CO₂) is investigated. Both high-speed photography and PIV are used to collect the experimental data.
1.3 Thesis Outline

The thesis consists of five chapters and a few appendices. It is organized as follows:

- **Chapter 1** describes applications and challenges of premixed biogas flames. Motivations and objectives are presented at the end of this chapter.
- **Chapter 2** provides background and literature review with more emphasis on biogas combustion stability challenges and the different flame stability techniques employed so far.
- **Chapter 3** presents a description of the experimental setup including the different burner configurations and swirl strength examined in this thesis. In addition, the measurement techniques and experimental test conditions employed in this thesis are reported in this chapter.
- **Chapter 4** presents the experimental results and their discussion.
- **Chapter 5** outlines the main conclusions of the study and offers some recommendations for future work.
Chapter 2

BACKGROUND AND LITERATURE REVIEW

2.1 Combustion challenges of biogas fuels

Biogas is mainly a combination of methane and carbon dioxide with negligible amount of other constituent gases and is produced from anaerobic digestion of hydrocarbons. The major practical challenge of biogas, like any low calorific value fuel, is due to its low burning velocity and consequently weak flame stability. The presence of diluted carbon dioxide in biogas fuel decreases the heating value of the mixture, as it absorbs a fraction of the released energy of the oxidation reaction of the fuel [45]. Several studies are devoted to investigating methods for enhancing the stability of premixed biogas flames. The combustion and stability challenges along with methods of stabilization will be reviewed below. The first section of this chapter highlights the combustion challenges arising from the presence of carbon dioxide in biogas fuels. Stabilization techniques, non-blended and blended techniques examined in the literature are also reviewed.

2.2 Combustion challenges of biogas fuel

The presence of carbon dioxide as a main component of biogas composition has led to several combustion challenges which weaken biogas flame stability. It was suggested that these challenges can be either chemical, transport, thermal diffusion, or a combination of them [46]. Chemically, the presence of CO2 dilution reduces the fuel concentration which leads to lower flame speed as a result of decreased reaction rate. Another challenge of biogas is the low burning velocity. This can
be due to the alteration of mass/thermal diffusivities and specific heat capacity as an effect of transport and thermal diffusion [46]. Benedetto et al. [47] revealed that the low flame speed of biogas can be related to the combustion rate and flame temperature. These two significantly decrease as the specific heat of the mixture increases. It was reported that radiation heat transfer has a role in lowering the flame speed upon CO\textsubscript{2} dilution because it can act as an absorber and radiator. That causes the flame to lose a considerable radiative heat and, therefore, decreases flame velocity, flammable zone and adiabatic flame temperature [48]. Methods of stabilizing and enhancing premixed biogas flames are discussed in the following section.

2.3 Premixed Flame Stabilization Techniques

The presence of combustion instabilities in any industrial combustion system must be precluded to ensure safe and effective operation. This section discusses the different techniques used to enhance flame stability. The focus will be put on non-blended techniques, such as bluff-body and swirl, and also on blended techniques such as the addition of hydrogen and oxygen.

2.3.1 Flame Stabilization

In industrial power generation systems, combustion instabilities cause unsafe operation conditions and system failure [49]. Combustion instabilities can lead to flashback or blowoff conditions. Additionally, combustion instabilities can cause overheating and fatigue which can consequently instigate equipment failure. The buildup of acoustics inside a combustor precipitates unwanted vibration [50], and physical deformation can occur in circumstances where the amplitude of pressure oscillations is adequately large [84]. Moreover, rate of fatigue failure can be accelerated
due to these conditions. Table 2.1 illustrates the specific components’ risks consequent upon the type of instabilities depending on the oscillation frequency. In order to visualize the importance of avoiding instabilities, Figure 2.1 shows the difference between an intact burner assembly and an assembly which is deformed by combustion instabilities. The presence and propagation of thermoacoustic instabilities is an evidence of such system deformation in industrial combustors [49]. In an extreme case, these instabilities can lead to system failure in the form of an explosion. Therefore, studying and developing strategies for enhancing flame stability is of great importance.

Several methods to avoid the presence or occurrence of instability of premixed combustion systems have been proposed. These methods include passive and active control techniques. Passive control techniques, such as burner geometries, bluff-body and swirling flow, do not alter the chemical properties of the fuel but modify the flow characteristics (e.g., flow time scale, turbulence, etc). In addition, these methods require a good understanding of the physical phenomena happening at the concerned combustion system. These methods will be referred to as Non-blending techniques. Meanwhile, active control techniques, such as blending with hydrogen, alter the fuel chemical characteristics (e.g., chemical time scale). These methods will be referred to as blending techniques. Both techniques will be discussed in the next sections.
Table 2.1: Potential hazards associated with combustion instability [51].

<table>
<thead>
<tr>
<th>Description</th>
<th>Frequency Range (Hz)</th>
<th>Component Risks</th>
<th>Potential Causes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low-frequency dynamics</td>
<td>0-100</td>
<td>-Swirler damage</td>
<td>-Flashback indications</td>
</tr>
<tr>
<td></td>
<td></td>
<td>-Basket damage</td>
<td>-Lean blowout</td>
</tr>
<tr>
<td></td>
<td></td>
<td>-Nozzle damage</td>
<td>-Damaged swirl</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>-Air-flow restriction</td>
</tr>
<tr>
<td></td>
<td>100-500</td>
<td>-Transition panels</td>
<td>-High injection flow rates</td>
</tr>
<tr>
<td></td>
<td></td>
<td>-Transition seals</td>
<td>-Pilot-nozzle distress</td>
</tr>
<tr>
<td></td>
<td></td>
<td>-Fretting</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>-Wear</td>
<td></td>
</tr>
<tr>
<td>Intermediate-frequency</td>
<td>500-1500</td>
<td>-Downstream components</td>
<td>-Equipment distress</td>
</tr>
<tr>
<td>dynamics</td>
<td></td>
<td>-Fretting</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>-Wear</td>
<td></td>
</tr>
<tr>
<td>Intermediate-frequency</td>
<td>500-500</td>
<td>-Baskets</td>
<td>-Over-firing</td>
</tr>
<tr>
<td>dynamics</td>
<td></td>
<td>-Cross-flame tubes</td>
<td>-Fuel composition</td>
</tr>
<tr>
<td></td>
<td></td>
<td>-Flashback thermocouples</td>
<td>-Fuel composition</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>-System damping</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>-Basket distress</td>
</tr>
</tbody>
</table>
Figure 2.1. Potential effects of combustion instabilities in a diffusion jet combustor: a) combustor after combustion instabilities. b) new burner assembly [52].

2.3.2 Non-blending techniques

These techniques focus on altering the flow time scale by adopting a bluff-body, swirl or changing burner outer tube configuration. These techniques promote an appropriate flow structure that consequently improves the mixing fuel-air, and hence enhance flame stability. A discussion of the stabilization principles of these non-blending techniques which enhances premixed biogas flames is provided below.
2.3.2.1 Bluff-body

This technique is an approach that has the ability to stabilize a flame through a strong recirculation zone which occurs just downstream of the bluff-body (termed as flame holder). Theoretically, the principle behind this model is that a critical amount of energy is released by recirculating hot gases into the unburned mixture. The velocity at which blowoff occurs can be determined as [53]:

\[ U_{\text{blowoff}} = 2\rho_b L \frac{s^2_L}{\rho_u \alpha_T} \]  \hfill 2.1

where \( \rho_b \) and \( \rho_u \) are the density of the burned and unburned gas, respectively; \( L \) represents the recirculation zone’s characteristic length, and \( \alpha_T \) is the turbulent thermal diffusivity.

![Schematic of bluff-body for flame stabilization](image)

**Figure 2.2.** Schematic of bluff-body for flame stabilization [88]

The physics of bluff-body and recirculation zone, as shown in Figure 2.2 above, can be explained as follows [54-55]. The velocity of reactants flow (U) increases outward from the solid surface of bluff-body with a distribution gradient relative to the boundary of the bluff-body. Consequently,
the combustion wave stretches, and due to the effect of convection, the heat does not return to the same element as it diffuses out of the element of the wave. The recirculation zone acts as a heat reservoir that compensates for the heat loss. Under the assumption that vortex region is well stirred, the blowoff velocity can be expressed as follows:

\[
\frac{u_{\text{blowoff}}}{p^{n-1}d} = \text{constant}
\]

where \(P\) is the pressure of the gas, \(d\) is the diameter of bluff-body, and \(n\) is the order of the chemical reactions [90]. It was found that the residence time in the recirculation zone increases up to eight times in a bluff-body during combustion independently of the mixture ratio [56]. This suggests that this approach does not reduce NOx emissions. Bespalov [57] reported that flame stability limits increase with a rise in the initial and final temperature of the mixture, which favours thermal NOx formation.

2.3.2.2 Swirling flows

Swirling flows are utilized in various combustion applications such as gas turbines, boilers and practical heating devices. The presence of a swirl is favorable in these applications because it can effectively control/enhance flame stability, mixing, pollutants emission and combustion intensity [57-58]. In premixed flames, the mixture entering a combustion chamber is injected through a set of swirl vanes which create a rotational flow motion. Consequently, a low-pressure region in the flow core is created which is accompanied by adverse pressure gradients in both axial and radial directions. At a certain swirl strength, a recirculation zone is generated which reverses the direction
of the flow [59]. In order to define the level of flow swirling strength, a non-dimensional parameter, known as swirl number, \( S \) was proposed in [54].

\[
S = \frac{\text{axial flux of angular momentum}}{\text{axial thrust} \cdot R} = \frac{G_{\text{ang}}}{G_x \cdot R} \tag{2.3}
\]

where \( R \) represents the throat radius of the burner, \( G_{\text{ang}} \) represents the axial flux of the circumferential momentum and \( G_x \) represents the axial flux of the axial momentum. Swirl number can be determined by integrating the velocity field as in equation 2.4. A complete derivation of swirl number is presented in Appendix B.

\[
S = \frac{\int_0^\infty \rho U WR^2 dr}{\int_0^\infty \rho (U^2 - 1/2W^2) r dr} \tag{2.4}
\]

where the axial and tangential velocity components are represented as \( U \) and \( W \), respectively, and \( \rho \) is the density of fluid.

A sufficiently high swirling flow can recirculate the incoming/injected flow by a toroidal recirculation zone created at the center of the flow field. Hence, the recirculation zone can enhance flame stability by improving the mixing properties of fuel, oxidizer (in diffusion systems) and intermediate combustion products. Figure 2.3 illustrates important regions that appear within swirling flames. The first region to highlight is the central recirculation zone (CRZ), followed by the precessing vortex core (PVC) found on the boundary of the CRZ. The aerodynamic
recirculation introduced downstream of a swirling flow is responsible for actively recirculating heat chemical species upstream to be mixed with the fresh reactants, and hence enhances flame stability [60-61]. The recirculation zone creates a low velocity region where the flame generally anchors. In addition, the recirculation of reactants increases the flame speed and thus increases the level of turbulence [62]. In another way, the recirculation zone acts comparably to the wake region created by a bluff-body flame holder where a steady supply of reactants is recirculated to ignite fresh incoming unburned reactants. Batchelor [63] provided a description of the physics behind the generation of recirculation zone in swirling flow. It was suggested that the swirling flow has a vorticity vector represented in the axial direction, while the recirculation zone has a vorticity vector pointing in the azimuthal direction. Since there is no other source of vorticity in this case, and because the vorticity transport equation states that the only source to create the recirculation zone is by transforming some of the axial vorticity into azimuthal vorticity, this can be achieved by the diverging side walls since each fluid elements moves along a helical path. As the side wall divergence increases, the helical path radius increases and alters the direction of the vorticity vector in a way that declines the axial components and increase the azimuthal components. A detailed description of recirculation zone formation can be found in Batchelor [64].

On the other hand, the role of PVC on flame stability is found to be significantly dependent on the nature of the recirculation zone and the strength level of swirling flow [65]. It was shown that PVC occurs in the inner shear layer, boundary of CRZ, of the swirling flow that generates vorticities and strain structures [66]. It was reported that mixing rates of the reactants-products, which cause a rapid mixture ignition, are significantly improved by PVC [67]. Likewise, the existence of PVC can lead to enlargement of the flame surface as a result of roll-up flame front [67]. Flame roll-up
at low stagnation point of the flame root acts as a source of heat and unburned reactants. This would favour a rapid ignition and local increase of heat release which enhances flame stability [66]. However, PVC can lead to unfavorable effects on oscillatory heat release patterns and combustion [49]. Moreover, excessive swirling flow can destabilize a flame due to high entrainment of cold ambient air into the reaction zone. Cooling the recirculation zone leads to heat dissipation and ultimately flame extinction at high swirling flow [68].

![Figure 2.3. Streamlines and flow structure of a typical high swirling flow [50].](image)

**2.3.2.3 Conical (quarl) and cylindrical outer configuration and nozzle geometries**

Varying the burner geometry is another approach to enhance flame stability. Lovett et al [69] investigated the effect of changing burner geometry on natural gas flame stability and pollutant emissions. It was reported that positioning the swirl upstream resulted in a reduction in swirl strength, which led to higher NOx emissions indicating a decrease of fuel-air mixing rate. Positioning the swirl downstream near the nozzle exit resulted in a dramatic decrease in NOx emissions. In the same study [69], the effect of recessing the bluff-body was investigated. It was revealed that the recirculation zone ends at a similar position, which means that recirculation zone length and volume is significantly influenced by recessing the bluff-body. The recirculation zone
size and volume both increase with the recess [69]. Consequently, flame stability is enhanced due to the additional surface area that exists to ignite the fresh reactants exiting the burner, and hence enhance the flame stability. This recess can shield part of the recirculation zone and protect it from perturbations and turbulence in the dump region. An experimental study by Motevalli [70] in a multi-staged combustion with swirl generation using natural gas as a fuel. It was revealed that burner geometry significantly influences flame stability and emissions such as CO and UHC (unburned hydrocarbons) through controlling the mixing. Overlapped combustion zones can form due to the variation of burner geometry, which can lead to a better control of combustion temperature field, and consequently improve combustion efficiency. A recent study by M. Saediamiri et al [71] investigated the effect of nozzle geometry on non-premixed biogas flames. The tested fuel nozzles have either a single central hole or a central hole surrounded by multiple holes (with exit angle of 0° and 15°), as shown in Figure 2.4. Multi-hole nozzle (referred to as N₄ in [105]), when employed with swirling co-flow, showed the highest stability limits among other nozzle geometries. This is a result of improved mixing and more importantly create low velocity regions suitable for stabilizing biogas flame. The study revealed that a stable recirculation zone located near the nozzle exit improved the multi-hole nozzle flame. However, the recirculation zone created when using single hole nozzle geometries was located further downstream of the burner. This made biogas flames susceptible to blowout and quenching at lower flow rates. Flame stability and structure in conical burners were investigated in many studies [16,32 and 34]. It was reported that quarl angle significantly influences flame stability and topology. Flame stability was improved due to the vorticity structure inside the cone [32] and increasing the expansion angle increases the thickness in the mixing layer [33]. In addition, increasing cone angle enabled more ambient air entrainment and this, in return, affected the structure of the stabilization core leading to elongate
and narrow it [34]. However, excessive quarl angle stabilized the flame closer to the nozzle which is not desirable as it eventually damages the burner due to thermal stress [16].

![Multi-hole nozzle](image)

**Figure 2.4.** Multi-hole nozzle employed in [79].

### 2.3.3 Blending techniques

These techniques improve flame stability by altering the chemical time scale of biogas flame. This approach of enhancing biogas flame stability is by blending with higher calorific values fuels, such as H\(_2\) and O\(_2\). This approach can produce higher heating value, laminar flame speed and heat release, which expands the stability limits and allows leaner flames compared to non-blended biogas flames [72].

To better use biogas, which is categorized as poor fuel (in terms of calorific value and burning speed), its characteristics can be enhanced by blending it with higher heating value fuel, such as hydrogen. The benefit of adding hydrogen arises from its low lean flammability limit along with high burning speed [73]. Such features can improve the performance of lean premixed combustion, where laminar flame speed and flammability limit are enhanced to great extent. Leung et al. [74] experimentally investigated the effect of biogas-hydrogen blended fuel and found that a small amount of hydrogen significantly widens the stability limits of biogas flames. Furthermore, it was revealed that hydrogen can accelerate the heat release rate and enhance the combustion stability of
biogas flames [75]. The increase in stability limits and flame velocity have been attributed to transport effects, which play an important role in the decrease of flame sensitivity to strain or stretch rates through a reduction in Lewis number for hydrogenated blended fuels [76]. In another word, the rate of molecular diffusion is more significant than the rate of thermal diffusion at smaller Lewis numbers. Consequently, weak and small vortices are formed at the surface of hydrogenated flame compared to non-hydrogen blended flames.

Biogas-hydrogen blending approach is a promising technique for improving flame stability and reducing pollutant emissions (CO) [50-52]. However, this technique still has several drawbacks which include a rise in NOx emissions, high tendency of flashback, storage risks (example; H\textsubscript{2} due to low ignition Φ and low flash point) and capital and operational costs, all of which limit the implementation of blended biogas combustion [53-54]. In addition, higher flame speed in hydrogenated flames may require design modification to optimize the burning performance [77].

Another biogas blending technique concerns the dilution with oxygen or burning biogas with oxygen enriched air, which was found to enhance flame stability [78]. It was reported that enriching air with oxygen improves the blowoff limits [67]. However, the tendency of flame flashback is a concern due to the increase of adiabatic flame temperature [79]. NOx emission is found to be insignificant in oxy-flames. However, CO emission can significantly increase due to the production of OH and CO from CO\textsubscript{2} chemical reaction. [69]. A similar conclusion was achieved in [80]. Oxy-flame is found to lower the thermal energy loss compared to pure air combustion. However, one of the primary problems is the cost involved in oxygen production and burner modification [81].
2.4 Literature on The Effect of Non-Blended Techniques on Premixed Biogas Flames

Several studies utilized non-blending techniques in order to stabilize premixed biogas flame. Lafay [5, 24] experimentally compared the stability and blowout limits between methane to biogas (with 12-30% CO2) flame. The reduced laminar flame speed of biogas decreases the stable flame domain and consequently creates flame velocity oscillation when varying the equivalence ratio (Φ) [5,24]. This oscillation creates flame instability at low frequency fluctuations, leading to the development of an unstable combustion region [5,24]. This phenomenon was confirmed by a study adopting similar configuration but using one third lower CO2 percentage [15]. The studies in [5,15,24] adopted a centerbody (bluff-body) and reported an enhancement of biogas flame stability via improvement in the internal recirculation zone (IRZ) that is created by high swirling flow and bluff-body, as well as a large corner recirculation zone (CRZ) resulting from sudden expansion of flow. The effect of swirl number (S=0.04-0.7), dilution (CO2=25, 50, 75%) and mass flow rates ($\dot{m} =100-500$ SLPM) on lean blowout limits (LBO) are studied [7]. Lean blowout (LBO) limits are enhanced with increasing S and $\dot{m}$ due to flame stretch [7]. Blowout limits of biogas improved as S increases (0.69 to 1.35 in 25% O2) due to the reduction of reagents turbulent mixing time [14]. It was reported that as swirl number increases, the biogas flame lifts as a result of the creation of a central recirculation zone [14].

The structure and stabilization of partially premixed low calorific value gases (LCVG) and CH$_4$/N$_2$ in conical non-swirl burner is studied by [32]. Flame stability was found to improve due to the vorticity structure inside the cone which is found to rely more on cone angle and less on flow speed [32]. Additionally, flame sensitivity to the used fuels composition was slightly reduced [32]. Quarl angle (10°,15°,45° & 60°) was found to significantly influence the flame liftoff and topology,
blowout and stability of swirling flows (S = 0.5-1.2) [16]. The increase in the expansion angle and swirl reduces the velocity and increases the thickness of the mixing layer, making flow velocity comparable to flame propagation velocity [33]. Higher quarl angle (45˚ and 60˚) stabilizes the flame close to the burner nozzle which can potentially damage the burner due to thermal stress [16]. In addition, increasing cone angle allows more ambient air entrainment and this will affect the structure of the stabilization core leading to elongate and narrow it [34].

However, literature associated with non-blending techniques is only one of several approaches to enhance LCVG stability. Another approach to enhance the stability limit of biogas is blending biogas with high calorific value fuels, such as H₂, O₂ or LPG. This approach yielded higher heating value, increased flame speed, flame temperature and heat release rate compared to non-blended biogas. Burning velocity monotonically increases by blending with higher LCV fuels [42–44], resulting in stability limits expansion and leaner flames [45–49]. Biogas blending has the potential to improve flame stability and reduce pollutant emissions (CO) [50–52]. However, blending has concerns such as a rise in NOx emissions, high tendency of flashback, storage risks (example; H₂ due to low ignition Φ and low flash point) and capital and operational costs which limit the implementation of blended biofuels flames [13, 53, 54].
Chapter 3

EXPERIMENTAL SETUP AND TEST CONDITIONS

3.1 Experimental Setup

3.1.1 Burner

The experimental setup, which was designed and fabricated in-house, consists of a central bluff-body surrounded by a swirling premixed biogas stream. The premixed flow (fuel and air) is discharged into a confined chamber as shown in Figure 3.1. A detailed schematic diagram of the experimental setup including the mixing pipes, flow control panel and seeding system is also presented in Figure 3.1. The chamber, which has a square cross section, has three optical windows, wherein one window has a height of 400mm and a width of 320mm, and the other two windows each has a height of 400mm and width of 270mm. The forth chamber wall has a door of a height of 350mm and a width of 210mm which provides access to the burner. This access is needed to alter the swirl generator and burner. This chamber size (360mm×36mm×900mm) is large enough as such it does not interfere with the flow field and flame. A vent is located at the top and connected to the exhaust. Fuel mixture (CH₄ and CO₂) and air were controlled using Matheson and Brooks flowmeters. The flowmeters are FM-1050 Series and have a maximum pressure of 250 PSIG and a maximum temperature of 250°F. Matheson Flowmeters have an accuracy of ±5% of full scale and a repeatability of 0.25% scale reading. Both air and fuel flowmeters were calibrated in house using wet gas flowmeter. To ensure a good mixing of biogas fuel components (CH₄ and CO₂) with oxidizer (air in this case), a 600 mm long cyclone-type mixing pipe is used. This pipe is connected
to a manifold which supplies the combustion chamber with premixed gas as shown in Figure 3.1. Air was supplied from a laboratory compressed line, and both methane and carbon dioxide with 99% purity were supplied by compressed cylinders. All experiments were performed at 20 PSI operational pressure, and a biogas composition of 75% methane (CH₄) and 25% carbon dioxide (CO₂) is maintained throughout all experiments. The burner consists of a bluff-body, an axial swirl generator and a conical or cylindrical outer tube as schematically shown in Figure 3.2. The burner dimensions are as follows; 806.45 mm in height, 42.45 mm in diameter with a 21 mm diameter centerbody. The cylindrical outer tube configuration has a total length of 43.5 mm, an inner diameter of 42.45 mm and an outer diameter of 68.2 mm. The conical outer tube configuration has a total length of 43.5 mm with an inner diameter of 50.8 mm and an outer diameter of 68.12 mm. The conical configuration has a discharge angle of $\beta = 15^\circ$, which is within the range of optimum angles [16,34].
Figure 3.1 Schematic diagram of the experimental setup. (A) Gas cylinder. (B) Valves. (C) Regulator. (D) Needle valve. (E) Matheson and Brooks flowmeters. (F) CH4-CO2 gas manifold. (G) Air flowmeter. (H) CH4-CO2-Air manifold. (I) Bypass line. (J) Seeder. (K) Distribution manifold. (L) Burner. (M) Exhaust. (N) Compressed air.
The axial swirl generator used in this thesis is schematically shown in Figure 3.3. The vanes angle of the swirl generator represents the magnitude of the angular velocity that determines the swirl strength. The vanes angles adopted in the current study are 25° and 60° which represent a swirl number (S) of 0.39 and 1.16, respectively. Zero-degree axial swirl generator was used as a reference, and it was tested in isothermal flows only. Swirl strength depends on several parameters like swirl generator vanes and flow conditions [78]. However, since burner geometry is fixed in this case, the swirl vane angles dominate the magnitude and strength of the swirl. More details on how the swirl number (strength) was calculated in the present study is reported in Appendix B.
3.1.2 Measurements Technique

Dantec Dynamics planar Particle Image Velocimetry (PIV) system was utilized to characterize and analyse the reacting and isothermal flow fields. The PIV system consists of a double pulses ND: YAG laser which provides a maximum power of 135mJ per pulse with a wavelength of 532nm at 15 Hz frequency, which was adjusted to 10 Hz in order to match the frequency of the camera. The camera is FlowSense EO 4M which is capable of capturing 10 double frames per second at a full resolution of 2048 x 2048 pixels² and 8µm pixel pitch coupled with a Nikon 60 mm AF lens. The employed laser consists of a laser head of 80 x 80 high power light sheet optics which is composed of a number of modules to adjust the sheet light thickness and divergence angle. These mentioned modules have a light sheet base module (9080 x 0901), and the laser sheet thickness that can be
adjusted manually via varying the focal point of the laser sheet through the lens module placed on the laser head. The nominal adjusted laser sheet thickness is 1 mm, which is sufficiently large to maintain particles within the sheet plane throughout their motion. A schematic diagram of Particle Image Velocimetry (PIV) setup is given in Figure 3.4. Processing of the captured images was conducted using Dantec Dynamic Studio software. This software basically performs the task of estimating the average velocity of the captured seeding particles in the interrogation area at specific pulse times, as well as allocate the images into a specific grid with small interrogation area. Determining the particles instantaneous velocity is achieved by calculating the displacement of a particle in the interrogation area of pairs of images. The seeding particles used in this study are Titanium dioxide TiO2, with an average size of one micron [78]. The seeder, which is designed and fabricated in-house, consists of a brush that is connected vertically to a shaft that agitates mechanically TiO2 particles. The seeding particles are placed at the bottom of the seeder/tank. The premixed gas is introduced to the seeder, and with the aid of the agitated brush, the premixed gas is mixed with the fine solid particles. Then, the gaseous mixture flows into the combustion chamber. To control the level of seeding particles in the flow, we used either a manual change of the brush rotation speed to ensure better mixing of solid and gaseous particles or via adjusting the by-pass valve to introduce more gas mixture into the seeder. Controlling the level of seeding particles is important to ensure the quality of flow field analysis.

PIV images are captured in the near field of the burner exit in a region of approximately 100mm$^2$ x 100mm$^2$. This leads to a spatial resolution of roughly 48.8µm/pixel. For each test case, 800 pairs of images were captured. These images were processed using adaptive correlation of 32 pixels x
32 pixels with 50% overlap window. The laser pulse time delay was adjusted for every flow condition relying on the size of interrogation area and flow local velocity.

Figure 3.4. Schematic diagram of PIV setup [78]

Laser Doppler velocimetry (LDV) was used to determine the swirl number of the axial swirl generator. The LDV system consists of an Innova 70C series 5W argon-ion laser, transceiver, beam separating device, photomultiplier device that converts signals from optical to electrical, signal processor and analysing software. The LDV has four equal intensity laser beams, two green and two blue of a wavelength of 514.5 and 488nm, respectively. These beams are intersected via a 363mm focal length lens and delivered through an optical fiber into a transceiver. The beam intersection at the lens focal point forms the probe volume that consists of a set of bright and dark fringes. As the flow seeding particles travel across the bright fringes, they scatter light. Photomultiplier Tubes (PMTs) (i.e. the Photo Detector Module i000, PDM 1000) converts the collected scattered light via the receiving optics. PDM 1000, in return, transmits an electrical signal to a TSI Flow Size Analyser processor 4000 (FSA 4000). FSA 4000 extracts the Doppler frequency
of the seedings particle that crossed the bright fringe. The distance between the fringes, called fringe spacing $\delta_t$, is dependent on wavelength of the beam laser light as well as the intersection angle of the beams. Fringe spacing is fixed since the geometrical setup does not change [90]. After determining the fringe spacing ($\delta_t$), it can be multiplied by the output frequency ($f$) and as a result the velocity ($u$) is obtained as follows:

$$u = \delta_t xf$$ \hspace{1cm} (3.1)

where the fringe spacing is determined as

$$\delta_t = \frac{\lambda}{2 \sin(k)}$$ \hspace{1cm} (3.2)

where $\lambda$ is the wavelength of the laser, and $k$ is the angle between the focused beams. The fringe spacing in this experiment was found to be $2.462 \times 10^{-6}$ and $2.34 \times 10^{-6}$, horizontally and vertically, respectively. A total of 20,000 sample data were taken at each channel in order to determine the vertical and horizontal velocities.
3.2 Test Conditions

Test conditions of the present study consisted of varying the burner configuration which include recessing the center body and outer tube shape (cylindrical or conical), swirl strength and Reynolds number (Re). Re was calculated using the following formula:

\[ Re = \frac{V_m \times De}{v} \]  

where \( V_m \) is the bulk velocity of the mixture at the burner exit, \( De \) is the hydraulic diameter, \( De = D_{tube} - D_{CB} \), and \( v \) is the mixture viscosity. The experiments, however, were conducted at a constant volumetric fuel ratio of 75% methane and 25% carbon dioxide. Table 3.1 provides a summery of the experimental test conditions that were used for PIV measurements. The tested configurations are presented in Figure 3.6. Figure 3.6a represents the reference case, at which no outer tuber configuration is added. In this case, the swirl has a recess of 25 mm upstream of the
tube exit, as shown in Figure 3.3. Furthermore, a conical or cylindrical outer tube configuration is tested with the center body, as shown in Figure 3.6b&c respectively. Figure 3.6d&e shows the cylindrical and conical outer tube configuration with a recessed center body. The procedure for determining the stability maps (at low and high swirl number) of premixed biogas flames was accomplished by gradually increasing the air flow rate of the premixed gaseous fuel (Re). The fuel rate is kept constant while increasing the air rate (Re) of the mixture. This leads to a leaner equivalence ratio and ultimately flame extinction. This approach was found to be effective because the initial ignition of a premixed lean flame is more challenging than the ignition of a fuel rich mixture. Therefore, after ignition of the fuel rich premixed mixture, the air flow rate is gradually increased while maintaining a constant fuel flow. The increase of air decreases the equivalence ratio while increases the overall inlet velocity. The blowoff occurs when the equivalence ratio is sufficiently lean.

<table>
<thead>
<tr>
<th>Geometry</th>
<th>( \Phi )</th>
<th>Reactant flow</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>S = 0.39</td>
<td>(LPM)</td>
</tr>
<tr>
<td>Ref</td>
<td>0 - 0.33</td>
<td>450 - reacting</td>
</tr>
<tr>
<td></td>
<td>S = 1.16</td>
<td>500 - isothermal</td>
</tr>
<tr>
<td>Ext-CB</td>
<td>0 – 0.33</td>
<td>450 - reacting</td>
</tr>
<tr>
<td></td>
<td>0 – 0.34</td>
<td>500 - isothermal</td>
</tr>
<tr>
<td>Cone-CB</td>
<td>0 – 0.335</td>
<td>450 - reacting</td>
</tr>
<tr>
<td></td>
<td>0 – 0.37</td>
<td>500 - isothermal</td>
</tr>
<tr>
<td>Ext-recess</td>
<td>0 – 0.34</td>
<td>450 - reacting</td>
</tr>
<tr>
<td></td>
<td>0 – 0.345</td>
<td></td>
</tr>
</tbody>
</table>

**Table 3.1:** Test conditions
Figure 3.6. Schematic of burner exit geometry (a) Reference case (Ref); b) tube extension with center body (Ext-CB); c) cone extension with center body (Cone-CB); d) tube extension with no center body (Ext-recess); and e) cone extension with no center body (Cone-recess).
Chapter 4

RESULTS AND DISCUSSION

4.1 Introduction

Experimental results on the stability limits of an attached premixed biogas flame as a function of different burner configurations are presented in this chapter. First section focuses on the stability limits of premixed biogas flames in terms of different burner configurations and swirl number. The second section reports PIV flow measurements and their discussion in relation to the variation of flame stability limits. Lastly, the effect of flow rate (Re) and heat release (Φ) is discussed.

4.2 Flame Stability Limits Map

The effect of different burner configurations on the premixed biogas flame (75% CH₄ and 25% CO₂) stability is presented in Figure 4.1a and 4.1b at swirl number of 0.39 and 1.16, respectively. Blowoff is defined here as the flame extinction of the attached flame. This appears when the local flow velocity exceeds the local burning velocity causing the flame to extinguish. Blowoff measurements were achieved by increasing the air flow rate of the mixture (represented by Re in this figure) while maintaining a constant fuel flow rate until flame extinction, where the equivalence ratio varies according to fuel to air ratio. The results show that the effect of burner configurations on biogas flame stability is more significant at higher swirl number (S = 1.16), as compared to that at low swirl number (S=0.39). Increasing the swirl number shifted the blowoff
limits to slightly higher equivalence ratio ($\Phi$) compared to that at low swirl number. The highest flame stability limit is achieved by recessing the centerbody (i.e., removing the centerbody) at low swirl number, 0.39 (Figure 4.1a). Recessing the centerbody at high swirl number, 1.16, exhibited the highest stability limits amongst other burner configurations. Moreover, Ext-CB (cylindrical outer tube with extended centerbody in Figure 3.6b) and Reference case (Ref. in Figure 3.6a) exhibited approximately similar trend, as shown in Figure 4.1a for the flame stability limits at low swirl number ($S=0.39$). In the case of high swirl number (1.16), positioning the swirler upstream (i.e., Ext-CB configuration) resulted in a reduction in the swirl strength compared to reference case (Ref.) due to the decay in swirl strength. Such a low stability limit is an indication of a decease in the mixing rate [68].

This figure also shows that among all examined burner configurations, the conical configuration has the lowest stability limit at low swirl number for both recessed and extended center body (see Figure 3.6c and Figure 3.6e). At low air flow rate (i.e., at low Re), the effect of burner geometry is not very significant especially at high swirl number, as shown in Figure 4.1b. Increasing the air flow rate of the mixture expands the stability limits, but its effect is not very significant in the case of Cone-CB (cone extension with no center body) configuration when compared to the other configurations. At both swirl numbers, 0.39 and 1.16, it can be noticed that the lowest flame stability is associated with Cone-CB configuration. On the other hand, the highest premixed biogas flame stability is achieved when recessing the centerbody, that is with Ext-recess configuration. Therefore, these configurations are chosen for further investigation and their results are discussed in section 4.3 as they represent the upper and lower limit of premixed biogas flame stability.
4.3 PIV Measurements and Discussion of Biogas Flame Stability Limits

This section attempts to discuss the biogas flame stability maps with the aid of PIV measurements. The results of these measurements are presented in terms of velocity flow field contours and turbulent level contours (turbulent kinetic energy).

4.3.1 PIV Measurement of Flow Field Characteristics

PIV measurements of premixed biogas flames were acquired to characterize reacting and non-reacting flow fields to help understand the stability maps presented in Figure 4.1. In isothermal flow (no combustion) cases, the flow field was investigated at $S = 0.39$ and $1.16$ with a flow rate of 500 LPM. In addition, the flow field of reacting (in the presence of combustion) cases were investigated at $S = 0.39$ and $1.16$ with a flow rate of 450 LPM. Figure 4.2 illustrates the mean velocity contours of premixed biogas flame for different burner configurations (cylindrical or conical with recessed/extended center body burner configurations). The PIV results revealed that increasing the distance between the swirl generator and the burner exit (compare Figure 4.2f to...
Figure 4.2g) led to a reduction in the central negative velocity region. This is due to the center body flame holding effect which weakens under these flow conditions and consequently decreases biogas flame stability. This finding is in a good agreement with that reported in [69] where similar conclusion was achieved for a natural gas flame tested using a similar burner configuration and different positioning of swirl generator. However, it can be observed in Figure 4.1 that by recessing the centerbody, the biogas flame exhibited the highest flame stability for both swirl numbers. Recessing the center body allowed the recirculated reactants to have more surface area to mix with, and thus, ignite the fresh/reactants mixture. This is because the exit area of the fresh mixture occupies the entire cross-sectional area of the outer tube, and hence the length and volume of the recirculation zone increased and consequently the biogas flame stability was enhanced significantly [69]. Recessing the center body can be considered as a shield to the recirculation zone from flow perturbations and turbulence in the dump region [69]. By recessing the centerbody, the instability oscillation seemed to gradually decrease and hence allowing the flame to stabilize over a much wide range of equivalence ratio [83-84]. This is caused by fact that the flame-vortex interaction is altered when recessing the centerbody, and hence affects positively the combustion process (i.e., stabilize the flame) [83]. This is due to the impact of the confinement (outer tube) on the flame, and the decreased interaction between the flame and vortex due to the position of the flame in this situation. The flame is positioned upstream of the dump plane where the vortex is formed [83-84].

It can be observed in this figure that the biogas flame of the Cone-recess configuration exhibited larger vortices and noticeable negative velocity central region compared to that of the Cone-CB configuration. Consequently, this is an indication of higher vortex strength and thus a larger flame
area downstream of the conical confinement (compared to Cone-CB case). Conical burner promotes flow divergence, and consequently greater interaction with the entrained cold flow into the flame. This can explain why the Cone-CB case experienced the lowest flame stability, since the high entrainment (indicated from higher flame divergence) from the cold surroundings quenches the flame roots. However, with the non-conical (i.e., cylindrical) cases, increasing Reynolds number led to short and stable biogas flames, which are anchored onto the center body and far from the outer tube wall (see flame images in Figure 4.3). In contrast, the strong flow divergence inside the cone configuration case led to more stable biogas flames interacting with the cone wall and far from the centerbody. This caused the wall temperature to increase which is not desirable as it leads to high thermal stress and thus burner damage [25]. A common feature of all examined burner configurations is the blue color flame, which is indication of less pollutant biogas combustion (see Figure 4.3).
Figure 4.2. Mean velocity contour maps and streamlines of (a-e) isothermal flow at 500 LPM and S=0.39, and (f-j) reacting flow at 450 LPM and S=0.39.

Figure 4.3. The corresponding biogas flame images of reacting flow cases at 450 LPM and S=0.39 for a) Reference case (Ref); b) tube extension with center body (Ext-CB); c) cone extension with center body (Cone-CB); d) tube extension with no center body (Ext-recess); and e) cone extension with no center body (Cone-recess).
The vector of velocity flow field shown in Figure 4.4 is in a good agreement with the measurements reported in [85]. The close section of the velocity flow field is shown in Figure 4.4b which exhibits a velocity deficit due to the presence of a recirculation zone caused by the swirling flow, followed by a wake region and a velocity deficit around the centerline. The central velocity shifts upstream as a result of the momentum which is transported/ transferred from the high velocity reactant stream. Therefore, the premixed biogas flame can propagate into the low velocity zone of the wake region. Such a low velocity region explains why the flame stabilizes onto bluff-body under such lean conditions as revealed by Schefer [85].

![Flow mean velocity vectors](image)

**Figure 4.4.** Flow mean velocity vectors (a) velocity field vectors and b) zoomed section of the central region

The distribution of turbulent kinetic energy, $q'$, of the examined different burner geometries/configurations for both isothermal and reactive flows is shown in Figure 4.5. Both cases were investigated in order to better understand the level of turbulence in the flow field of both cylindrical and conical outer tubes configurations. The contours in Figure 4.5 show the turbulent intensity distribution which is needed to explain the flame stability of the premixed biogas flames. The turbulent kinetic energy was calculated using the following equation (4.1):
\[ q' = 0.5 \sqrt{u'^2 + v'^2} \]

where \( u' \) and \( v' \) are the rms fluctuation velocities in the axial and radial direction, respectively.

This figure shows that, for non-reacting flows, the high turbulence level is located in the core region of the wake flow as well as in the shear layer due to large velocity gradient [86]. In case of conical configurations, the highest turbulence level is found in the shear layer between the recirculation zone region and annular flow (see Figure 4.5c,e,h,j). The distribution of turbulence intensity level in the reacting flows shows a different trend than those of their counterparts’ isothermal, as seen in Figure 4.5. A common feature of all configurations is that the turbulence kinetic energy is found to be the highest in the shear region. Johnson et al. [86] attributed such increase of turbulence kinetic energy to the high mean velocity across the oblique wrinkled flame front. In addition, the turbulence level is found to decline within the recirculation zones. The turbulence level is decreased in the region where there is an increase in viscosity and a decrease in density, which are found in the region of highest flame temperature. The increase in viscosity in the recalculation zone is the main reason of the decrease in the turbulence level [86,87]. However, it should be noted that recessing the centerbody in non-conical configuration, shows a different turbulence level trend compared to the other geometries. As shown in Figure 4.5i, \( q' \) increases in the core region downstream of the recirculation zone. Driscoll et al [28] reported that this could be due to asymmetric recirculation zone oscillation (see Figure 4.56a point 4&5) where its fluid dynamic center is sometimes far above the geometric centerline, and other times it is far below the centerline. The asymmetric recirculation zone tends to have large rms values, as shown in Figure 4.5i, as a result of large variation of the axial velocity created by the oscillation of the asymmetric recirculation zone [28]. For conical configurations, it is obvious that the centerbody influences the
turbulence level in the flow. The turbulent kinetic energy distribution is high in the shear region only for the case of Cone-CB burner configuration, while the turbulence level is found relatively equal between shear layer region and downstream of recirculation zone when recessing the centerbody.

**Figure 4.5.** Mean velocity contour maps of (a-e) isothermal turbulent kinetic energy cases at 500 LPM and S=0.39, and (f-j) reacting turbulent kinetic energy cases at 450 LPM and S=0.39
4.3.2 Effect of Flow Rate (Reynolds Number) and Heat Release (Equivalence Ratio) on recirculation zone

Recirculation zone size and shape significantly affect the flame stability via controlling the residence time [88,89]. Therefore, this section examines the effect of heat release (equivalence ratio) and flow rate (Re) on the recirculation zone shape to better understand the biogas flame stability limits of the two extremes cases (highest and lowest stability limits). The effect of heat release and flow rate on the flow field is investigated at low swirl number (0.39), as shown in Figure 4.5 a&b. Two burner geometries were selected, Cone-CB and Ext-recess, since they represent the lowest and highest stability limits. The non-reacting flow rate, Ext-recess, was not strong enough to create a recirculation zone, and the axial flow radially curls toward the center. For the case of Cone-CB, small recirculation zone is formed near the wake region of the centerbody in non-reacting (isothermal) flow. However, by increasing the heat release (that is, increasing Φ), Ext-recess burner configuration exhibits an asymmetric recirculation zone as a result of enhanced Reynolds shear stress and the anisotropy of turbulence. Further increase of equivalence ratio when using non-conical burner geometry (compare point 3 to 2 in Figure 4.6a) led to a stronger and slightly symmetric recirculation zone as shown in Figure 4.6b. Increasing the air flow rate (Re) at constant Φ, point 3 to 4 in Figure 4.6a, produced an adverse effect on the recirculation zone for the Ext-recess case. The negative velocity region decreased, and a single recirculation zone is observed by comparing point 4 to 3, as seen in Figure 4.6b. Such an asymmetric recirculation zone can enhance the mixing which promotes better flame stability for Ext-recess case.

Relatively similar flow field was obtained for Ext-recess case with a further increase of Φ (see point 5 in Figure 4.6a&b). This may indicate that the effect of air flow rate (Re) is more dominant
than the effect of equivalence ratio at higher Reynolds numbers. Conical configuration and Cone-CB produced a different trend than non-conical configuration. As mentioned above, small recirculation zones are formed near the wake region (point 2 shown in Figure 4.6b) and increasing the air flow rate (Re) enhanced the negative central region and the recirculation zones (point 4). However, neither increasing the air flow rate (Re) nor heat release (Φ) is found to considerably affects the flow field of conical geometries, in particular Cone-CB case as shown in the corresponding streamlines in Figure 4.6b. This can explain the low stability limits of Cone-CB at both swirl numbers.

Figure 4.6a. diagram of equivalence ratio vs. Re number at S=0.39 for different burner configurations (Cone-CB (Case III) & Ext-recess (Case IV)).
Figure 4.6b. Mean velocity vectors and the corresponding streamlines of the different burner configurations (Cone-CB (Case III) & Ext-recess (Case IV)) represented by points 1 to 5 in diagram of Figure 4.6a.
CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

5.1 Concluding remarks

This thesis experimentally investigated the effect of burner geometry on the stability of turbulent premixed biogas flame. Experiments were conducted at standard temperature and pressure, and the surrogate biogas composition is kept constant (75% CH₄ and 25% CO₂). In this study, the burner geometry (Conical or cylindrical with or without a centre body), swirl number (S = 0.39 and 1.16) and equivalence ration were all varied. The main findings of this thesis can be summarized as follows:

1- The effect of burner geometry (conical or cylindrical) is more pronounced at higher swirl number. However, increasing the swirl strength slightly lifts the blowoff limits toward higher equivalence ratio. In both burner geometries, Ext-recess (case IV in Figure 3.6) showed the highest stability limits where Cone-CB (case III) produced the lowest stability limit. In addition, positioning the swirl generator upstream led to a reduction in the swirl strength, which in return reduced flame stability limits due mainly to a drop the mixing level.

2- Particle image velocimetry measurements revealed that increasing the distance between the swirl generator and burner exit reduced the central region of negative velocities (which acts as flame holder), which in turn weakened the biogas flame stability.
3- Particle image velocimetry data revealed that recessing the centerbody caused an axisymmetric vortex breakdown in the central region, which consequently led to improved flame stability.

4- Flow divergence is significant when using conical burner configuration. This caused the flame to become in direct contact with the conical inner wall. Unlike the conical configuration, the cylindrical burner configuration was found to induce a short and stable biogas flame anchored to the centerbody away from the burner wall.

5- The instability of biogas flame under lean fuel conditions can be attributed to the velocity deficit in the central flow region. Particle image velocimetry showed that the recirculation zones generated by the swirling flow with the aid of the wake region generated by the bluff-body gradually decreases the velocity deficit. This condition, therefore, allowed the premixed biogas flame to propagate upstream through the low velocity region and attaches to the bluff-body.

6- In the case of non-conical burner configuration, non-reacting flows exhibited the highest turbulence level in the core region and in the shear layer as a result of large velocity gradients. Meanwhile, the conical configurations exhibited the highest turbulence level in the recirculation zones. This further explains the difference in the biogas flame stability limits between the two distinct burner geometries (i.e., cylindrical versus conical).

7- Turbulence intensity distribution of reacting flows in the case of the non-conical configuration was found to be the highest in the shear region. However, turbulence level distribution of case IV, Ext-recess, was found to be significant in the core region and shear region which is the reason for the significant stability limits of case IV.
8- The turbulence intensity distribution of the reacting flows for the conical configuration was found high mainly in the core region. However, recessing the centerbody led to a turbulence distribution that is relatively similar in the shear layer and recirculation zone. The low stability limits associated with this burner geometry can be related to the decrease in the turbulence level within the recirculation zones of the burner conical configuration.

9- The effect of flow rate and equivalence ratio on the flow field revealed that, for case IV, increasing the equivalence ratio at a constant flow rate led to an asymmetric recirculation zone. However, further increase in equivalence ratio resulted in a strong and relatively symmetric recirculation zone.

10- Case III of the conical configuration produced a different trend compared to that of case IV, where the recirculation zones were not significantly affected by varying the equivalence ratio and flow rate.

11- A common feature for all burner geometries is that increasing swirl number led to higher flow divergence which negatively affected biogas flame stability.

5.2 Recommendations for future work

To extensively investigate the stability of premixed biogas flames, additional experimental research is required. Some suggestions for further research are as follows:

1- Investigate the flame stability over a wide range of biogas composition. This is because biogas composition is not always fixed and depends on feedstock.

2- Extend the study of the burner conical geometry by varying the exit angles and also the bluff-body size/shape.
3- Blending biogas with higher calorific value fuels, such as hydrogen or oxygen, to investigate the coupled effect of burner geometry and blending on biogas flame stability.
REFERENCE


[50] Lang, W., Poinsot, T., and Candel S. “Active control of combustion instability.”


[51] Lieuwen, T. “CDMS helps prevent forced outages, tune engine after overhaul.”


In: Lieuwen, T., Yang, V., ed. Combustion instabilities in gas turbine engines:

operational experience, fundamental mechanisms, and modeling. Prog. Astronaut.

Aeronaut. 210, 163-175, 2005.


[73] Lieuwen, T. “CDMS helps prevent forced outages, tune engine after overhaul.”


[82] TSI LDV/PDPA system Instruction Manual, July 200r, combustion Laboratory, Room E3-340 Engineering building, university of Manitoba, Winnipeg.


[90] Petersson, P. “Laser Diagnostics Applied to Lean Premixed Swirling Flames - Simultaneous Flow Field and Scalar Measurements”, Division of Combustion Physics, Department of Physics, Lund University. 2014.
Appendix A

Isothermal Flow Field of Various Flow Rates and Swirl Strength

A1. An overview of the isothermal flow field and swirl strength

The effect of flow rate (75, 200 and 500 LMP) on the streamlines flow fields of the geometry variation for different swirl numbers, $S=0^\circ$, $30^\circ$, $45^\circ$ and $60^\circ$, Figure A1 shown below. Case V, Cone-CB, at swirl number ($30^\circ$) has two small recirculation zones even at low flow rate of 75 LPM. These recirculation zones get stronger and smaller in size when flow rate equals 500 LPM. Meanwhile, the small recirculation zones dissipated by increasing flow rate and instead one recirculation zone is found by removing the center body. Different trend is observed with higher swirl number ($60^\circ$), the cone with and without center body exhibit strong recirculation zones even at low flow rate and they become smaller and stronger as flow rate increases. When extension is added with center body at low swirl number, it is observed that straight lines or insignificant recirculation zone is created at low flow rate, but with increasing flow rate to 500 LPM they are formed. Similar trend was observed for reference (Case I). However, removing the center body from the extension, Case IV, caused the straight line at low flow rate number to curl radially towards the centerline as flow rate increases. For higher swirl number, different trend was revealed. The recirculation zones are not equal in size and as flow rate increases they both become equal in size, shorter and stronger in case of Ext-CB. While Ext-recess and Ref has long and large recirculation zones at 75 LPM, and they become asymmetric in size and shape at high flow rate. In addition, straight stream lines are obtained with zero vanes angle, $S=0^\circ$, for recessed centerbody geometries. For case I and case II, increasing the flow rate to 500LPM caused the recirculation zones to be asymmetric. While Case V had small recirculation zones on centerbody.
various burner geometries of swirl strength with vane angle 45° was approximately similar to those of swirl strength 60°.

Cone-CB (case III) $\dot{m}=75-200-500$ LPM, $S=0^\circ$

Cone-recess (case V) $\dot{m}=75-200-500$ LPM, $S=0^\circ$

Ext-CB (case II) $\dot{m}=75-200-500$ LPM, $S=0^\circ$

Ext-recess (case IV) $\dot{m}=75-200-500$ LPM, $S=0^\circ$
Ref (case I) $\dot{m}=75$-200-500LPM, $S=0^\circ$

Cone-CB (case III) $\dot{m}=75$-200-500LPM, $S=30^\circ$

Cone-recess (case V) $\dot{m}=75$-200-500LPM, $S=30^\circ$
Ext-CB (case II) $\dot{m}=75\text{-}200\text{-}500\text{LPM}$, $S=30^\circ$

Ext-CB (case IV) $\dot{m}=75\text{-}200\text{-}500\text{LPM}$, $S=30^\circ$

Ref (case I) $\dot{m}=75\text{-}200\text{-}500\text{LPM}$, $S=30^\circ$
Cone-CB (case III) $\dot{m}=75$-200-500LPM, $S=45^\circ$

Cone-recess (case V) $\dot{m}=75$-200-500LPM, $S=45^\circ$

Ext-CB (case II) $\dot{m}=75$-200-500LPM, $S=45^\circ$
Ext-recess (case IV) $\dot{m}=75-200-500\text{LPM}$, $S=45^\circ$

Ref (case I) $\dot{m}=75-200-500\text{LPM}$, $S=45^\circ$

Cone-CB (case III) $\dot{m}=75-200-500\text{LPM}$, $S=60^\circ$
Cone-CB (case III) $\dot{m}=75-200-500$ LPM, $S=60^\circ$

Ext-CB (case II) $\dot{m}=75-200-500$ LPM, $S=60^\circ$
Ext-recess (case IV) $\dot{m}$=75-200-500LPM, $S=60^\circ$

Ref (case I) $\dot{m}$=75-200-500LPM, $S=60^\circ$

Figure A1. Case I-V at 75, 200 and 500 LPM, $S=0^\circ$, 30°, 45° and 60°
Appendix B

Theoretical and Calculated Swirl Strength

Theoretical Swirl Strength:

\[ S = \frac{axial \ flux \ of \ angular \ momentum}{axial \ thrust \cdot R} = \frac{G_{ang}}{G_x \cdot R} \]

\[ = \frac{\int_0^R U W r^2 dr}{R \int_0^R U^2 r^2 dr} \ldots \ldots (1) \]

Where \( U, W \) are the mean axial and circumferential velocities, \( r \) is the radial coordinate and \( R \) is the tube radius.

After the derivation;

Theoretical swirl strength is obtained by;

\[ S = 2 \cdot x \cdot \frac{1-(d_h-d)^3}{1-(d_h-d)^2} x tan(\theta) \ldots (2) \]

Where \( d_h \) and \( d \) are the diameter of centerbody and swirler [100], respectively. While, \( \theta \) is the swirl vane angle. From equation (2), swirl strength is calculated by the burner geometry.
Calculated Swirl Strength:

This process has been conducted by integrating the area under the curve, as seen in the example figure below (Figure B). Laser Doppler Velocimetry (LDV) has been used to obtain the axial velocity data, from which the figure is created.

![Figure B. Mean velocities of LDV, axial velocity (blue) and radial velocity (red)](image)

<table>
<thead>
<tr>
<th>Swirl vane angle</th>
<th>Calculated</th>
<th>Measured</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>30°</td>
<td>0.39</td>
<td>0.44</td>
</tr>
<tr>
<td>45°</td>
<td>0.71</td>
<td>0.77</td>
</tr>
<tr>
<td>60°</td>
<td>1.16</td>
<td>1.30</td>
</tr>
</tbody>
</table>