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Burnout, NO, Flame Temperature, and Radiant Intensity from Oxygen-Enriched Combustion of a Hardwood Biomass

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Brigham Young University

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Increasing concern for energy sustainability has created motivation for the combustion of renewable, CO$_2$ neutral fuels. Biomass co-firing with coal provides a means of utilizing the scaled efficiencies of coal with the lower supply availability of biomass. One of the challenges of co-firing is the burnout of biomass particles which are typically larger than coal but must be oxidized in the same residence time. Larger biomass particles also can increase the length of the radiative region and alter heat flux profiles. As a result, oxygen injection is being investigated as a means of improving biomass combustion performance.

An Air Liquide designed burner was used to investigate the impact of oxygen enrichment on biomass combustion using two size distributions of ground wood pellets (fine grind 220 µm and medium grind 500 µm mass mean diameter). Flame images were obtained with a calibrated RGB digital camera allowing a calculation of visible radiative heat flux. Ash samples and exhaust NO were collected for a matrix of operating conditions with varying injection strategies. The results showed that oxygen can be both beneficial and detrimental to the flame length depending on the momentum of the oxygen jet.

Oxygen injection was found to improve carbon burnout, particularly in the larger wood particles. Low flow rates of oxygen enrichment (2 to 6 kg/hr) also produced a modest increase in NO formation up to 30%. The results showed medium grind ~500 µm mass mean diameter particle combustion could improve LOI from 30% to 15% with an oxygen flow rate of 8 kg/hr. Flame images showed low flow rates of O$_2$ (2 kg/hr) in the center of the burner with the fine particles produced a dual flame, one flame surrounding the center oxygen jet and a second flame between the volatiles and secondary air. The flame surrounding the center oxygen jet produced a very high intensity and temperature (2100 K). This center flame can be used to help stabilize the flame, increase devolatilization rates, and potentially improve the trade-off between NO and burnout.
ACKNOWLEDGEMENTS

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# TABLE OF CONTENTS

List of Tables ................................................................................................................................ vi
List of Figures ................................................................................................................................... vii
Nomenclature ................................................................................................................................... ix

1 Introduction ................................................................................................................................... 1
   1.1 Formation of Nitric Oxides ............................................................................................. 2
   1.2 Swirl Stabilized Burner ............................................................................................... 4

2 Literature Review ......................................................................................................................... 6
   2.1 Enriched Air Combustion (EAC) ................................................................................ 7
   2.2 Oxygen Injection Combustion (OIC) .......................................................................... 10

3 Experimental Setup and Method .................................................................................................. 11
   3.1 BYU Combustion Facility ............................................................................................ 11
   3.2 Fuel Analyses ................................................................................................................ 13
   3.3 Burner Configurations and Operating Conditions ...................................................... 13
   3.4 Flame Image Capturing and Processing .................................................................... 15
      3.4.1 Flame Temperature .............................................................................................. 16
      3.4.2 Flame Intensity .................................................................................................... 17
   3.5 NO Measurements ........................................................................................................ 19
   3.6 Ash Collection and Loss on Ignition .......................................................................... 19

4 Results and Discussion .............................................................................................................. 21
   4.1 Flame Images ................................................................................................................ 21
   4.2 Flame Intensity ............................................................................................................. 26
   4.3 Flame Temperature ....................................................................................................... 32
   4.4 Loss on Ignition (LOI) ............................................................................................... 36
   4.5 Nitric Oxide (NO) ....................................................................................................... 39
5 Summary and Conclusions................................................................................................................. 44
References................................................................................................................................................... 47
APPENDIX A: BYU Burner Swirl............................................................................................................... 50
LIST OF TABLES

Table 3.1: Proximate and ultimate analysis (as received), heating value, and particle size of the two deliveries of hardwood biomass.................................................................13

Table 3.2: Test matrix of burner geometry configurations, flow rates, and condition nomenclature ..................................................................................................................15

Table 4.1: Representative flame images for all operating conditions using fine particles. Images represent 19 cm vertically .................................................................23

Table 4.2: Comparison of the same image at 2 kg/hr O2, FP/lg/cen burner configuration. ....24

Table 4.3: Representative flame images for all operating conditions using medium particles. .................................................................................................................................26

Table A.1: Selected dimensions of BYU swirl burner ..........................................................53

Table A.2: Measured values of $\xi$ .........................................................................................55

Table A.3: Multiple rod rotations matched with corresponding swirl numbers. ...............56
LIST OF FIGURES

Figure 1.1: Reaction mechanism converting fuel nitrogen to NO and N2 ................................................. 4

Figure 1.2: Burner cross section schematic demonstrating the effect of swirl on combustion dynamics. ................................................................. 5

Figure 3.1: Schematic diagram of the Burner Flow Reactor (BFR) and supporting equipment ................................................................. 12

Figure 3.2: Half the cross section of the Air Liquide pipe in pipe burner showing locations of 4 concentric flows ................................................................. 14

Figure 4.1: Intensity [W/m\(^2\)-sr-µm] vs distance from burner exit [cm] showing fluctuations of flame intensity for air case and 8 kg/hr oxygen injection. (a) fine particles, (b) medium particles ................................................................. 28

Figure 4.2: Intensity [W/m\(^2\)-sr-µm] vs. distance from the burner [cm] of three burner configurations: (a) FP/sm/cen*, (b) FP/sm/cen, and (c) FP/lg/cen ................................................................. 30

Figure 4.3: Intensity [W/m\(^2\)-sr-µm] vs. distance from the burner exit [cm] for configurations with fine particle size: (a) air combustion, (b) 2 kg/hr O\(_2\) addition, (c) 8 kg/hr O\(_2\) addition, (d) 18 kg/hr O\(_2\) addition ................................................................. 31

Figure 4.4: Intensity [W/m\(^2\)-sr-µm] vs. distance from burner exit [cm] for medium particles. (a) Air condition (b) 8 kg/hr oxygen addition ................................................................. 32

Figure 4.5: Temperature [K] vs. distance from the burner [cm] for fine particle configurations: (a) FP/sm/cen*, (b) FP/sm/cen, (c) FP/lg/cen ................................................................. 34

Figure 4.6: Temperature [K] vs. distance from the burner exit [cm]. (a) air only (b) 8 kg/hr Oxygen addition ................................................................. 36

Figure 4.7: LOI [%] vs. O\(_2\) flow rate rate [kg/hr] for different particle sizes under burner configurations 2, 3, 4 and 6 (see Table 3.2) ................................................................. 37

Figure 4.8: LOI [%] vs. O\(_2\) flow rate [kg/hr] comparing oxygen addition location ................................................................. 38

Figure 4.9: LOI [%] vs. O\(_2\) flow rate [kg/hr] comparing the effects of changing center tube diameter size ................................................................. 39

Figure 4.10: LOI [%] vs. O\(_2\) flow rate [kg/hr] for all fine particle configurations ................................................................. 39

Figure 4.11: NO [ppm] vs. O\(_2\) flow rate [kg/hr] for different particle sizes under burner configurations 2, 3, 4 and 6 (see Table 3.2) ................................................................. 40

Figure 4.12: NO [ppm] vs. O\(_2\) flow rate [kg/hr] for all medium particle configurations ................................................................. 41

Figure 4.13: NO [ppm] vs. O\(_2\) velocity [m/s] for all medium particle conditions ................................................................. 42
Figure 4.14: NO [ppm] vs. O2 flow rate [kg/hr] for all fine particle configurations..................43

Figure A.1: Burner cross section schematic ..................................................................................51

Figure A.2: Top view of swirl plate blocks. The smaller blocks rotate to change the swirl number. ..................................................................................................................................51

Figure A.3: Linear curve fit of $\xi$ to the rotations of the threaded rod. ......................................55
### NOMENCLATURE

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Definition</th>
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<tr>
<td>AL</td>
<td>Air Liquide</td>
</tr>
<tr>
<td>ASTM</td>
<td>American Society for Testing and Materials</td>
</tr>
<tr>
<td>BFR</td>
<td>Burner Flow Reactor</td>
</tr>
<tr>
<td>BYU</td>
<td>Brigham Young University</td>
</tr>
<tr>
<td>CO₂</td>
<td>Carbon Dioxide</td>
</tr>
<tr>
<td>LOI</td>
<td>Loss on Ignition</td>
</tr>
<tr>
<td>K</td>
<td>Kelvin</td>
</tr>
<tr>
<td>g/hr</td>
<td>Grams Per Hour</td>
</tr>
<tr>
<td>kg/hr</td>
<td>Kilograms Per Hour</td>
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<tr>
<td>kW&lt;sub&gt;th&lt;/sub&gt;</td>
<td>Kilowatts of Thermal Power</td>
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<tr>
<td>m</td>
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</tr>
<tr>
<td>mm</td>
<td>Millimeters</td>
</tr>
<tr>
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</tr>
<tr>
<td>μs</td>
<td>Microseconds</td>
</tr>
<tr>
<td>ND-IR</td>
<td>Non-Dispersive Infrared</td>
</tr>
<tr>
<td>NO</td>
<td>Nitric Oxide</td>
</tr>
<tr>
<td>ppm</td>
<td>Parts Per Million</td>
</tr>
<tr>
<td>W&lt;sub&gt;Dry&lt;/sub&gt;</td>
<td>Weight of Ash with Moisture Removed</td>
</tr>
<tr>
<td>W&lt;sub&gt;CFA&lt;/sub&gt;</td>
<td>Weight of Carbon Free Ash</td>
</tr>
<tr>
<td>wt%</td>
<td>Weight Percent</td>
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<tr>
<td>Y&lt;sub&gt;Ash&lt;/sub&gt;</td>
<td>Mass Fraction of Ash</td>
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1 INTRODUCTION

According to the Environmental Protection Agency, in 2010 84% of greenhouse gases emitted from human activities in the United States were CO\textsubscript{2} [1]. There is a growing concern that these emissions are causing an imbalance of carbon in the atmosphere resulting in multiple unnatural effects including global warming. Fossil fuel combustion used in the generation of electricity is the most prevalent source of CO\textsubscript{2} emissions in the country [1]. One of the most inexpensive ways to reduce net CO\textsubscript{2} emissions from coal-based power plants is to co-fire biomass [2], a carbon neutral fuel, which often produces an added benefit of decreased NO\textsubscript{x} and SO\textsubscript{x} pollutants as well. One of the challenges associated with co-firing is the energy expensive process of pulverizing biomass to the size of pulverized coal. As a result, it is common practice in industry for biomass particles to be larger than coal particles which leads to concerns of flame stability and carbon burnout. Larger particles in a pulverized fuel flame take longer to heat up, ignite, and release their volatiles causing increased lift off of the flame from the burner exit and inconsistency in the flame. Larger particles also require longer residence times in the furnace in comparison to the smaller coal particles to produce equivalent particle burnouts. Other issues involved in replacing fossil fuels with biomass are combustion efficiency and load reduction.

Studies have shown that oxygen enrichment during combustion can increase particle burnout and mass loss rates of biomass fuels [3][4][5]. Oxygen addition also produces higher temperatures and increased amounts of soot resulting in a higher visible intensity which can
increase the radiant heat transfer[6][7]. While increasing oxygen concentration may increase particle burnout, this increase in oxygen can negatively impact NO formation. Bool et al. [8][9] studied several methods of oxygen injection into coal flames and concluded that NO could be reduced by certain methods of oxygen injection.

This work uses selective oxygen injection to study the effects of high localized oxygen concentrations on the combustion of a hardwood biomass. The variables measured to evaluate the characteristics of this oxygen injection combustion include carbon burnout, NO\textsubscript{x} emissions, flame temperature, and flame intensity. Several burner configurations and conditions will be evaluated that employ various combustion dynamics while two particle sizes of biomass will be tested.

1.1 Formation of Nitric Oxides

The formation of nitric oxides in biomass and coal combustion generally follow three chemical paths often called thermal, prompt, and fuel NO\textsubscript{x}. Thermal NO\textsubscript{x} is formed via the extended Zeldovich mechanism shown in equation 1.1 through 1.3 [10]. NO\textsubscript{x} formed by the thermal mechanism is generally not significant unless gas temperatures exceed 1800K. Although the results discussed later will show flame temperatures above 1800K the formation of NO\textsubscript{x} by this mechanism is slow and is considered to be formed in the gases after the flame which did not exceed this temperature limit [10].

\begin{align*}
O + N_2 & \rightleftharpoons NO + N \quad (1.1) \\
N + O_2 & \rightleftharpoons NO + O \quad (1.2) \\
N + OH & \rightleftharpoons NO + H \quad (1.3)
\end{align*}
Prompt NO\textsubscript{x} is formed by the Fenimore mechanism shown in equation 1.4 and 1.5\textsuperscript{[10]}. This chemical path is properly named due to the quick rate at which the NO\textsubscript{x} is formed. In this situation hydrocarbon radicals attack the diatomic nitrogen in the air creating cyanides. The rest of the reaction from the cyanides to NO\textsubscript{x} follows the same pathway that will be discussed with fuel NO\textsubscript{x}.

\[
CH + N_2 \Leftrightarrow HCN + N \quad (1.4)
\]

\[
C + N_2 \Leftrightarrow CN + N \quad (1.5)
\]

Generally thermal or fuel NO\textsubscript{x} dominates in pulverized fuel combustion, and since the gas temperatures in this work are not substantial, NO\textsubscript{x} formed from fuel nitrogen will be the main focus. Fuel nitrogen refers to the nitrogen in the molecular structure of the biomass or coal. The nitrogen is typically released from biomass and coal as ammonia (NH\textsubscript{3}) and hydrogen cyanide (HCN) respectively. The path to take HCN to NO is shown in the sequence of reactions in equations 1.6 through 1.9 \textsuperscript{[10]}.

\[
HCN + O \Leftrightarrow NCO + H \quad (1.6)
\]

\[
NCO + H \Leftrightarrow NH + CO \quad (1.7)
\]

\[
NH + H \Leftrightarrow N + H_2 \quad (1.8)
\]

\[
N + OH \Leftrightarrow NO + H \quad (1.9)
\]

This mechanism standing alone suggests that all of the nitrogen from the fuel is converted into NO\textsubscript{x} but this is not the case. Figure 1.1 maps a reaction mechanism for converting fuel nitrogen into NO and N\textsubscript{2}. It can be seen in this figure that if NO is introduced to atomic nitrogen instead of hydroxide (OH) or O\textsubscript{2} then the NO can be reduced to diatomic nitrogen. This can be done more often in fuel rich conditions where less oxygen is available. It is partially because of
this fact that coal and biomass are often burned in swirl stabilized burners like the one used herein. The next section will discuss the details of how fuel rich zones are formed in swirl stabilized burners to reduce fuel NO$_x$.

![Figure 1.1: Reaction mechanism converting fuel nitrogen to NO and N2 [11]](image)

1.2 Swirl Stabilized Burner

Adding swirl to a burner has a number of effects on the combustion dynamics of a flame including altered mixing patterns and shortened flame lengths. Figure 1.2 demonstrates the effect of adding swirl to the secondary or outer air channel. When swirled air exits the tube the momentum from the swirling motion causes the air to move outward. This outward movement creates a void directly under the burner which draws combustion products back towards the burner. This combined motion creates two pockets where the fuel and other burner inlets are held momentarily and mixed together. These pockets are generally fuel rich because the air is creating the swirling motion on the outside. These fuel rich zones as discussed previously are reduction zones for NO$_x$ where it is converted to N$_2$. Different amounts of swirl will affect the size and shape of the fuel rich zones and therefore there is an ideal amount of swirl to generate the least amount of NO$_x$. Swirl also affects the carbon burnout by determining the amount of mixing between fuel and the oxidizer and the residence time of individual particles inside the reactor. Higher swirl amounts generally allow the fuel and oxidizer to be well mixed producing increased
burnout while lower swirl amounts create larger fuel rich pockets more affectively reducing NO$_x$. In this work oxygen will be added to the center of the burner to slightly increase the oxygen concentration within the fuel rich pocket. The theory is that this oxygen may add an early heat source aiding the release of volatiles from the fuel so that the NO$_x$ will have more time to be reduced and that the burnout process will happen more quickly. Several strategies varying oxygen injection location, velocity, and overall amounts are applied in this investigation and discussed in section 3.3.

![Burner cross section schematic demonstrating the effect of swirl on combustion dynamics.](image)

**Figure 1.2:** Burner cross section schematic demonstrating the effect of swirl on combustion dynamics.
2 LITERATURE REVIEW

Oxygen enhanced combustion is used in many different applications where the benefits of additional oxygen are needed like higher thermal efficiencies and improved flame characteristics[7]. There are three main types of oxygen enhanced combustion. The first type, oxy-fuel combustion (OFC), requires all of the oxidizer for a boiler to be provided by the cryogenic separation of oxygen and nitrogen in air which is an expensive and energy intensive retrofit technology for existing boilers. Most of these applications use CO$_2$ or recycled flue gas as a diluent to keep equipment from melting and to increase the flow rate of the hot gas to maintain convective heat transfer. Reviews on this technology have been written by Buhre et al. [12], Toftegaard et al. [13], and Wall et al. [14].

The second type, enriched air combustion (EAC), includes applications where the concentration of oxygen in the secondary air is increased, typically between 21% and 50% oxygen. This method requires a smaller fraction of the total oxidizer to be supplied by oxygen and is therefore less expensive to implement and operate than OFC. The third type, oxygen injection combustion (OIC) is when oxygen is added to a specific location relative to the flame via a lance either originating within or external to the burner. This method typically utilizes the least amount of oxygen with the intent of stabilizing a flame, lowering emissions, or improving ignition. While this work will focus on OIC, results from the other forms of oxygen enhanced combustion will be beneficial to understanding trends in OIC.
This work will investigate oxygen injection in a biomass fueled, swirl stabilized burner including images, temperature profiles, visible flame intensity, NO\textsubscript{x}, and burnout for specific injection strategies focused on injection velocity and flow rate. Two particle sizes of hard wood biomass will be investigated. These data are unique because they are the first reported for biomass particles of various size and because of the comprehensive nature of the data obtained which includes flame images, calibrated axial intensity measurements, axial flame temperature, effluent NO\textsubscript{x} and particle burnout.

2.1 **Enriched Air Combustion (EAC)**

The studies discussed in this section involve enriched air combustion where oxygen was added to the secondary air prior to entering the burner. It is evident in EAC that the temperature of the flame increases as a result of the oxygen enriched atmospheres\[15][16][6]. This is due to the decreased amount of N\textsubscript{2} diluents present in the flame allowing more energy to be used to raise the temperature rather than heat up the N\textsubscript{2}. The higher flame temperatures also produce higher wall temperatures.

Joshi and Becker \[15\] recorded temperatures of an oxygen enhanced natural gas flame in a 5.8 MW\textsubscript{th} water cooled reactor using a suction pyrometer. They reported that increasing the oxygen concentration from 21% to 35% produced increased furnace temperatures, increased NO\textsubscript{x} emissions, and decreased furnace wall temperature uniformity. It was also stated in their work that lower levels of oxygen enrichment could be used to increase the productivity of the furnace or heat flux without significantly changing flame length or shape but that higher levels of oxygen enrichment would require an evaluation of the effects on the flame characteristics.

It is well known that soot concentration and temperature are the main contributors to the luminous intensity of a flame. Santos et al. measured soot concentration by laser extinction in
oxygen enriched flames of acetylene [17] and natural gas [18] using a small co-flow burner 100 mm in diameter. In both of these studies he found OEC to increase soot formation, heat transfer by radiation and flame luminosity while varying firing rates and oxygen concentrations. He also noted that there were less overall NOx emissions from the oxygen enriched conditions than there were from the air conditions.

Other investigations using EAC include experiments of small scale laminar flow particle streams or single particle studies. Bejarano and Levendis [19] burned coal chars in a drop tube furnace at a rate of 2 g/hr in an oxygen enriched environment. Using two color pyrometry they measured particle temperatures and intensities while the furnace was electrically heated to 1300 - 1500 K and oxygen concentrations were 21%, 50%, and 100%. They found that this increase in oxygen raised particle temperatures up to 45%, decreased combustion durations up to 87%, and also increased the apparent reactivity of the particles. Fan et al.[20] burned a Chinese bituminous coal in a drop tube furnace at a rate of 1.16 kg/hr concluding that the combustion properties of the fuel greatly improved with the addition of oxygen. Specifically, it was reported that there were significant decreases in ignition temperature with increasing oxygen concentration. Ignition temperature in the air condition was reported to be around 800 °C while increasing oxygen concentration to 40% decreased the ignition temperature by about 34% and an oxygen concentration of 80% reduced ignition temperature by 56%.

Hu et al.[21] used what they termed the pulse technique to study coal-char combustion with a mixture of nitrogen and oxygen as the pulse gas. They reported that the reaction rate for each of the coal chars was sensitive to the concentration of oxygen in the pulse gas. Momeni et al. [22] studied air enriched combustion of a biomass pine wood in a single particle reactor rig. They concluded that intra-particle temperature gradients were significant factors to the speed of
devolatilization and burnout when dealing with larger particles. It was also clear that increasing oxygen concentration lowered ignition and devolatilization temperatures and increased the rate at which the char was burned out. It was also stated that oxygen concentration had a greater effect at lower oxidizer temperatures.

Murphy and Shaddix [23] studied oxygen enriched combustion of coal chars in a Hencken burner at Sandia National Laboratories. Using a particle sizing pyrometer they conclude that increasing oxygen concentration increased char combustion temperature and reduced the char burnout time. Shaddix and Molina [24] also used a Hencken burner and compared enriched air combustion with oxy-fuel combustion. They found that higher oxygen concentrations lowered the temperature required for particle burnout. It was also noted that particle devolatilization proceeded more rapidly with higher oxygen concentration but decreased with oxygen in carbon dioxide because of lower diffusion rates of oxygen and fuel volatiles.

Daood et al. [4] performed a unique study burning coal in a 20 kWth reactor using staged combustion. During their investigations oxygen was added to the secondary air and/or to the over fire inlet further downstream in the reactor. Typically air staging is not effective for improving burnout however, Daood found that air staging with oxygen enrichment improved burnout from 97.2% with air to 98.3% with 25% oxygen concentration and 98.7% with 35% oxygen. Measurable decreases in NOx could also be accomplished by varying the stoichiometric ratio in different locations of the reactor.

Luo et al.[3] investigated air enriched combustion of a very small particle biomass they called micro fuel in a thermogravimetric analyzer and cyclone furnace 900 mm in height and a diameter of 400 mm. Conclusions from this work indicate oxygen concentration to be optimized
around 30 to 40%. Oxygen concentration increases mass loss rates and temperatures while lowering ignition, volatile release, and burnout temperatures.

### 2.2 Oxygen Injection Combustion (OIC)

Marin et al. [25] performed an investigation using oxygen injection combustion in a 1.5 MW boiler using Illinois coal focused on NO\textsubscript{x} and carbon burnout. Also, in this study data were shown suggesting around 30% oxygen enrichment to be optimal for reducing time to complete devolatilization and char oxidation. Many details of the location of oxygen injection within the burner are not provided; however, they were able to decrease loss on ignition and NO formation to 60% and 72%.

Bool et al. [9] [8] investigated oxygen injection for pulverized coal in a 17 kW, a 1.5 MW, and a 7.3 MW reactor. The investigations performed in this study involved staged air with oxygen injection strategies being implemented in a staged burner. Staging creates a fuel rich region near the burner where NO is reduced to N\textsubscript{2} but is detrimental to burnout because it delays char oxidation. Adding oxygen while remaining overall fuel rich near the burner increased temperatures near the burner aiding burnout, but temperatures remained below the limit for thermal NO formation and burner failure. Bool et al. found that adding neat oxygen injection to the fuel rich stream reduced NO\textsubscript{x} by up to 45% of the air only case and improved burnout. They also experimented with different lance designs stating the lance design causing early premixing of fuel and oxygen resulted in higher flame temperatures near the burner.
3 EXPERIMENTAL SETUP AND METHOD

This section provides details of the equipment and facilities used to performed the experiments in this work. Other information recorded in this section include fuel analysis, equipment calibrations, and experimental methods.

3.1 BYU Combustion Facility

Figure 3.1 shows a schematic of the BYU combustion facility representing the major pieces of equipment used in this work. Three types of data were collected including flame images, gas samples, and ash samples. Flame images were taken using an RGB camera discussed in section 3.4. Gas samples were analyzed using a Horiba gas analyzer measuring O₂ (galvanic cell), CO₂ (ND-IR), CO (ND-IR), and NOₓ (chemiluminescence). The ash samples were collected in a post combustion cyclone. The method for analyzing the ash will be discussed in detail in section 3.6.

The BFR is a down-fired, cylindrical, 150 kW th reactor consisting of six 0.4 m long cylindrical sections with a total height of 2.4 m, an inside diameter of 0.75 m. Each section is refractory lined on the inside, insulated around the refractory and surrounded by water cooled walls on the outside. Each section contains four access ports 90 degrees apart allowing optical and probe access over a large portion of the axial profile. The access port is a rectangle with a vertical height of 28 cm and a horizontal width of 9 cm. Neat oxygen was stored in a 365 liter
liquid dewar and was supplied to the BFR through a series of safety regulated flow meters and pressure gauges documented by Zeltner [26]. The fuel was delivered to the reactor via a bulk bag unloader and gravimetric, computer-controlled, loss-in-weight, dual auger feeder. A water barrel sits at the base of the reactor to catch slag and other large deposits dropped by the reactor. An Ingersol Rand Compressor supplied primary and secondary air. The burner, which will be discussed in more detail in the operating conditions section, is a pipe in pipe design supplied by Air Liquide. After passing the water barrel the exhaust enters a cyclone where residual particles larger than 2 µm are collected in an ash barrel. The exhaust then continues through the exhaust fan and out the stack.

![Diagram of Burner Flow Reactor (BFR) and supporting equipment.](image)

**Figure 3.1:** Schematic diagram of the Burner Flow Reactor (BFR) and supporting equipment.
3.2 Fuel Analyses

Two types of hardwood biomass fuels of different particle sizes were used in this work. The proximate and ultimate analyses of the fine and medium particles are displayed in Table 3.1. Although the fuels are both hardwood biomass, the fine particles originated in the southeastern U.S. while the larger particles originated in Wisconsin. The two fuels are therefore not from the same biomass stock. Nevertheless, the properties are very similar although there are small differences in moisture content, volatiles, and fixed carbon.

Table 3.1: Proximate and ultimate analysis (as received), heating value, and particle size of the two deliveries of hardwood biomass

<table>
<thead>
<tr>
<th></th>
<th>Fine Particles</th>
<th>Medium Particles</th>
</tr>
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<tbody>
<tr>
<td>Moisture (wt.%)</td>
<td>5.83</td>
<td>5.28</td>
</tr>
<tr>
<td>Ash (wt.%)</td>
<td>0.54</td>
<td>0.30</td>
</tr>
<tr>
<td>Volatiles (wt.%)</td>
<td>76.42</td>
<td>79.06</td>
</tr>
<tr>
<td>Fixed Carbon (wt.%)</td>
<td>17.21</td>
<td>15.36</td>
</tr>
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</table>

<table>
<thead>
<tr>
<th>Ultimate (as received, wt.%)</th>
<th>Fine Particles</th>
<th>Medium Particles</th>
</tr>
</thead>
<tbody>
<tr>
<td>H (wt.%)</td>
<td>5.36</td>
<td>5.40</td>
</tr>
<tr>
<td>C (wt.%)</td>
<td>49.87</td>
<td>49.85</td>
</tr>
<tr>
<td>N (wt.%)</td>
<td>0.32</td>
<td>0.26</td>
</tr>
<tr>
<td>S (wt.%)</td>
<td>0.10</td>
<td>0.09</td>
</tr>
<tr>
<td>O (wt.%)</td>
<td>37.98</td>
<td>38.82</td>
</tr>
<tr>
<td>HHV (kJ/kg)</td>
<td>17,638</td>
<td>17,463</td>
</tr>
<tr>
<td>Mean Size (µm)</td>
<td>224</td>
<td>500</td>
</tr>
</tbody>
</table>

3.3 Burner Configurations and Operating Conditions

The burner used in this work was a pipe in pipe design supplied by Air Liquide. This design allowed four concentric flows to enter the reactor simultaneously. Figure 3.2 shows half the cross section of the Air Liquide burner with four concentric flows numbered horizontally across the channels. Flow channel 1 represents the center tube of the burner through which neat
oxygen and occasionally air were injected. The exit of this channel had a variable diameter allowing the injected flows to have varying exit velocities. Two diameters are used in this work for channel 1 and are referred to as the small and large diameter. Flow channel 2 is referred to as the annulus and was also used for neat oxygen or air flows. Whenever air was supplied through channel 2 or through channel 1 it was done so at a rate of 2 kg/hr but oxygen flow rates varied in these channels. Flow channel 3 was the primary fuel/air channel. Flow rates were held constant at 19 kg/hr air and 27 kg/hr biomass. This flow channel has the option of holding a 30° swirl vanes that extend 10 cm from the burner exit into the flow channel. Flow channel 4 contained the secondary air at a baseline flow rate of 160 kg/hr passing through 45° swirl vanes extending 10 cm into the channel. On the outside of flow channel 4 was a water cooled quarl that tapers away from the centerline of the reactor downstream of the burner exit.

Figure 3.2: Half the cross section of the Air Liquide pipe in pipe burner showing locations of 4 concentric flows

A fixed set of burner hardware and locations for oxygen injection will be referred to as burner configuration. For each configuration, various flow rates were swept by changing the flow in a channel from 2 kg/hr air to various flow rates of oxygen ranging from 2 – 18 kg/hr. The
matrix of operating conditions are summarized in Table 3.2. This sweep of oxygen flow rates was studied for three different burner geometries, two oxygen addition locations, and two particle sizes.

There are three variables that specify a burner configuration: center tube diameter, \( \text{O}_2 \) addition location, and primary swirl. For each burner configuration there can be two fuel sizes and a range of air or oxygen flow rates. Fuel size is designated as medium (MP) and fine particle (FP). The burner configuration is designated by small (sm) and large (lg) center tube diameters, and center (cen) and annular (ann) oxygen addition. Swirled primary air is designated by an asterisk “*”. For example, a configuration labeled MP/lg/ann indicates medium particle size (MP), large center tube diameter (lg), and oxygen addition through the annular (ann) tube. Only one condition with the fine particles utilizes swirl for the primary annulus (flow channel 3) and all conditions have swirl vanes on the secondary air.

**Table 3.2: Test matrix of burner geometry configurations, flow rates, and condition nomenclature**

<table>
<thead>
<tr>
<th>Burner Configuration</th>
<th>Abbreviation</th>
<th>Flow rates (kg/hr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>FP/sm/cen*</td>
<td>2(air); 2, 8, 18 (( \text{O}_2 ))</td>
</tr>
<tr>
<td>2</td>
<td>FP/sm/cen</td>
<td>2(air); 2, 8, 18 (( \text{O}_2 ))</td>
</tr>
<tr>
<td>3</td>
<td>FP/lg/cen</td>
<td>2(air); 2, 8, 18 (( \text{O}_2 ))</td>
</tr>
<tr>
<td>4</td>
<td>MP/lg/cen</td>
<td>2(air); 8 (( \text{O}_2 ))</td>
</tr>
<tr>
<td>5</td>
<td>MP/lg/ann</td>
<td>2(air); 2, 4, 6, 8 (( \text{O}_2 ))</td>
</tr>
<tr>
<td>6</td>
<td>MP/sm/cen</td>
<td>2(air); 2, 4, 6, 8 (( \text{O}_2 ))</td>
</tr>
</tbody>
</table>

*Condition with swirl vanes on the primary fuel burner inlet

### 3.4 Flame Image Capturing and Processing

Flame images were captured using a UNIQ, UC-600CL, 10-bit, RGB, CCD camera operated with XCAP version 3.7 software. This camera and calibration was previous used and details can be found in Draper [27] The images were taken from the three upper access ports on the north side of the reactor. These ports vertically span distances of 0.05-0.33 m, 0.45-0.73 m,
and 0.85-1.13 m from the burner exit. Typically a set of 40-100 images were taken at a given operating condition with a significant amount of variability from one image to the next caused by the turbulent nature of the flame. More images were captured from flames that appeared visually to be more inconsistently luminous. This was particularly true in the lower access ports. At windows where the flame was more consistent, typically the top port, 40 images were captured with 100 milliseconds between each image. At windows where the flame was more inconsistent, typically the lower two ports, 100 images were taken in 5, 20 image periods with 100 milliseconds between individual images and 1-2 minutes between each set of images.

### 3.4.1 Flame Temperature

Temperatures for every pixel in a particular flame image were calculated using the red and blue pixel response from the digital camera and two-color pyrometry. Equation 3.1, referenced from Draper [27], shows the relationship between the pixel count, emissivity, and temperature of the flame. For details on the derivation of this relationship see the work by Draper [27].

\[
P_l = s_i \int_{t_1}^{t_2} \int_{\lambda_1}^{\lambda_2} \left[ \frac{C_1}{\pi \lambda^5 \left( \exp \left( \frac{C_2}{\lambda T} \right) - 1 \right)} \right] \left[ 1 - \exp \left( -\frac{K L}{\lambda^a} \right) \right] \zeta_\lambda \beta_\lambda d\lambda dA dt \tag{3.1}
\]

\( P_l \) represents an integer pixel response of a color in the range of 0 to 1023. The pixel sensitivity, \( s_i \) is a constant for each color calibrated using black body cavity radiation at a known temperature. The first quantity in the square brackets represents the emissive power from the flame where \( C_1 \) and \( C_2 \) are Plank’s constants (Incropera et al. [28]), \( \lambda \) is the wavelength over which the detector receives energy from the flame, and \( T \) is the temperature. The second set of square brackets represents the Hottel and Broughton model of the emissivity. In this model \( K \) is...
an extinction constant and \( L \) is the distance across the flame through which the pixel is collecting data. The parameter “KL” is the optical thickness, and is proportional to the soot mass within the measurement line of sight. The expression is from an empirical model of soot absorption by Hottel and Broughton [29] where \( \alpha \) was measured to be 1.39 in the visible region. In some cases a neutral density filter was used to keep the flame from saturating the detector. The transmittance of the filter is given by \( \zeta_\lambda \). These filters were not used in this work. The spectral response, \( \beta_\lambda \), of the camera sensor was assumed to be as provided by the manufacturer. All of these values are then integrated over wavelength, area, and time. After integrating it was found that the pixel count increased linearly with both time and aperture area.

The MATLAB code used to solve for temperature and emissivity used a function called “fsolve.” Two equations, one for the red pixel response and one for the blue pixel response, were organized while MATLAB function “trapz” approximated the integrals. “fsolve” used a guess and check method to solve for what values correctly solved both equations for temperature and KL. Because this solving technique was time intensive a smaller set of images were processed for temperature data. The images were selected by inspecting all 40-100 images and selecting 5 images that were representative of the whole. Even though this process was subjective it is useful in providing a general description of the flame.

### 3.4.2 Flame Intensity

A calibrated luminosity was also collected from the camera in order to produce a quantitative radiative flux. Using the cameras pixel response in the range of the wavelengths associated with the red color (550-750 nm) the Table 4.3 emissive power measurement was used to calculate an average intensity over the red spectrum. Solving for emissive power was accomplished by rearranging Equation 3.1 into Equation 3.2.
\[ \bar{E}_\lambda = \frac{P_i}{s_i \beta_\lambda A \lambda t} \left[ \frac{W}{m^2 \mu m} \right] \]  

(3.2)

\( \bar{E}_\lambda \) is the average spectral emissive power and \( \beta_\lambda \) is the average spectral response over the indicated wavelengths. After solving for the spectral emissive power the spectral intensity can be found by dividing by pi as shown in Equation 3.3.

\[ \bar{I} = \frac{\bar{E}}{\pi} \]  

(3.3)

\( \bar{I} \) in this case is the average intensity over the visible red spectral band of the camera. While this intensity is not indicative of the total radiative heat transfer emitted from the flame it is an indicator of radiation produced by a combination of temperature and soot which dominates in the near flame region.

This method was tested to ensure that data measured and recorded as average intensity was reasonable based on the emissivity calculated from the KL measurement and temperatures predicted from two-color pyrometry. To accomplish this an image was taken from the configuration FP/sm/cen* and an area of the flame from that image was selected for examination. From that area an average temperature (1679 K) and emissivity (.0725) were produced by processing the digital image. The intensity calculated from Equation 4.3 was (125 W/m²-sr-µm). Using the measured emissivity and temperature at the red color wavelength of .630 mm, the intensity calculated from Wien’s Equation, 3.4 was within 14%.

\[ I_b = \frac{C_1}{\pi \lambda^5 (e^{C_2 \lambda} - 1)} \]  

(3.4)

The method to process intensity data is much less time intensive than processing temperature due to the solver applied to the temperature relationships. Because of this intensity data was processed for all of the existing images captured at each location instead of the 5 representative images chosen for temperature data.
3.5 NO Measurements

A continuous flow of the exhaust gas was sampled near the reactor exit shown in Figure 3.1. A PG-250 portable multi-gas analyzer measured \( \text{O}_2 \) (galvanic cell), \( \text{CO}_2 \) (ND-IR), and \( \text{NO}_x \) (chemiluminescence). Before each day of testing the gas analyzer was calibrated with two tanks of calibration gases. The first supplied a measure of \( \text{NO}_x \) and the other a measure of the \( \text{CO}_2 \) and \( \text{O}_2 \) together providing a zero and a high measurement for each measured concentration.

3.6 Ash Collection and Loss on Ignition

Ash samples were collected by a cyclone shown in Figure 3.1 at the base of the reactor system. Before collecting ash samples time was taken sufficient to assume that the reactor was running at steady state. The reactor was initially heated with natural gas for 4 to 5 hours or until several thermocouples in the reactor wall reported >1000 K then transitioned to biomass fuel which ran for a minimum of 30 minutes before taking a sample. Changing conditions did not result in significant changes in furnace temperature and therefore only 5 minutes was taken between operating conditions before collection. Before beginning to collect the sample a barrel was cleaned and paper was placed inside covering the base of the barrel.

ASTM procedure D7348 was used to determine loss on ignition (LOI) of the collected ash samples. LOI is a means of estimating the unburned carbon in a sample but because LOI includes the weight loss of sulfur and other elements that are removed in the LOI measurement, it tends to predict higher than the real measure. The inference has to be made that the total mass loss is equal to the carbon mass loss. Equation 3.4 shows the method used to measure LOI, where \( W_{\text{dry}} \) represents the weight of the ash after the drying process and \( W_{\text{cfa}} \) represents the carbon free ash after being heated to high temperatures for several hours. Equation 3.5 represents
the relationship of LOI to carbon burnout. For the purposes of this work we will be focused on the data represented as LOI.

\[ LOI = \frac{W_{dry} - W_{cf a}}{W_{dry}} \]  

(3.4)

\[ Carbon\ Burnout\ % = \frac{1 - Y_{ash} - LOI}{(1 - LOI) * (1 - Y_{ash})} \]  

(3.5)
4 RESULTS AND DISCUSSION

Experimental results are presented in this chapter including flame images, flame intensity, flame temperature, NO exhaust gas measurements, and LOI (loss on ignition) measurements for fine (~220µm) and medium (~500µm) sized particles.

4.1 Flame Images

Flame images for fine particles captured from the top port, spanning a distance from 0.05 m to 0.33 m below the burner exit are shown in Table 4.1. The images in the table were selected to represent 40 other images taken at this location. Details of the image capturing process are included in section 3.4. Each row of Table 4.1 shows a specified center tube flow rate increasing in amounts of oxygen addition from top to bottom while each column, left to right, shows a geometrical configuration of the burner with an assumed increase in mixing. The center column has a configuration with a smaller center tube which will produce a higher jet velocity and momentum in comparison to the left column and the column on the right is collected with the small center tube and a swirled primary fuel tube. The gain above each image compares the product of the aperture area and exposure time with the most common value of this product set to 1.0. Both aperture area and exposure time are linear with the pixel response of the camera as shown in Draper [27]. Therefore, these values can be easily combined to give an overall gain relating the total incident visible energy to the camera of one image compared to another. A high
gain indicates that an image is not as bright as it appears in comparison to a low gain image. The first row of images represents air only combustion while the succeeding rows represent increasing values of oxygen addition. When the gain is considered, it can be seen that although the images in the first row appear brighter, the flames in the second row are actually more luminous.

It can be seen in Table 4.1 that conditions with oxygen enrichment always produce a flame of higher intensity. Flame intensity is produced by a combination of flame temperature and soot particle concentration. Flame temperature is expected to increase with increasing oxygen concentration while premixing of fuel and oxidizer prior to ignition should reduce soot formation. Generally when moving from left to right in the images, the intensity is seen to decrease which is consistent with the expected increase in premixing of fuel and oxidizer prior to ignition produced by the burner configuration. This increase in premixing is expected because it is assumed that the configurations produce higher swirled flow moving left to right in the table. Also when moving from top to bottom, the intensity is seen to increase by oxygen addition. The reason for the intensity change will be more clearly identified when these images are used to determine flame temperature and soot concentration.

The image in column 1 of row 2 of Table 4.1 is shown twice in Table 4.2, once as shown in Table 4.1 and a second time with the intensity of all pixels increased. These images show that there are actually two flames zones in the 2 kg/hr images; a very intense center flame surrounding the oxygen injection and a less intense flame similar to that of air-only combustion surrounding the oxygen flame. The introduction of oxygen in the center jet is therefore seen to produce a double flame with the inner flame at a higher intensity than the outer flame.
Table 4.1: Representative flame images for all operating conditions using fine particles. Images represent 19 cm vertically.

<table>
<thead>
<tr>
<th>Condition</th>
<th>FP/lg/cen</th>
<th>FP/sm/cen</th>
<th>FP/sm/cen*</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td><img src="image1.png" alt="Image" /> GAIN=3.69</td>
<td><img src="image2.png" alt="Image" /> GAIN=5.90</td>
<td><img src="image3.png" alt="Image" /> GAIN=7.37</td>
</tr>
<tr>
<td>2 kg/hr O&lt;sub&gt;2&lt;/sub&gt;</td>
<td><img src="image4.png" alt="Image" /> GAIN=1</td>
<td><img src="image5.png" alt="Image" /> GAIN=1</td>
<td><img src="image6.png" alt="Image" /> GAIN=1</td>
</tr>
<tr>
<td>8 kg/hr O&lt;sub&gt;2&lt;/sub&gt;</td>
<td><img src="image7.png" alt="Image" /> GAIN=1</td>
<td><img src="image8.png" alt="Image" /> GAIN=1</td>
<td><img src="image9.png" alt="Image" /> GAIN=5.90</td>
</tr>
<tr>
<td>18 kg/hr O&lt;sub&gt;2&lt;/sub&gt;</td>
<td><img src="image10.png" alt="Image" /> GAIN=1</td>
<td><img src="image11.png" alt="Image" /> GAIN=1</td>
<td><img src="image12.png" alt="Image" /> GAIN=7.37</td>
</tr>
</tbody>
</table>
The influence of the burner geometries on the center flame can be seen by comparing the three images of flames at 2 kg/hr of oxygen flow. The two flames on the right have the oxygen flowing through a smaller diameter tube than the flame on the left. This smaller diameter produces a higher velocity and elongates the inner oxygen flame in comparison to the flame imaged on the left.

Moving down Table 4.1 to higher flow rates of oxygen in the center tube, the ability to visually identify the inner oxygen flame from the air flame becomes more difficult. The inner flame was larger and penetrating the entire length of the top port due to the higher momentum in the oxygen jet and the boundary between the air and oxygen flame was not as sharp. This higher momentum appears to increase mixing between the oxygen and the surrounding gas. For the geometries represented in the two columns to the right, the ignition of the center flame is seen to be delayed with increasing oxygen flow rate allowing entrainment of fuel and other gases into the center oxygen jet prior to ignition.

Table 4.3 shows representative images for operating conditions that used medium sized particles. Like Table 4.1 the burner configurations are increasing in assumed oxygen premixing from left to right and increasing oxygen flow rate from top to bottom. Of the 8 kg/hr operating
conditions for medium particles, only the MP/lg/cen condition shows signs of an inner oxygen flame as was shown in the fine particles cases of Table 4.1. The flame image at this condition shows evidence of large scale flame structures unlike the air flame cases where the flame structures are smaller and more random and turbulent. If this condition does in fact have an inner oxygen flame it should also produce higher values of intensity and temperature. The intensity and temperature data to be discussed in section 4.2 and 4.3 respectively support this conclusion.

The MP/lg/cen configuration shows the greatest difference in the gain of the three images between air and oxygen conditions. Since the assumed oxygen premixing increases from left to right and the images decrease in intensity from left to right, the images are consistent with the hypothesis that premixing reduces soot formation. With the addition premixing there may also be more mixing of recirculated combustion products that may lower flame temperature which also affects the measured intensity. A possible reason that the inner flame is not as apparent or is not as distinct in the flame with medium particle size is because of the increased time required for devolatilization. The fine particles may rapidly release their volatiles much closer to the burner exit create a more dense volatile cloud than exists with medium particles. The slower devolatilization of the medium particles may allow the oxygen, fuel, and air streams to premix before volatilization is complete which may result in lower soot formation and a lower intensity flame that looks more similar to the air fired cases. If mixing is slow between the fuel and oxidizer, as in the MP/lg/cen case, the combustion dynamics are such that a distinct inner flame is evinced.
Table 4.3: Representative flame images for all operating conditions using medium particles.

<table>
<thead>
<tr>
<th></th>
<th>MP/lg/cen</th>
<th>MP/lg/annulus</th>
<th>MP/sm/cen</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Air</strong></td>
<td>GAIN=3.00</td>
<td>GAIN=3.73</td>
<td>GAIN=3.73</td>
</tr>
<tr>
<td><strong>8 kg/hr O2</strong></td>
<td>GAIN=0.48</td>
<td>GAIN=1.19</td>
<td>GAIN=3.00</td>
</tr>
</tbody>
</table>

### 4.2 Flame Intensity

Two techniques were used to produce quantitative data from the images. In the first, the calibrated camera was used to turn the digital pixel counts into an absolute intensity as discussed in the method section. Figure 4.1 shows two plots with Intensity vs. distance from the burner for fine particles (Figure 4.1a) and for medium particles (Figure 4.1b). The error bars in the figure, representing one standard deviation, have been included to show the fluctuations in the visible intensity of the flame caused by its turbulent nature. It is important to note that the intensity values shown here are an average of many different images where the brightness of the flame
fluctuates greatly. What can be learned from the figure is the difference in the degree of fluctuations from the fine particles to the medium particles. The fine particles show a more consistent flame than the medium particles. This is due to the fact that the smaller particles release their volatiles more quickly producing a more dense volatile cloud whereas the medium particles may volatilize more slowly creating a lower fuel to air ratio causing inconsistencies in the flame. The day to day variation in the flame is shown as the error bars of the air case for the medium particles.

The objective was to produce an axial flame intensity representative of the flame. This was accomplished by dividing an image into nine sections, each imaging a 3.5 cm segment of the flame. The pixels within each section of an image were averaged to create a single intensity representing that axial position. Each averaged intensity was then ensemble averaged with intensities from all of the images obtained which was typically between 30 and 100 images.

The resulting intensity profiles obtained from fine particle images in the first, second, and third ports beginning at the top of the reactor are shown in Figure 4.2. The intensities are arranged with variable flow conditions for each of the three burner geometries. This allows for the comparison of intensity as a function of flow rate. The intensity plots shown in Figure 4.2 are consistent with the previous explanation of the flame images in Table 4.1. The intensity of the air-only flames is consistently lower than any of the oxy-fired cases. In each of the burner configurations the intensity of the 2 kg/hr oxygen case produces a very large peak indicative of the center oxygen flame seen in the images. This high intensity is seen for all three burner geometries and it is interesting to note that regardless of the high initial value the intensity quickly returns to an intensity level similar to the air case.
Figure 4.1: Intensity [W/m²-sr-µm] vs distance from burner exit [cm] showing fluctuations of flame intensity for air case and 8 kg/hr oxygen injection. (a) fine particles, (b) medium particles.

As the flow rate of oxygen increases the location of the peak intensity moves downstream. The 8 and 18 kg/hr oxygen cases reach a maximum toward the end of the first port and beginning of the second port respectively. The 18 kg/hr case reaches the highest intensities of all flow rates and produces higher intensities downstream in the third port. The two higher oxygen flow rates appear to ignite later and produce lower intensities than 2 kg/hr in the vicinity near the burner. The high intensity of the 2 kg/hr oxygen cases suggests a flame with fuel on one
side and pure oxygen on the other side. At higher flow rates, the oxygen appears to become increasingly diluted by recirculated products and air.

The same intensities shown in Figure 4.2 are rearranged in Figure 4.3 to show differences between burner configurations at each flow rate condition. For each of the flow conditions it can be seen that the burner geometry with the large diameter of lowest flow velocity produced the highest intensity while the small diameter, primary swirled geometry produced the lowest intensity. This is consistent with the expectation that premixing of volatiles and oxidizer are lowest for the large diameter and highest for the small diameter center and primary swirled fuel. Mixing appears to produce lower soot formation and overall flame temperature and therefore lower luminosity. This arrangement of the intensities also makes it clear that the shape or location of the peak intensity is mostly a function of the burner geometry while the magnitude of the peak is primarily a function of the flow rate. For each flow condition the FP/lg/cen configuration has a more rapidly declining intensity after the peak.

Intensity data from the medium particle operating conditions are shown in Figure 4.4. As previously discussed, the MP/lg/cen configuration was the only condition using medium sized particles that looked to have a center oxygen flame. In Figure 4.4 the oxygen enriched condition is clearly more intense than the air configurations. It also shows that intensity has a tendency to decrease when the burner geometry favors higher mixing. This may be caused by lower soot concentrations as well as lower temperatures produced by thorough mixing of fuel, oxidizer, and products. It also appears that configurations where the intensity of the air condition is higher the difference between oxygen and the air condition is also higher. This suggests the impact of oxygen addition on intensity is not as great when the center oxygen flow is allowed to premix more completely before oxidation.
Figure 4.2: Intensity \([\text{W/m}^2\text{-sr-\(\mu\)m}]\) vs. distance from the burner [cm] of three burner configurations: (a) FP/sm/cen*, (b) FP/sm/cen, and (c) FP/lg/cen.
When comparing the intensities of flames from the same operating conditions except for the particle size, the intensities are slightly higher for the large particles but differences are within the uncertainty of the measurements. One would expect the medium particles to release volatiles more slowly than the fine particles and therefore have more premixing prior to ignition. The location of peak intensity of both particle sizes is also similar which indicates a common ignition time. It is not expected however that the medium particles would have a higher intensity peak because of the prolonged volatile releasing process. Therefore the differences could be due to a difference in the chemical structure of the fuel. As stated in the method section the fuels were taken from two separate locations and therefore could have different properties giving the medium particles a higher likelihood of producing soot.
Flame Temperature

The temperature measurements obtained come from the spectral analysis of radiation in the visible range of wavelengths as produced primarily from soot particles. Soot particles oxidize as they pass through a flame front and based on an analysis of their thermal mass are very close to the gas temperature within the flame front. Soot particle temperature and flame temperature are therefore used interchangeably in the discussion which follows. The adiabatic flame temperature of a flame is controlled primarily by the amount of diluent or inert gases participating in the reaction. Typically, nitrogen (N₂) is the primary diluent in a flame as it is a

Figure 4.4: Intensity [W/m²·sr·µm] vs. distance from burner exit [cm] for medium particles. (a) Air condition (b) 8 kg/hr oxygen addition

4.3 Flame Temperature

The temperature measurements obtained come from the spectral analysis of radiation in the visible range of wavelengths as produced primarily from soot particles. Soot particles oxidize as they pass through a flame front and based on an analysis of their thermal mass are very close to the gas temperature within the flame front. Soot particle temperature and flame temperature are therefore used interchangeably in the discussion which follows. The adiabatic flame temperature of a flame is controlled primarily by the amount of diluent or inert gases participating in the reaction. Typically, nitrogen (N₂) is the primary diluent in a flame as it is a
major component of air. When oxygen is used in place of air, the adiabatic flame temperature increases dramatically because of the reduction of N₂ diluent.

Flame temperature can also be reduced because of increased heat transfer. Two forms of heat transfer of particular importance are convective heat transfer from product gases such as CO₂ and H₂O which are mixed with incoming fuel and radiative heat transfer from soot and gases in the flame. When product gases are partially cooled before participating in coal particle heat-up and devolatilization, they also act as diluents and reduce flame temperature.

Temperatures obtained from flame images are shown in Figure 4.5 for the fine particle operating conditions. As might be expected, because of the high concentration of nitrogen diluent in air, the air-only flow configuration produced the lowest flame temperature. Although not always true, the trend in the top port between 0.05 and 0.33 m from the burner is that the temperature increases with increasing oxygen flow rate or decreasing nitrogen diluent.

The oxygen flame temperatures are highest in the burner configuration with primary swirl in comparison to the other two configurations. It is unclear if the high temperature is a result of higher overall oxygen concentration and lower nitrogen diluent in the first 0.33 m or if the heat loss from product gases used to ignite the fuel stream is lower for this operating condition. Perhaps, the answer is some combination of both.

The primary swirled burner configuration also produced the largest temperature difference between the oxygen and air-only cases. In Figure 4.5c it can be seen that the 8 kg/hr oxygen flow of the FP/sm/cen* configuration begins at a higher temperature near the burner than the other two configuration but decreases in temperature more rapidly with increasing distance from the burner. This suggests that this configuration has higher heat transfer rates cooling the flame more quickly than the air combustion. For some flow conditions, the temperature could not
be determined because the intensity was too low to be measured or to produce a measurable temperature. The data in Figure 4.5 support the conclusion that burner configurations assumed to have less overall mixing result in lower near burner temperatures and higher downstream temperatures for the oxygen injected cases. It is also interesting to note that the intensity peak does not correlate to a higher temperature.

![Figure 4.5: Temperature [K] vs. distance from the burner [cm] for fine particle configurations: (a) FP/sm/cen*, (b) FP/sm/cen, (c) FP/lg/cen.](image)

Figure 4.6 shows the temperature data of the air only and the 8 kg/hr oxygen addition cases for the variable configurations involving medium sized particles. Figure 4.6a shows the results of the air only conditions for each configuration whereas Figure 4.6b displays 8 kg/hr oxygen addition conditions of each configuration including a condition where 8 kg/hr oxygen was added to the secondary air flow (channel 4). It can be seen in Figure 4.6b that the MP/lg/cen oxygen condition produces a higher flame temperature than the three other data sets in the plot.
This is the same condition that evinced the center oxygen flame and therefore should show a higher temperature than the conditions without a pure oxygen flame. The condition assumed to have the most mixing (MP/sm/cen) because the oxygen would be added at a higher velocity shows the lowest temperature as would be expected. It is interesting to note that oxygen addition through the secondary has a similar temperature to addition through the annulus.

Looking further at Figure 4.6 the higher mixing conditions (MP/lg/ann & MP/sm/cen) showed little to no difference in temperature from air combustion to 8 kg/hr oxygen addition. Oxygen addition, then in order to have a significant effect on the flame temperature for medium sized particles should be introduced to the furnace in such a way that it is slow to mix with the surrounding products and produces an inner flame. Also when this inner flame is visible we find that temperature and intensity decrease faster downstream suggesting increased heat flux rates over other conditions. This is again seen in Figure 4.6b where MP/lg/cen is initially the highest temperature and then around the second window becomes almost the lowest.

In both plots of Figure 4.6 as the data gets further from the burner there is a tendency for the data line to be inconsistent showing major temperature differences from one point to the next. Images selected from the lower two ports were representative of the average flame in that location. Often towards the bottom of the port the flame did not exist and therefore when temperature data were collected it is possible that only one or two flame images were harvested for one data point and one or two different images for the next data point. Thus the inconsistency of the flame caused relatively high uncertainties.
Loss on Ignition (LOI)

Loss on Ignition data as a function of oxygen flow rate are shown in Figure 4.7 for medium and fine particles where configurations 2 and 3 were averaged for the small particles and 4 and 6 for the medium particles (see Table 3.2) where error bars represent one standard deviation. The LOI of the fine particles was found to be in the range of 10-15% with little variation over the range of O$_2$ flow rates. However, there is a significant decrease in LOI with increasing O$_2$ flow rate for the medium particles. This trend was seen for all of the burner operating conditions when O$_2$ flow rate was increased for the medium particles. In the near burner region, the char oxidation
The rate is expected to be limited by the transport of oxygen to the char surface. The amount of char oxidation is expected to be a strong function of oxygen concentration in the vicinity of the particle and the particle residence time. The addition of oxygen to the center of the burner is expected to increase oxygen concentration without significantly impacting residence time. Due to the small ash fraction of this biomass, the carbon burnout for the fine particles with 10% LOI is 99.94%. At this level of burnout, the char porosity is decreased and active oxidation sites are decreased making burnout slow and changes in burnout more difficult to observe. It is therefore likely that improvements in burnout beyond what is already achieved with air fired combustion are difficult to achieve and are more difficult to detect.

![Figure 4.7: LOI [%] vs. O$_2$ flow rate [kg/hr] for different particle sizes under burner configurations 2, 3, 4 and 6 (see Table 3.2).](image)

The impact of oxygen addition location on LOI of the medium particles is shown in Figure 4.8 where two data sets are compared, one adding oxygen in the center and the other in the annulus. Repeated data points for a given oxygen flow rate were collected on different days and are used to show repeatability of the results. Not all data has been repeated thus for data that does not show an error bar the value is either smaller than the symbol or only one piece of data was collected from that condition. The center oxidizer addition was found to produce lower LOI
results than the annulus. It is difficult to reason how the flow location (center or annulus) impacted the variables of oxygen concentration and residence time in this complicated system.

Figure 4.8: LOI [%] vs. $O_2$ flow rate [kg/hr] comparing oxygen addition location.

Figure 4.9 shows LOI as a function of oxygen flow rate for two different center tube diameters. The smaller of the two diameters produced a higher velocity and momentum and was more disruptive to the central recirculation zone. As with the other large particle data results, increasing oxygen flow rate produced lower LOI, however, the high velocity jet from the small diameter tube was found to be less sensitive to oxygen addition.

Loss on ignition data for each of the flow conditions and each burner configuration for the fine particles are shown in Figure 4.10. As was mentioned earlier the loss on ignition is seen to be very low for this biomass, probably due to the fine particle size of the fuel. There does not appear to be a strong trend with LOI and oxygen flow rate, but instead the LOI correlated more closely with the burner configuration. The lowest LOI was obtained with the burner configuration that had the highest swirl and showed indications of being the most well mixed. This is consistent with the fact that char burnout is typically diffusion limited, not temperature limited. Therefore higher concentrations of surrounding oxygen should produce better burnout.
The difference in LOI from the other burner configurations is not well understood except to say that differences in all of the LOI are small and burnout was very good.

![Figure 4.9: LOI [%] vs. O2 flow rate [kg/hr] comparing the effects of changing center tube diameter size](image)

![Figure 4.10: LOI [%] vs. O2 flow rate [kg/hr] for all fine particle configurations](image)

4.5 Nitric Oxide (NO)

While oxygen addition was found to improve burnout, it is important to also evaluate the impacts of oxygen addition on NO. Exhaust concentrations of NO are shown in Figure 4.11 as a function of oxygen flow rate where all of the NO data have been averaged for fine and medium
particle sizes. For small amounts of oxygen addition, from 0 to 4 kg/hr of oxygen, there is only a small increase in NO for both particle sizes. With a further increase in oxygen flow rate, NO increased modestly for the large particles but dramatically for the small particles.

In order to obtain a low-NOx flame, the dynamics of the combustion process must produce a fuel rich zone where nitrogen in the fuel and NO have the opportunity to be reduced and form N₂. In the case of center oxygen injection, the oxygen flow was introduced into an area where a fuel rich zone is normally formed by flow recirculation. With low flow rates of oxygen the impact on NO appears minor, but at higher flow rates, this recirculation zone appears to be significantly impacted.

Because the small particles heat more rapidly, they release their volatiles and burn out more quickly. Thus a higher fraction of fuel nitrogen has been released in the near burner zone where the oxygen is being released. The mixing of oxygen with these volatiles produced higher rates of NO formation.

![Figure 4.11: NO [ppm] vs. O₂ flow rate [kg/hr] for different particle sizes under burner configurations 2, 3, 4 and 6 (see Table 3.2)](image)

Figure 4.12 shows NO concentration in the exhaust as a function of oxygen flow rate for two locations, center and annulus and two center tube diameters. Looking at all of the data, there is a
clear trend of increasing NO with increasing oxygen flow rate. The largest increases were on the order of 35% for the small center tube, center injection of oxygen. NO concentrations were lowest for the annular oxygen injection and large diameter center tube. These conditions produce less NO because there was less overall mixing of the oxygen into fuel rich areas of the furnace and less disturbance of the recirculation holding fuel rich pockets that allow the NO to reduce.

Figure 4.12: NO [ppm] vs. O₂ flow rate [kg/hr] for all medium particle configurations

An attempt was made to correlate NO emissions with the velocity of the oxygen flow exiting either the center tube (channel 1) or the annulus (channel 2). The results are shown in Figure 4.13. In this Figure it can be seen that exhaust NO correlates well with oxygen flow velocity for all of the burner configurations except the small center tube velocity. NO might be expected to correlate with O₂ velocity if the O₂ velocity were an indicator of mixing between the fuel and oxygen streams. Higher mixing should produce higher rates of NO formation. It is possible that three of the four burner configurations allowed the burner recirculation zone to remain primarily in place but simply increased the mixing within that reverse flow zone while the small tube diameter produced a momentum large enough to penetrate through the recirculation zone and produced a fundamentally different fluid flame structure.
The flue gas concentrations of NO are shown for each of the flow conditions and burner configurations for fine particles in Figure 4.14. Error bars on the figure show plus and minus one standard deviation where more than one replication of the operating condition were obtained. The results show a general trend of increasing NO concentration with increasing oxygen flow rate where zero oxygen flow rate represents the air-only flow condition. A small amount of oxygen injection (2 kg/hr) shows little if any increase in NO, while further increases in O$_2$ produce more significant increases in NO. Given the flame images shown in Table 4.1, the 2 kg/hr flow rate of oxygen produced a small flame surrounding the oxygen addition location which was not well mixed with the fuel. The high temperature produced by this inner flame may increase the devolatilization rate of the fuel without significantly increasing oxygen availability to the fuel and therefore, NO formation is not increased. With higher O$_2$ flow rates, the increase mixing between fuel and oxygen and the destruction or hampering of the recirculation zone causes the increased NO concentrations. It is unclear as to why the FP/sm/cen configuration at 18 kg/hr oxygen did not produce similar NO levels to its 18 kg/hr counterparts.
Figure 4.14: NO [ppm] vs. O2 flow rate [kg/hr] for all fine particle configurations
5 SUMMARY AND CONCLUSIONS

A burner capable of various oxygen injection strategies was used to study the impact of oxygen injection flow rate and momentum on a biomass flame with mass mean particle sizes of 224 and 500 microns. The flame shape, intensity and temperature were examined using a two-color digital camera. Flame images, NO\textsubscript{x}, and LOI data were obtained for an matrix of operating conditions of variable oxygen flow rate and burner geometries.

The addition of a small flow rate of oxygen (2 kg/hr) with the fine particles was found to produce a second oxygen rich flame in the center of an air flame. This oxygen flame was of very high intensity and temperature (2100 K). The presence of this flame did not increase NO\textsubscript{x} or change burnout but increased the near burner heat flux. This oxygen flame was not apparent at the same low oxygen flow rate for medium particles, presumably due to the delayed release of volatiles. However, the medium particles did show a center oxygen flame in one of the burner configurations with 8 kg/hr of oxygen injection and the large center tube. This condition recorded the highest temperatures and flame intensity with the best carbon burnout of the medium particle conditions; however, it also increased NO\textsubscript{x} formation by 30%.

Increasing the oxygen flow rate to 8 and 18 kg/hr produced an elongated oxy-flame that produced higher temperatures and intensity over the length of the luminous signal. Introducing only 2 kg/hr of oxygen resulted in high temperatures and intensity near the burner exit but temperature and intensity quickly returned to values similar to the air condition downstream. The
shape of the intensity profile was consistent across burner configurations with similar oxygen flow rates while the magnitude of the intensity profile varied. Presumably, configurations that were conducive to large pockets of high oxygen concentrations produced the highest intensity and temperature.

The location of oxygen injection had little impact on NO emissions. The addition of oxygen to the center and annular tubes of the burner both produced a small increase in NO emissions. The increase in NO emissions was significantly higher for small particles than it was for medium particles. This may be due to the more rapid release of volatile nitrogen from small particles.

Increasing oxygen flow rate for all burner configurations decreased LOI and increased particle burnout for medium (500 µm mass mean) particles. Increasing oxygen flow rate had little impact on the LOI of small (224 µm) particles. Oxygen addition within the burner was more beneficial for improving the combustion of medium particles than it was for small particles.

Many of the results recorded and explained in this work have verified other work in the literature. An example of this information includes the impact overall oxygen concentration has on flame temperature, radiative intensity, burnout and NO\textsubscript{x} emissions. This work uniquely investigates the impact of localized oxygen addition which is not explicitly covered in the literature. The internal oxygen flame shown in these experiments create an opportunity for biomass flames to match the intensity and temperature of coal flames in cofiring applications. The inner flame has the potential to heat up particles more quickly causing early devolatilization and increase burnout with significantly increasing NO\textsubscript{x} or the overall oxygen concentration.
REFERENCES


APPENDIX A: BYU BURNER SWIRL

The BYU burner is a movable block swirl type, that simulates a commercial low-NO\textsubscript{x} burner by producing a long, rich flame which reduces the flame temperature and NO\textsubscript{x} concentration. Figure A.1 displays the scheme of the BYU burner where the arrows represent the secondary air flow throughout the burner. Upon entering the burner, at the top and from the sides, the air is fed through a distributed material that straightens the flow before it enters the swirl plates. The swirl plates split the flow into axial and tangential portions that can be seen in Figure A.2. These plates are stacked with interlocking blocks. The top plate is free to rotate via a threaded rod accessible from the outside of the burner. As it is rotated the smaller blocks motion relative the larger blocks from the base plate change the fraction of tangential and axial flow. Figure A.2 also illustrates the variables used in the derivation of flow equations from which the characteristic swirl number of the flow through the plates can be estimated.
The swirl number, $S$, is a widely used parameter for characterizing swirling flows. This non-dimensional criterion is defined as the following by Beer and Chigier in 1964.
where $G_\theta$ is the axial flux of angular momentum,

$$G_\theta = \int (\omega \cdot r) \rho u \, dA \tag{A.2}$$

$G_z$ is the axial flux of axial momentum,

$$G_z = \int u(\rho u) \, dA + \int p \, dA \tag{A.3}$$

and $R$ is the burner radius. In Equations 2 and 3, $u$ and $\omega$ respectively represent the axial and tangential velocities, $\rho$ represents the flow density, $A$ is the flow area, $p$ is the static pressure, and $r$ is the radius.

Work has been done previously to analytically estimate the swirl number from the BYU burner. Todd Reeder referenced Beer and Chigier (1972) for Equations A.4 and A.5, excluding the square on the denominator of A.5, in his master thesis at BYU.

$$S = \frac{\sigma R}{2B} \left[ 1 - \left( \frac{R_h}{R} \right)^2 \right] \tag{A.4}$$

$$\sigma = \frac{\pi}{z\xi_m} \sin(\alpha) \frac{\cos(\alpha) [1 + \tan(\alpha) \tan(\xi/2)](\xi/\xi_m)}{\{1 - [1 - \cos(\alpha) (1 + \tan(\alpha) \tan(\xi/2))]\xi/\xi_m\}^2} \tag{A.5}$$

$S$ is the swirl number and $\sigma$ is the angular momentum flux. $R_h$ and $R$ are the inner and outer radii of the secondary air exit, respectively, $\alpha$ is the vane angle, $B$ is the vane thickness, $\xi$ and $\xi_m$ are the rotation and maximum rotation of the swirl plate respectively, and $z$ is the number of blocks. These variables are shown in Figure A.1 and Figure A.2 while values for fixed variables are shown in Table A.1.

In 2003 Fudihara et. al. added the square on the denominator of Equation A.5 referencing personal communication with Beer. Due to this correction the swirl is being reevaluated in this work. Stan Harding provided an in depth derivation of the swirl number correlation in his
dissertation at BYU. Equations A.6 and A.7 were gleaned from his work for theoretical secondary swirl and angular momentum flux.

\[ S^t_s = \frac{\sigma A_e}{2\pi BR} \]  

(A.6)

\[ \sigma = \frac{\pi}{L} \frac{\sin(\alpha) \tan(\xi/2) \cos(\alpha - \xi/2)}{\left\{\tan(\xi/2) \cos(\alpha - \xi/2) + \sin\left(\frac{\xi_m - \xi}{2}\right)\right\}^2} \]  

(A.7)

\( A_e \) in the swirl equation represents the exit area of the secondary swirl. Mathematically this equation is identical to Equation A.4. The \( L \) represents the number of blocks as \( z \) does in Equation A.5. If we assume \( \xi/2 \) to be a very small angle such that \( \sin(\xi/2) = \xi/2 \) it can be shown that Equation A.7 is mathematically identical to Equation A.5, thus further supporting the claim that the denominator in the angular momentum flux equations should be squared.

Assumptions made in the derivation of the previous equations include the conservation of all momentum from the swirl blocks to the burner exit, the pressure term from Equation A.3 is neglected, and the flow is assumed to be incompressible and inviscid.

Table A.1 shows dimensional values for the burner. Similar values were reported Todd Reeder’s thesis that are not equivalent to all the values shown here. The source of his values is unknown.

<table>
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<tr>
<th>Variable</th>
<th>Value</th>
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<tbody>
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<td>( R_h )</td>
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</tr>
<tr>
<td>( R )</td>
<td>2.00 in.</td>
</tr>
<tr>
<td>( B )</td>
<td>0.59 in.</td>
</tr>
<tr>
<td>( \alpha )</td>
<td>45.00°</td>
</tr>
<tr>
<td>( \xi_m )</td>
<td>21.56°</td>
</tr>
<tr>
<td>( N )</td>
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Table A.1: Selected dimensions of BYU swirl burner
Changing the swirl number of burner can be done during the operation. Since this adjustment is made by rotating the threaded rod that extends outside the burner a correlation must be made between the swirl number and rotations of the rod. Figure A.3 shows a plot of measured $\zeta$ angle versus rotations of the threaded rod. The data plotted in this figure is also recorded in Table A.2. When measuring the exact angle of $\zeta_m$ on the swirl plate itself is different from the full angle the plate can rotate. Though $\zeta_m$ is 21.56° the plate only rotates about 20° due to tolerances and possibly dirt build up. Thus the missing 1.56° was added to the entire range e.g. a $\zeta$ value of 19.78° represented a measurement of 19°.

A least squares approximation was fit to the data which produced a relationship between rotations and $\zeta$ shown by the equation in Figure A.3. The swirl number was estimated at every half turn using the $\zeta$ value obtained by the least squares fit. Table 3 shows each half turn and the correlating swirl number. Notice at the beginning of this table the rotations jumps to 2. This is due to the hysteresis in the mechanism.

interface between the threaded rod and swirl plate as the plate does not begin moving until the rod has been rotated about 1.5 times.
Figure A.3: Linear curve fit of $\xi$ to the rotations of the threaded rod.

Table A.2: Measured values of $\xi$

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</tr>
<tr>
<td>5</td>
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Table A.3: Multiple rod rotations matched with corresponding swirl numbers.

<table>
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<tr>
<th>Turns</th>
<th>Swirl</th>
<th>Turns</th>
<th>Swirl</th>
<th>Turns</th>
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