EFFICIENCY AND EMISSIONS STUDY OF A RESIDENTIAL MICRO-COGENERATION SYSTEM BASED ON A MODIFIED STIRLING ENGINE AND FUELLED BY A WOOD DERIVED FAST PYROLYSIS LIQUID-ETHANOL BLEND

by

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A thesis submitted in conformity with the requirements for the degree of Master of Applied Science
Graduate Department of Mechanical and Industrial Engineering
University of Toronto

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Abstract

Efficiency and Emissions Study of a Micro-Cogeneration System Based on a Modified Stirling Engine and Fuelled by a Wood Derived Fast Pyrolysis Liquid-Ethanol Blend

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Master of Applied Science

Graduate Department of Mechanical and Industrial Engineering
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A residential micro-cogeneration system based on a Stirling engine unit was modified to operate with an 80 % vol. wood derived fast pyrolysis liquid (bio-oil) and 20 % vol. ethanol blend. A pilot stabilized swirl combustion chamber was designed to replace the original evaporative burner due to bio-oil’s nondistillable nature, and the control systems of the original engine had to be modified. Efficiencies for the bio-oil/ethanol blend were found to be higher than those of diesel due to the higher heat loss incurred with diesel through the non-optimized combustion chamber. In commercial applications, heat loss through the optimized combustion chamber will be negligible. In order to evaluate the performance between bio-oil and diesel for an optimized system, a modified power efficiency, which neglects the heat loss through the combustion chamber walls, is proposed. The modified power efficiency for diesel and the bio-oil/ethanol blend were comparable, therefore making bio-oil a suitable replacement for diesel in Stirling engines. CO emissions for the bio-oil/ethanol blend were very high due to poor atomization and a high fuel-air equivalence ratio, leading to larger droplets, and poor mixing and spray quality. Higher primary and atomizing air could not be used due to the onset of acoustic fluctuations in the combustion chamber which hindered flame stability. CO emissions for diesel at steady state were significantly lower than bio-oil due to its fully distillable nature. NO\textsubscript{x} emissions were higher for bio-oil than for diesel due to the formation of fuel bound NO\textsubscript{x} from bio-oil’s inherent nitrogen content. Extended operation with the bio-oil/ethanol blend had to be stopped after 97 minutes due to the clogging of the hot end heat exchangers which restricted the primary air flow and reduced the power output. Based on the operating experiences, a set of design recommendations were proposed such as the inclusion of air preheating, refractory lining, and a better working fluid which would increase the efficiencies and emissions.
Acknowledgments

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## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>$I_{ALT}$</td>
<td>Alternator current</td>
</tr>
<tr>
<td>$U$</td>
<td>Axial Component of velocity</td>
</tr>
<tr>
<td>$G_\phi$</td>
<td>Axial flux of angular momentum</td>
</tr>
<tr>
<td>$G_x$</td>
<td>Axial flux of axial momentum</td>
</tr>
<tr>
<td>$V_{BAT}$</td>
<td>Battery voltage</td>
</tr>
<tr>
<td>$B$</td>
<td>Depth of swirl blocks</td>
</tr>
<tr>
<td>$I_{FID}$</td>
<td>FID current</td>
</tr>
<tr>
<td>$Q$</td>
<td>Heat input</td>
</tr>
<tr>
<td>$Q_{out}$</td>
<td>Heat input during isothermal compression</td>
</tr>
<tr>
<td>$Q_{in}$</td>
<td>Heat input during isothermal expansion</td>
</tr>
<tr>
<td>$\dot{Q}$</td>
<td>Heat transfer rate</td>
</tr>
<tr>
<td>$Q_{regen}$</td>
<td>Heat transfer to and from the regenerator</td>
</tr>
<tr>
<td>$m$</td>
<td>Mass</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>Mass flow rate</td>
</tr>
<tr>
<td>$\dot{\dot{W}}_{net}$</td>
<td>Net power output rate</td>
</tr>
<tr>
<td>$n$</td>
<td>Number of swirl blocks</td>
</tr>
<tr>
<td>$% O2$</td>
<td>Oxygen content (volume)</td>
</tr>
<tr>
<td>$P_{OUT}$</td>
<td>Power output by engine</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>$r$</td>
<td>Radius</td>
</tr>
<tr>
<td>$R_1, R_2$</td>
<td>Resistors</td>
</tr>
<tr>
<td>$N_{PUMP}$</td>
<td>Rotation speed of fuel pump</td>
</tr>
<tr>
<td>$h$</td>
<td>Specific enthalpy</td>
</tr>
<tr>
<td>$R$</td>
<td>Specific gas constant</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat at constant pressure</td>
</tr>
<tr>
<td>$p$</td>
<td>Static pressure</td>
</tr>
<tr>
<td>$R_h$</td>
<td>Swirl generator inner radius</td>
</tr>
<tr>
<td>$S$</td>
<td>Swirl number</td>
</tr>
<tr>
<td>$W$</td>
<td>Tangential component of velocity</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature</td>
</tr>
<tr>
<td>$R_t$</td>
<td>Thermal resistance</td>
</tr>
<tr>
<td>$V_{in}$</td>
<td>Voltage into op-amp</td>
</tr>
<tr>
<td>$V_{out}$</td>
<td>Voltage out of op-amp</td>
</tr>
<tr>
<td>$V_{O2}$</td>
<td>Voltage signal from oxygen sensor</td>
</tr>
<tr>
<td>$V_{fuel}$</td>
<td>Voltage to fuel pump</td>
</tr>
<tr>
<td>$V$</td>
<td>Volume</td>
</tr>
<tr>
<td>$W$</td>
<td>Work output</td>
</tr>
</tbody>
</table>

**Greek Symbols**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\xi$</td>
<td>Adjustable swirl block angle</td>
</tr>
</tbody>
</table>
\( \alpha \)  
Fixed swirl block angle

\( \eta_{\text{stirling}} \)  
Ideal efficiency of Stirling engine

\( \xi_m \)  
Maximum opening angle

\( \eta_{\text{power,modified}} \)  
Modified power efficiency

\( \eta_{\text{power}} \)  
Power efficiency

\( \eta_{\text{thermal}} \)  
Thermal efficiency

\( \eta_{\text{energy}} \)  
Total energy efficiency

**Acronyms**

CO\(_2\)  
Carbon Dioxide

CO  
Carbon Monoxide

CRZ  
Central Recirculation Zone

CHP  
Combined Heat and Power

DAQ  
Data Acquisition

EtOH  
Ethanol

FLOX®  
Flameless oxidation

FTIR  
Fourier Transform Infrared (spectrometer)

HHV  
Higher heating value

LHV  
Lower heating value

NO\(_x\)  
Oxides of Nitrogen

PLS  
Partial least squares
<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>PM</td>
<td>Particulate Matter</td>
</tr>
<tr>
<td>ppm</td>
<td>Parts per million (volume)</td>
</tr>
<tr>
<td>RMSE</td>
<td>Root mean square error</td>
</tr>
<tr>
<td>RPM</td>
<td>Rotation per minute</td>
</tr>
<tr>
<td>SLPM</td>
<td>Standard Liters per Minute</td>
</tr>
<tr>
<td>UHC</td>
<td>Unburned hydrocarbons</td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

1.1 Motivation

Many remote communities in places like Northern Ontario and the Yukon rely on diesel generators to fulfill their electricity needs. Diesel generators are the first choice for most of these localities for several reasons; there is a wide network of diesel generator distributors which makes delivery relatively easy, they are easy to install and require little maintenance, and these distributors offer attractive financing options [1]. However, there are many disadvantages associated with this technology and its application to remote power generation. Since diesel fuel is not readily available at these remote locations, it has to be transported, which is costly where the location is not easily accessible. Transportation and handling of diesel fuel makes it susceptible to spills. In one study, spills are estimated to be between 100 to 3000 gallons per month for a community [1]. The costs associated with remote power production are heavily subsided by the government to make it affordable for residents. Emissions from these diesel generators are another concern, resulting in greenhouse gasses and particulate matter. Greenhouse gasses accumulate in the atmosphere and are known to be one of the primary contributors to accelerating global climate change.
With the increased price and limited supply of fossil fuels, and concerns over global climate change, the issue of providing remote communities with electricity has been of major concern to governments around the world including countries like Canada, the United States, and Australia [1, 2]. In Canada, this has sparked an interest in renewable sources of energy, specifically biofuels and biomass, because of its large biomass resources. Many of these remote locations are located in the proximity of these biomass resources which can be used to offset the diesel fuel consumption [2]. Biomass, via the fast pyrolysis process, can produce fast pyrolysis liquid (or bio-oil) which is a renewable liquid fuel and greenhouse gas neutral. The energy density of this fuel is comparable, on a volumetric basis, to most petroleum based fuel oils. Bio-oil is ideally suited for applications such as cogeneration, remote power generation, peak shaving and bio-refinery applications.

Stirling engines are an ideal means to create electricity from low quality fuels such as bio-oil since they only require an external source of heat and can use a steady flame. There are many commercial Stirling engines, both residential and industrial, in the market, mostly fuelled by diesel fuel and natural gas, which are the basis for a cogeneration system. Cogeneration in this context implies the simultaneous production of both heat and electricity. In cogeneration systems, after the hot exhaust products have been used to produce electricity, the energy from the waste heat is recovered. This leads to better utilization of the fuel energy, thus increasing the overall efficiency of the system.

Micro-cogeneration systems are an extension of cogeneration systems which are used for residential applications (< 20 kW), providing heat for space heating and electricity for consumption or net metering back into the grid. In micro-cogeneration applications, the power output is governed by heat-demand, with electricity as a byproduct. The use of biomass for micro-cogeneration applications results in reduced greenhouse gas emissions [3]. Implementing a micro-cogeneration system based on a Stirling engine is an ideal application for bio-oil, as the biomass resources and mobile conversion systems such as that produced by Agri-Therm Inc. are available [4].
1.2 Objectives

The objective of this research is to operate a Stirling engine on fast pyrolysis liquid. The Stirling engine used for this study is the Whispergen DC micro-combined heat and power (micro-CHP) system, developed by Whispertech Ltd. of New Zealand. The original combustion chamber of the Stirling engine features an evaporative burner, which is not suitable for non-distillable fuels such as bio-oil. This will require the designing, building, and testing of a new combustion chamber, which is compatible with bio-oil, along with the various components which go alongside.

The operation of the engine with the new combustion chamber will require the control systems to be modified as the engine relies on these controls to ensure correct operation. A comparative study of efficiency and emissions will be performed between bio-oil and diesel, which is the engine’s original fuel. The efficiency analysis will be based on an energy balance, and emissions measurements will be performed. Testing will also be performed to determine the maximum length of operation of the new system with bio-oil.

It is emphasized at this point that the goal is not to optimize the system, rather a proof of concept. However, the operation of the engine will bring about issues with durability, maintenance, efficiency, and operation which will be addressed. Based on these issues, a recommendation will be made as to the design parameters which should be considered if this were to reach the commercial stage.
Chapter 2

Background

2.1 Stirling Engines

Stirling engines are reciprocating external combustion engines which operate on a closed thermodynamic cycle. There are many benefits associated with the Stirling engine, including higher efficiencies (even at partial loads), quiet operation, and long maintenance-free periods, compared to conventional engines such as the internal combustion engine [5]. Since an external source of heat is required, this allows for the flexibility to use a variety of heat sources, including solar energy, process heat, biomass, and biofuels. Continuous combustion of fuels can be well controlled resulting in significantly lower emissions than internal combustion engines. Many applications have been investigated for the Stirling engine, including automotive [6] and space [7], but the primary use currently is towards combined heat and power (CHP), or cogeneration, systems. Although Stirling engines are expensive to produce, their benefits outweigh the costs [8].

2.1.1 Ideal Thermodynamic Cycle

Stirling engines operate on a closed regenerative cycle known as the Stirling cycle, where the working fluid is contained within the thermodynamic system. The cycle operates on four
processes which are outlined in the pressure-volume and temperature-entropy plots in Figure 2.1. An illustration of the hypothetical Stirling engine is also displayed in Figure 2.2. The working fluid inside the cylinder goes through the following steps [9]:

1-2: Isothermal expansion

2-3: Constant volume regenerative heat rejection

3-4: Isothermal compression

4-1: Constant volume regenerative heat addition

Figure 2.1: Pressure-Volume and Temperature-Entropy plots of the ideal Stirling cycle [9].

During the first process, isothermal expansion, the working fluid, which is at the maximum pressure, absorbs heat from an external heat source and expands isothermally. During expansion, work is done on the piston by the working fluid. In the next process, the displacer transfers the working fluid through the regenerator, which is a type of heat exchanger, to the cooling medium at a constant volume. The working fluid transfers heat as it passes through the regenerator, and ends the process at the same temperature as the cooling medium, with a significantly lower pressure. In process 3-4, the fluid is compressed, isothermally, and heat is transferred to the cooling medium. The final process is where the working fluid is again passed through the regenerator which delivers heat, thus raising the working fluid temperature to that of the hot space, and increasing the pressure. The work done by the piston on the fluid in process 3-4 is relatively low compared to the work done by the fluid on the piston, thus resulting in a net work output [10].
Figure 2.2: Hypothetical Stirling engine [9].

The net work output of the ideal Stirling cycle is represented by Equation 2.1, where \( m \) is the mass, \( R \) is the specific gas constant of the working fluid, \( V \) represents the volume and \( T_H \) and \( T_C \) are the hot and cold temperature, respectively [10]. The subscripts 1 and 2 represent the state of the working fluid. Heat transfer into the working fluid is shown in Equation 2.2, where the variables are the same as those of the previous equation. These two equations can then be used to calculate the efficiency of the system, which is shown in Equation 2.3. It is worth noting that the efficiency of the Stirling cycle is equal to the efficiency of the Carnot heat engine, thus implying that the Stirling cycle, under the second law of thermodynamics, has the maximum possible efficiency [10].

\[
W = -mR\ln\left(\frac{V_2}{V_1}\right)(T_H - T_C) \tag{Equation 2.1}
\]

\[
Q = mR\ln\left(\frac{V_2}{V_1}\right)(T_H) \tag{Equation 2.2}
\]

\[
\eta_{STIRLING} = 1 - \frac{T_C}{T_H} \tag{Equation 2.3}
\]
One of the most important parts of the Stirling engine is the regenerator, which is essentially an intermittent heat exchanger. The hot working fluid is passed through the cold regenerator, increasing the temperature of the regenerator and decreasing the temperature of the working fluid. The opposite is true when the cold working fluid is passed through. In practical Stirling engines, the regenerator is typically made from a stainless steel or ceramic mesh, and designed to maximize heat transfer while minimizing flow losses [11].

2.1.2 Non-Ideal Behavior and Effects on Engine Performance

There are many deficiencies in the actual cycle which decrease the maximum theoretical efficiency. Actual Stirling engines, which implement the Stirling cycle, are composed of complex drive mechanisms and components (heater, regenerator, and a cooler connected in series) which lead to losses. These losses, which lead to a decrease in efficiency, are due to working fluid friction, working fluid leakage, heat loss, dead volume, and mechanical losses.

The working fluid interaction with the walls and the regenerator introduces friction losses, and pressure drops, which occur from constantly moving the fluid from the hot end to the cold end. The losses incurred through the regenerator account for about 90% of the total internal losses and decrease the net work output [12]. Due to the high pressure of the working fluid, leakage is another issue resulting from imperfections in the seals, bearings, and the clearance volume between the piston and cylinder walls [13]. This also contributes to the decrease in net work output. Heat losses are incurred through conduction, via the cylinder from the cold end to the hot end, and also through the various components (pistons, displacers, and regenerators) [14]. The dead volume of the Stirling engine accounts for the volume of working fluid which does not contribute to the work output. This can account for up to 50% of the total internal volume in practical Stirling engines [12]. Mechanical losses are incurred through the drive shaft and other components which transfer the pistons motion for the production of power [14].

A major factor towards determining the performance of the Stirling engine is the choice of working fluid. To reach the maximum performance levels, the working fluid should have a high thermal conductivity, low heat capacity, and low viscosity [11]. A high thermal conductivity is required to provide increased heat transfer rates for the working fluid from the hot end to the cold end. A low heat capacity allows for large changes in temperatures for a
smaller energy input. Having a low viscosity will reduce the friction losses inside the cylinder. Given the criteria above, an ideal choice for a working fluid would be hydrogen. However there are many safety issues associated with its use in Stirling engines, including a high flammability limit and diffusion rates which make containment extremely difficult. The next best option is helium, although it has a viscosity which is twice as much as hydrogen and is relatively expensive compared to the available alternatives. Nitrogen and air are other choices, which are readily available and safe for high pressure applications. Table 2.1 outlines the properties of common working fluid choices for the Stirling engine.

Table 2.1: Properties of common working fluids for Stirling engines [15].

<table>
<thead>
<tr>
<th></th>
<th>Units</th>
<th>Hydrogen</th>
<th>Helium</th>
<th>Nitrogen</th>
<th>Air</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal conductivity</td>
<td>W/m·K</td>
<td>0.16835</td>
<td>0.14264</td>
<td>0.02400</td>
<td>0.02394</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>kJ/mol·K</td>
<td>0.029</td>
<td>0.020</td>
<td>0.029</td>
<td>0.029</td>
</tr>
<tr>
<td>Viscosity</td>
<td>cP</td>
<td>0.00865</td>
<td>0.01863</td>
<td>0.01657</td>
<td>0.01695</td>
</tr>
</tbody>
</table>

2.1.3 Engine Configuration

Stirling engines have three distinct configurations, which follow the same cycle, but differ in mechanical design characteristics. These configurations are alpha, beta, and gamma as shown in Figure 2.3 [16]. In this figure, H represents the hot end, C represents the cold end, P represents the piston, and D represents the displacer which is used in moving the working fluid from one cylinder to another. These configurations can be in either a single- or double-acting mode of operation. In the single-acting mode, the working fluid is in contact with only one side of the piston. In the double-acting mode, the working fluid is in contact with both sides of the piston, i.e. the expansion space of one cylinder is connected to the compression space of the other cylinder, resulting in an increase in the working space.

The alpha-configuration employs a hot and cold piston instead of a displacer inside the cylinder. The pistons move evenly to provide the constant volume heating and cooling to the working fluid. During the heat transfer process, the working fluid is transferred from one cylinder to the other. While one piston is stationary, the other works to compress the working
The beta-configuration employs a displacer and a piston inside the cylinder. The working fluid inside the cylinder moves from the top, through the heater, regenerator, and cooler, to the other. The compression of the working fluid occurs in the cold side, after which it is sent to the hot cylinder where it is goes through expansion.

The gamma-configuration also employs a displacer and a piston, but they are located in different cylinders. The working fluid is passed from the displacer cylinder, through the heater, regenerator and cooler to the piston containing cylinder. The piston both compresses and expands the working fluid.

The four-cylinder double-acting (FCDA) configuration, which is a variation of the alpha type, offers the highest possible mechanical efficiency [17]. This unique configuration features four interconnected cylinders, where the expansion of the working fluid in one cylinder is connected to the compression space of the other cylinder via a regenerator. This arrangement is proven to be significantly better than the other configurations in terms of the engine’s power output [17].
2.1.4 Applications

Cogeneration, or CHP technology, is the production of more than one useful form of energy, most commonly heat and power, from a single fuel source. As a result, CHP systems exhibit an increase in the efficiency of energy conversion, which is accompanied with a net reduction in greenhouse gas emissions and costs [18]. Micro-CHP systems are an extension of CHP systems which are used for residential applications (< 20 kW), providing heat for space heating and electricity for consumption or net metering back into the grid. Micro-CHP systems offer many advantages which include using the electricity generated, thus the amount of electricity that must be drawn from the local distribution network [19]. Using a micro-CHP system to produce electricity and heat close to where they are to be utilized, results in much more efficient use of fuel, which reduces overall emissions and cost [19]. Micro-CHP systems are designed to fulfill the thermal load for residential applications with electricity being a by-product.

Stirling engines have several traits which make them more suitable for CHP, especially micro-CHP, applications than internal combustion engines or fuel cells. These benefits have been outlined in the beginning of Section 2.1. However, Stirling engines are more expensive to produce and offer lower electrical efficiencies than internal combustion engines [8, 20]. But the overall efficiency, for heat and electricity, is significantly higher than the other options available.

2.1.5 Applications with Alternative Fuels

As mentioned previously, Stirling engines can operate on a wide variety of fuel or energy sources. Aside from petroleum based fuels, fuels such as biomass and energy sources such as solar and waste heat have also been used in several studies.

Biedermann et al. [21] developed a 35 kW_{el} and 220 kW_{th}, small scale CHP plant based on a Stirling engine fuelled by solid wood chips, and reported on operational experiences. The engine features four cylinders, in a square configuration, and helium is the working fluid, at a pressure of 4.5 MPa. The system featured a two stage combustion chamber for low NOx. Hot end temperatures were reported to be in the range of 680 to 780 °C. The primary combustion air was preheated to approximately 360 °C using the flue gas leaving the exhaust, thus increasing the
temperatures and increasing the overall efficiency. The system was fully automated and tests were conducted for 5000 hours without any major issues. The major concern posed by the authors was clogging of the narrow passages of the heat exchangers due to the ash content of the feedstock and the particles resulting from combustion. Clogging and deposition on the heat exchangers would have a detrimental effect on heat transfer into the working fluid, thus reducing the work output.

The same group addressed the issue of clogging and deposition in another study [22], where they implemented a pneumatic and fully automated cleaning system which would blast compressed air through nozzles in order to unclog the passages and remove any deposits. They found that using traces of sand and grit mixed with the compressed air proved more effective is removing the deposits. Even with the implementation of this cleaning system, manual cleaning was necessary every 500-600 hours of operation. This is unavoidable, but the goal is to prolong the manual maintenance period for as long as possible. Electrical and overall efficiencies of the system were reported to be 9.2% and 90%, respectively. The results indicate approximately 20 to 95 mg/m³ of NOₓ, 35 to 135 mg/m³ CO, and 20-40 mg/m³ Unburned Hydrocarbons (UHC), and negligible particulates.

A similar study performed by Carlsen et al. [23], focused more on the design issues and considerations on a 40 kW_{EL} Stirling engine also fuelled by wood chips. The working fluid was helium at a pressure of 4 MPa. The combustion chamber was designed similarly to that of a wood chip burning furnace. The combustion chamber featured a long section so as to increase the residence time for thorough burnout of the fuel. The issue of heat exchanger clogging and ash deposition was addressed again, and a unique solution was provided. From the main combustion chamber, the direction of the fuel gases changes direction before flowing into the Stirling engine hot end heat exchangers. This minimizes the amount of larger particles which enter the Stirling engine as depicted in Figure 2.4. The distance between the heat exchanger fins and tubes were kept relatively large in the design. Another issue which was discussed was the difficulty in implementing a control strategy which could address the changes in feedstock variations, or in other words, variations in the heating value. This is a major issue for the Stirling engine because large variations in temperatures lead to changes in the power output. Electrical and overall efficiencies of the system were 21% and 85%, respectively.
The two major studies outlined above are among the many which are available on Stirling engine operation on solid fuels and biomass [24-29]. The major issue which consistently arises is heat exchanger fouling and ash deposition, the effects of which have been studied by Kuosa et al. [30] in detail, within the context of biomass as the fuel. They developed a numerical model, using a fouling factor which can be calculated from experiments, for finding the effects of fouling on efficiency, performance, maintenance, and cost. At high temperatures, the ash in the exhaust begins to melt and create a residue which builds onto the heat transfer surfaces of the Stirling engine [30]. As expected, the power output of the engine decreases significantly as the fouling increase [30]. This however leads to a higher amount of recoverable heat after the flue gas passes through the heat exchanger.

2.2 WhisperGen DC micro-CHP

2.2.1 Specifications

The commercial Stirling engine used for this study is a diesel fuel-fired micro-CHP unit developed by Whisper Tech Ltd, primarily designed for marine applications. Figure 2.5 shows the engine in its original configuration and housing. Its nominal power and heat output are rated
at 0.8 kW and 5.5 kW, respectively. The engine has four-cylinders in a FCDA configuration, which contain nitrogen as the working fluid at a pressure of 2.8 MPa.

**Figure 2.5: Whispergen DC micro-CHP System.**

The working fluid is heated by an external combustion chamber, which houses an evaporative burner, and is cooled using a circulating coolant. This repeated heating and cooling of the working fluid leads to changes in pressure, which cause the piston to move creating mechanical motion. The mechanical motion, via a patented “wobble yoke” mechanism, is used to rotate an alternator which produces an AC electrical power supply. The AC electrical power supply is sent through a rectifier for conversion to DC electrical power. This electricity is used to charge a lead-acid battery, which can then be used to power electrical appliances such as a refrigerator, via an inverter. Table 2.2 summarizes the detailed specifications of the engine.

Figure 2.6 shows a component schematic of the original laboratory set-up used in previous studies [31-32]. The exhaust from the combustion chamber passes through a plate heat exchanger which dumps the heat from the exhaust in the same coolant which exits from the engine block. The cooled exhaust exits the hot end heat exchangers and the air preheater at 60 to 70 ºC, and is vented outside. A shell and tube heat exchanger is employed to load the thermal output from the coolant using cold water. The engine houses various sensors to ensure optimal operation.
The unit was designed for marine applications where it would provide an auxiliary source of electricity and the primary purpose would be for heat production. This heat could be used for space heating or hot water generation. The unit requires servicing every year, or 3000 hours of operation, whichever is first. The manufacture advises not to use this as the primary source of any electrical power.

**Table 2.2: Whispergen micro-CHP specifications [16, 33].**

<table>
<thead>
<tr>
<th>Feature</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Configuration</td>
<td>FCDA with kinematic wobble yoke mechanism</td>
</tr>
<tr>
<td>Cylinders</td>
<td>4</td>
</tr>
<tr>
<td>Working Fluid</td>
<td>Nitrogen</td>
</tr>
<tr>
<td>Working Fluid Pressure</td>
<td>2.8 MPa</td>
</tr>
<tr>
<td>Heat Output</td>
<td>5.5 kW (Nominal)</td>
</tr>
<tr>
<td>Power Output</td>
<td>0.8 kW (Nominal)</td>
</tr>
<tr>
<td>Fuel</td>
<td>No. 2 (Diesel)</td>
</tr>
<tr>
<td>Fuel Flowrate</td>
<td>1 L/hour</td>
</tr>
<tr>
<td>Size</td>
<td>450 (W) x 500 (D) x 650 (H) mm³</td>
</tr>
<tr>
<td>Engine Speed</td>
<td>1200-1500 RPM</td>
</tr>
</tbody>
</table>

**Figure 2.6: Schematic of the WhisperGen System [31].**

2.2.2 Modes of Operation

The WhisperGen has three modes of operation: auto charge, heat management, and maintenance. In the auto charge mode, the engine starts automatically when the battery charge falls below a specified level, and automatically stops when the battery is completely charged. In
the heat management mode, the engine maintains a set coolant temperature specified by the user, regardless of the battery charge. The burner power level and the generator output voltage are controlled to meet the heat management and battery charging requirements. If the battery is completely charged and the engine continues to operate, the excess electricity produced is used to activate a clamp heater which increases the temperature of the coolant. Aside from the two normal operating modes, the engine can also be operated in a test mode, for maintenance purposes, which overrides all other settings.

Figure 2.7: Battery charging cycles [33].

The battery has three charging cycles, bulk-charging, absorption-charging, and float-charging. The amount of current and voltage supplied to the battery is shown in Figure 2.7. During the bulk charge phase, the battery is accepting all the available charge, until the charge reaches 80%. After that, the battery enters the absorption-charge phase, where the current supplied is limited to the amount which the battery is able to absorb at a specific voltage. After the battery is completely charged, it enters the float charge phase where it is kept at a lower constant voltage to compensate for self-discharging in order to maintain full charge.

2.2.3 Sensor Operation

Engine operation is controlled by a complex feedback loop which looks to satisfy the energy demand based on the operating mode. Appendix A shows the circuit diagram for the WhisperGen’s controller. During operation, the control systems constantly optimize the system parameters to maximize combustion efficiency and minimize exhaust emissions. Figure 2.8
shows a schematic of the various components and control/data interaction within the system. The control systems can detect a number of possible fault conditions relating to the system components, and in most cases (i.e. volatile faults), will automatically go through a shut down and restart sequence to try and recover from the fault condition.

Figure 2.8: Whispergen micro-CHP control/data transfer.

Air and fuel are the inputs into the combustion chamber. Air is brought into the system using a Pulse Width Modulation (PWM) speed controlled fan. When more power is required, the fan speed is increased by the control systems. If the amount of power is to be decreased, or the exhaust temperature exceeds the set point, then the fan speed is decreased. The fan is capable of providing a maximum flow rate of about 350 L/min. The fuel is controlled using a piston/solenoid type positive displacement pump which also uses a PWM signal. The fuel flow rate is controlled by the control systems and is increased when the oxygen content in the exhaust increases above the set point. Normal operation of the pump falls between 2 to 16 Hz, corresponding to a maximum fuel flow rate of 19 mL/min for diesel.
Components attached to the evaporative burner include a 400 W glow plug, flame rod, and a thermocouple. A glow plug is essentially a heating element which, due to its low electrical resistance, radiates heat when power is supplied. It is used during start-up in order to preheat the evaporator burner and combustion chamber to vaporize the fuel and ignite the fuel-air mixture. Once temperatures are stable within the combustion chamber, the glow plug is turned off. A flame rod is an insulated electrode that protrudes into the flame and detects if the flame is present [34]. The principal of operation relies on ionization, or separating electrons from the atoms resulting in an ion during the combustion process due to the high temperature. The free electrons resulting from ionization move towards the outer zone of the flame, whereas the heavier positive ions shift towards the base of the burner. Therefore the flow of current through the flame rod can be correlated to the flow of electrons through the flame rod. The control systems measure the current through the flame rod, which starts from 0 μA signifying no flame, and increases towards a maximum value of 10 μA as the intensity of flame increases. This signal is used to determine that no flame is present prior to ignition, that ignition has taken place once the fuel pump is initiated, and that a flame is present during start up and operation of the burner. If at any time during operation the signal is 0 μA, signifying no flame present, the engine will shut down. The temperature of the exhaust leaving the combustion chamber, just before it goes into the exhaust heat exchanger, is measured using a K-type thermocouple. The exhaust temperature value plays a key role in engine operation as it is used by the control systems for several purposes. The value of the exhaust temperature is used to determine when to crank the engine during the heat up stage, when to shut off the glow plug and the amount of air required through the system during the ignition phase. The exhaust temperature is set to about 480 ºC, for normal operating conditions and is not to exceed 545 ºC.

A zirconium dioxide (ZrO$_2$) oxygen sensor (P.N. Honeywell GMS-10-18) is placed immediately after the exhaust temperature thermocouple to measure the oxygen concentration in the exhaust gas. ZrO$_2$ sensors are based on a Nernst cell, which is a solid-state electrochemical fuel cell. Its two electrodes provide a voltage which can be related to the oxygen concentration in the exhaust gas. The sensor is calibrated before operation by flowing fresh air through and assuming an oxygen content of 21 % vol. It is employed to control the fuel-air equivalence ratio after the fuel pump is initiated. The sensor set point is about 8.5 – 9.5 % vol. of O$_2$, which corresponds to an air to fuel equivalence ratio of approximately 0.52 for diesel.
The data sent from the Stirling engine to the control systems includes temperature of the hot end heat exchanger and the coolant exiting the engine block. The current produced by the alternator is also monitored. The load dump, or the clamp heater, is activated when the battery is above an 80% charge state. The excess power produced is then used to activate a resistive element in the engine block which increases the temperature of the coolant. The voltage of the battery is also monitored, and used, along with the current produced by the alternator, to calculate the total power being generated by the engine as shown in Equation 2.4. In this equation, $P_{OUT}$ represents the power output, $V_{BAT}$ is the battery voltage, and $I_{ALT}$ is the current produced by the alternator. The power output value displayed by the engine’s controller does not take into account the consumption of power by the pump, fan and electronics.

$$P_{OUT} = V_{BAT}I_{ALT}$$  \textbf{Equation 2.4}

2.2.4 Engine Operation

The engine goes through a sequence of steps during startup, steady state operation, and shut down. When in idle mode, the engine is disconnected from the battery, but the charge is constantly being monitored by sensing wires. Once the engine receives the start signal, battery power is made available to power auxiliary devices such as the coolant pump, fan, and glow plug. A pre-check is executed where the data from the sensors is tested for correct values. The combustion chamber is then heated to about 400 ºC using the glowplug and the fuel pump in initiated. A signal from the flame rod is continuously monitored for the presence and stabilization of the flame. The engine is cranked after a stable flame is detected and temperatures inside the combustion chamber are approximately 450 ºC. During the run mode, the power and heat output are controlled to meet the operation and battery charging requirements. All variables are continuously monitored and some are modified to ensure optimal operation. If a stop signal is received, the shutdown sequence is initiated. The fuel pump is shut off and the air is decreased for a short period so the working fluid can extract heat for power. The fan speed is then increased, to maximum, to assist in the cooling of the combustion chamber, thus bringing the engine to a stop.
2.2.5 Combustion Chamber and Evaporative Burner

The combustion chamber of the WhisperGen is a unique design which features an exhaust heat exchanger. Figure 2.9 [35] shows a section schematic of the combustion chamber with the inlet and outlet highlighted. The combustion chamber is surrounded by a series of stainless steel sheet metal pieces which are welded together in a concentric configuration. Ambient air enters the chamber through the inlet, shown in blue, and passes through the shells where it is directed into the fixed swirl evaporative burner shown in Figure 2.10. Due to the high temperatures in this region, initially from the glow plug, the fuel vaporizes in the evaporative burner and creates a lean mixture with air. This mixture continuously enters the main combustion section, ignites, and the flame flattens out due to the presence of the diffuser and the recirculation created by the swirl.

![Combustion Chamber Diagram](image)

Figure 2.9: WhisperGen’s Combustion chamber [35].

The products of combustion, in red, are then passed through the hot end heat exchangers of the Stirling engine where heat is transferred to the working fluid, and then through the cavities of the burner to the outlet. During the passage to the exhaust, heat is transferred into the incoming air, thus preheating the incoming air to 500 °C at steady state operation. Preheating the
incoming air increases the flame and exhaust gas temperatures leading to a higher efficiency. These higher temperatures also lead to faster fuel evaporation, higher oxidation rates and thus higher combustion efficiency. At the base of the burner is a refractory ceramic plate which acts as insulation and provides an excellent source of radiant heat transfer to aid in preheating and the ignition of fuel. The system is so well designed that the external shell of the combustion chamber, the one which is exposed, is at ambient temperatures, thus leading to minimal heat losses.

Figure 2.10: WhisperGen’s evaporative burner [35].

2.2.6 Hot End Heat Exchangers

Increasing the rate of heat transfer from the combustion gasses to the working fluid can have a significant impact on the overall efficiency of the system. The rate of heat transfer from the combustion gasses to the hot end heat exchanger can be calculated using Equation 2.5 [16], where $T_H$ represents the hot end temperature, $T_{GAS}$ represents the temperature of the combustion gasses, and $R_T$ is the thermal resistance. Using this relationship, there are several means by which the rate of heat transfer can be increased. By using preheat, the flame temperature will increase. Ensuring that the flame is placed close to the heat exchanger surface will increase the surface temperature. Radiation from the surrounding surfaces can be maximized by using a refractory lining. Increasing the surface area of the heater head, by using fins, also has a significant effect on the rate of heat transfer. It has been reported that finned surfaces increase the heat transfer rates in the heater head up to by three times [16]. However, the spacing between the fins has to be such that it does not introduce a large pressure drop, which makes it difficult for naturally aspirated engines to operate [16].
The hot end heat exchangers of the WhisperGen are shown in Figure 2.11. The exhaust enters these annular finned slots, made from high-temperature stainless steel and transfers heat into the working fluid to facilitate the motion of the pistons. These fins provide a high surface to volume ratio to increase heat transfer through convection and radiation. Surrounding the ceramic plate is a rope seal, which prevents the exhaust gasses from bypassing the hot end heat exchangers as this would lead to a decrease in heat transfer to the working fluid and thus lower power output.

\[ Q_{HEATER} = \frac{(T_H - T_{GAS})}{R_T} \quad \text{Equation 2.5} \]

Figure 2.11: Hot end heat exchangers.

2.2.7 Operation on Alternative Fuels

Aliabadi et al. [32] performed a detailed efficiency and emissions analysis on this model with diesel and biodiesel. Operation with diesel resulted in power and thermal efficiencies of 11.7% and 78.7%, respectively. No modifications were required for the conversion to bio-diesel, and efficiencies were reported to be 11.5% power and 77.5% thermal, slightly lower than diesel. However, the particulate emissions for bio-diesel were an order of magnitude higher at 69.2 mg/kW·hr, whereas with diesel they were approximately 2.3 mg/kW·hr. CO, NOx, and UHC were comparable. The discrepancy in the performance of the system from diesel to bio-diesel is
attributed to the presence of more higher boiling point compounds in bio-diesel. This property has an effect on flame stability, fuel evaporation, and the complete burnout of the fuel.

Another study on the same engine [31] compared the efficiency and emissions with diesel and ethanol (EtOH). Modifications to the fuel system had to be made because the heating value of diesel is greater than that of EtOH, thus requiring a greater volumetric fuel flow rate (approximately 1.7 times) to achieve the same energy input. When operated on diesel, the power and thermal efficiencies, based on the lower heating value (LHV), were 12.1% and 73.3%, respectively. Operation on EtOH resulted in efficiencies of 11.7% power, and 73.7%. Emissions for EtOH were slightly lower than that for diesel. The difference in performance is attributed to the lower flame temperature of EtOH, which results in a lower heat transfer rate to the working fluid. However lower power efficiency was compensated by a higher thermal efficiency due to the increase in the exhaust temperature. The discrepancy between the diesel efficiency values between the two studies was due to the difference in operation modes. In the Aliabadi et al. [32] study, the engine was operated under conditions where the battery entered absorption mode in which power was used to create heat. In the other study, the engine was operated under the bulk-charging mode during operation.

Pourmovahed et al. [36] used a similar model of the WhisperGen, which is fuelled by natural gas, to operate on biogas with a composition of 62% methane, 37% carbon dioxide, and 1% nitrogen. The fuel supply was switched from natural gas to biogas and no modifications were made. The overall output for natural gas was significantly higher than biogas. The fuel flow rates were the same in both tests, although the heating values differed, with that for natural gas being greater than that for biogas. At steady state operation, the power and thermal efficiencies were very similar at about 6% and 65%, respectively, with natural gas being slightly higher. A preliminary emissions analysis determined the emissions for natural gas were higher than biogas.

2.3 Fast Pyrolysis Liquid

This section serves as an overview of woody biomass derived fast pyrolysis liquid, a more detailed literature review pertaining to combustion applications can be found elsewhere [37].
2.3.1 Production

Fast pyrolysis liquid, or bio-oil, is a liquid fuel derived from biomass, which is composed of more than 300 compounds [38]. The three primary components of biomass are cellulose, hemicellulose, and lignin. The fast pyrolysis process thermally decomposes biomass, in the absence of oxygen, to produce vapors, aerosols, and char [39]. The vapors are then condensed, via cooling, to produce bio-oil. High liquid yield in the process is based upon fulfilling the following requirements [38-39, 40-41]:

1. High heat transfer rates to the biomass (requiring small biomass particles).
2. Intermediate reactor temperatures of 400 to 500 °C.
3. Vapor residence time of < 2 seconds.
4. Rapid quenching or cooling into the final liquid product.

Figure 2.12: Fluid bed fast pyrolysis process [42].

Although there are numerous reactor configurations which can be used for the fast pyrolysis process, fluidized beds depicted in Figure 2.12, are the most common due to ease of operation [42]. The process commences with the drying of wet biomass, to about 10 wt. % [39].
This in turn minimizes the amount of water seen in the products. To obtain faster reaction, the biomass goes through a grinder where the particles size is reduced to 2 mm [42]. The solid biomass is then put through the bubbling fluidized bed reactor and passed through the cyclone to separate the gasses and particulate matter. The gases are then condensed into bio-oil. Process heat is used to fulfill the thermal requirement in the process.

2.3.2 Fuel Properties

Bio-oil is characterized by two phases, an aqueous (water soluble) phase and a non-aqueous (water insoluble) phase in a ratio of 3:1 by weight percent, respectively. The unique chemical composition of bio-oil leads to properties which are very distinct from other petroleum based fuel oils. Table 2.3 shows the properties of bio-oil along with two other conventional petroleum fuel oils, No. 2 and No. 6.

**Table 2.3: Physical properties of bio-oil and petroleum fuel oils [41, 43-47].**

<table>
<thead>
<tr>
<th>Physical Properties</th>
<th>Units</th>
<th>Bio-oil</th>
<th>No. 2 Fuel Oil</th>
<th>No. 6 Fuel Oil</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon</td>
<td>wt. % dry</td>
<td>50-60</td>
<td>87</td>
<td>85</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>wt. % dry</td>
<td>5-7</td>
<td>13</td>
<td>11</td>
</tr>
<tr>
<td>Oxygen</td>
<td>wt. % dry</td>
<td>35-40</td>
<td>-</td>
<td>1</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>wt. % dry</td>
<td>0-0.3</td>
<td>&lt;0.01</td>
<td>0.3-0.5</td>
</tr>
<tr>
<td>Sulfur</td>
<td>wt. % dry</td>
<td>0-0.5</td>
<td>0.1</td>
<td>1</td>
</tr>
<tr>
<td>Ash</td>
<td>wt. %</td>
<td>0-0.03</td>
<td>-</td>
<td>0.1</td>
</tr>
<tr>
<td>Solids</td>
<td>wt. %</td>
<td>0.2-1</td>
<td>-</td>
<td>0.1-1</td>
</tr>
<tr>
<td>Moisture</td>
<td>wt. %</td>
<td>15-30</td>
<td>-</td>
<td>0.1-7</td>
</tr>
<tr>
<td>Acidity</td>
<td>pH</td>
<td>2-3</td>
<td>7</td>
<td>7</td>
</tr>
<tr>
<td>Density</td>
<td>kg/m3</td>
<td>1200</td>
<td>830-860</td>
<td>900-1000</td>
</tr>
<tr>
<td>LHV</td>
<td>MJ/kg</td>
<td>13-18</td>
<td>41-43</td>
<td>40-43</td>
</tr>
<tr>
<td>Viscosity</td>
<td>cSt at 40 °C</td>
<td>10-100</td>
<td>2-6</td>
<td>200-2000</td>
</tr>
<tr>
<td>Flash Point</td>
<td>°C</td>
<td>60-100</td>
<td>60-90</td>
<td>60-80</td>
</tr>
<tr>
<td>Evaporation Residue</td>
<td>wt. %</td>
<td>Up to 50</td>
<td>0.1</td>
<td>1</td>
</tr>
</tbody>
</table>
A high amount of variability is seen in the fuel properties of bio-oil. Various parameters in the production process, namely the biomass type, reactor temperature, residence time, and heating rate profiles, can have an effect on the fuel properties. Oxygen is present in most of the 300 plus compounds found in bio-oil, and makes up 35-40 wt. % of bio-oil [43]. Higher oxygen content leads to a lower energy per unit mass and immiscibility with other hydrocarbon fuels. Among the many compounds present in bio-oil, water is the most abundant. The source of water is the original moisture content from the feedstock and the hydrogen and oxygen content in the fuel. High water content results in lower viscosity which is beneficial for atomization and pumping [43]. The presence of water also lowers the heating value, flame temperature, and increase the ignition delay. The lower temperature does, however, decrease the formation of thermal NOx. There are significant amounts of non-volatile components, including sugars and oligomeric phenols [43]. When heated, bio-oil stops evaporating at about 250-280 °C leaving about 35-50% residue [43]. A decrease in viscosity occurs at high temperatures, at rates which are higher than most petroleum based fuels [43]. Significant reductions in viscosity can also be made with the inclusion of polar solvents such as methanol or EtOH [43]. Bio-oil is acidic in nature due to the presence of organic acids [43]. The heating value for bio-oil is significantly lower than that of conventional petroleum based fuels on a mass basis, but comparable on a volumetric basis [41]. The lower heating value of bio-oil is approximately 20 MJ/L, which is approximately 40% less than that of diesel. On a mass basis, this difference is approximately 60%.

2.3.3 Applications in Engines

*Single Droplet Combustion*

Before discussing the applications of bio-oil in engines, it is important to understand its unique combustion process. Single droplet studies of bio-oil, performed by Wornat et al [48], have shown a process of combustion comprising of four stages depicted in Figure 2.13. The first stage is the quiescent burning phase where the highly volatile surface compounds are burned leaving behind less volatile material with higher molecular weight compounds. Due to the high heat, the surface of the droplet tends to polymerize forming a shell under which volatile compounds exist. The shell restricts the diffusion of these volatile components to the outer
surface, and causes the volatile components to heat leading to microexplosions. The droplet becomes fragmented while the volatiles exit and result in a sooty burnout for the remainder of the burnout time. This characteristic droplet combustion is similar to other residual fuel oils emulsified in water which causes the microexplosions.

**Figure 2.13: Burnout sequence of bio-oil droplet solid residue [48].**

*Small Scale Burners*

Tzanetakis et al. [49] have performed a comparative study between diesel and bio-oil in a 10 kW atmospheric burner, and have found that the emissions for bio-oil are higher than those of diesel. They attribute this to the fact that diesel is fully distillable, can achieve better atomization and has improved spray ignition characteristics than bio-oil. NO\textsubscript{x} present in the exhaust is primarily from the inherent nitrogen content, whereas for diesel it is formed due to the thermal mechanism. CO is also higher for bio-oil, which is again due to the thorough burnout of diesel which is possible at a wide range of operating conditions.

*Diesel Engines*

Due to the non-distillable nature of bio-oil, applications in diesel engines are restricted to direct injection. There are several studies reported in the literature ranging from 5 kW to 1.5 MW. Issues identified in these studies include poor ignition qualities of bio-oil, corrosion of various components (i.e. injector needle, fuel pump, and orifice), injector clogging due to solid residue, and coke deposition within the cylinder [41, 43, 50].

The issue of poor ignition is addressed, to a certain extent, using cetane improving additives which are typically nitrated alcohols added in concentration of 5-10 % vol. [51-52].
Other studies have included air preheat or a diesel pilot flame to improve ignition and combustion within the cylinder [43, 53]. Corrosion is preventable if the components which interact with the bio-oil directly are made from stainless steel [47]. Erosion of components from the char particles is preventable by reducing the total solids content of the fuel through varying the production process [47, 50]. Engines would be started with diesel, switched to EtOH, and then to bio-oil. Cetane improved EtOH would be periodically used to clean the injector nozzle, and prevent it from becoming clogged [51-53]. Bio-oil exhibits longer ignition delay periods, even with the inclusion of cetane improving additives, than diesel [52-53]. Once the fuel ignites, burn durations are very short [51-52]. Due to the increased ignition delay in bio-oil, the primary mode of combustion is premixed in the cylinder [50, 53]. An important point to note is that the longer ignition delay is a result of slow chemical kinetics and not physical processes such as poor atomization, evaporation and mixing [53].

In terms of performance, results show low cyclic variability and a similar thermal efficiency as diesel [53]. Emissions in most studies show comparable or higher results in terms of UHC, NOx, CO, and PM [41, 51-52]. It is recommended that pure bio-oil be used with low to medium speed diesel engines that have high compression ratios [43, 51-52], but that blends (including methanol or EtOH, with additives) could be used with high speed engines [43].

Gas Turbines

Preliminary studies with bio-oil have been performed on gas turbine engines ranging from an electrical output between 40 kW_{EL} to 2.5 MW_{EL}, with run conditions from idle to full-load. Issues raised in these studies were the corrosion of construction material, turbine blade erosion through high PM loading in the exhaust, and hot alkali corrosion of the blades from the ash content in the fuel [43, 54].

The fuel was filtered and heated in order to reduce viscosity, which improves atomization quality to achieve improved burnout of fuel droplets [55-58]. Filtration of the fuel helps to reduce the PM in the exhaust. Even with this measure taken, significant deposition was observed on the combustion chamber surfaces and turbine blades after operation with bio-oil. Results have indicated higher CO, UHC, and PM emissions compared to the original fuel, whereas NOx emissions are comparable or lower. The higher emissions for bio-oil are attributed to the lower atomization quality, evaporation rates, and mixing within the combustion chamber.
**Stirling Engines**

In the literature, only one study of Stirling engines fuelled by bio-oil is available [59]. The group modified a “flameless oxidation” (FLOX®) burner, originally meant for propane gas, to burn fast pyrolysis liquid and attached it to a 25 kW SOLO Stirling CHP system. When furnace temperatures are about 1000 ºC and air preheated to 650 ºC, it creates conditions under which no flame is present, hence flameless oxidation, as depicted in Figure 2.14 [60]. Under these conditions, the fuel completely oxidizes leaving very little CO and no NOx due to the relatively low temperatures with the absence of a flame [60]. The FLOX® burner is suitable for low quality fuels such as fast pyrolysis liquid because of a large residence time leading to increased oxidation rate. The spray burner featured a pressurized air atomizer with an external mixing chamber. The study reported higher thermal and electrical efficiencies than other fuels in a similar power range. Results indicated approximately 20 to 95 mg/m³ NOx, 35 to 135 mg/m³ CO, and 20 to 40 mg/m³ UHCs, and negligible particulates.

![Figure 2.14: Nozzle of the FLOX® burner in operation [60].](image)

2.3.4 **Design principles for Bio-oil Combustion in Engines**

Because of the acidic nature of bio-oil, the components which interact with the fuel should be made from acid resistant materials such as stainless steel or high density plastics such as Teflon®. These components include fuel lines, nozzle, and all associated pipe fittings and connections. At standard temperatures of 25 ºC and depending on the fuel properties, bio-oil can be very viscous. To reduce the viscosity, fuel preheat should be utilized, to bring the temperature of the fuel between 60 to 90 ºC. Care has to be given to the heating process as bio-oil has a
tendency to polymerize at a certain temperature range, for extended periods of time. Since bio-oil does not completely evaporate, a suitable residence time has to be chosen for thorough burning of the fuel. The quality of bio-oil used for engine must be very high. Properties should include a low ash, char, solids and residue.
Chapter 3

Combustion Chamber Design

3.1 Overall Assembly

Since bio-oil does not completely evaporate, the original evaporative burner of the WhisperGen had to be replaced. A pilot stabilized swirl burner was chosen based on a detailed study on small scale bio-oil combustion [49]. The combustion chamber features five main components: swirl box, fuel nozzle, pilot flame system, diffuser section, viewport and exhaust section. Drawings and dimensions can be found elsewhere [37].

Figure 3.1 shows the layout of the burner assembly, with all the major components labeled. The nozzle is placed within the burner assembly, pointing downwards, and the primary air flow is from top to bottom. This configuration also negates the effects which buoyancy would have on the flame if it were in a horizontal configuration. A vertical burner with the spray pointed upwards is also common geometry [61, 62], but would not work well as there is a potential for large amounts of PM to fall back towards the intake.

The air box, swirl generator, and the swirl throat are fabricated from aluminum since they are only exposed to air, whereas the remaining components are made from stainless steel. The main combustion chamber has 3.2 mm thick walls with no refractory lining. Sealing between the flanges of the various components is with either silicone rubber gaskets (P.N. 8525T44,
McMaster-Carr), for regions where the temperature is below 260 °C or graphite gaskets (P.N. 95715K73, McMaster-Carr) for areas of high temperatures. Areas of high temperature include all components after the swirl throat.

![Schematic of combustion chamber assembly.](image)

**Figure 3.1: Schematic of combustion chamber assembly.**

### 3.2 Variable Swirl Generator

#### 3.2.1 Swirling Flows

Swirling flows are characterized by having components of velocity and momentum in the angular direction. The presence of an angular velocity component in flows, if large enough, can induce a large radial pressure gradient which forms an area of low pressure in the center of the flow. The decay of these large gradients downstream creates a strong axial pressure gradient in the direction of the bulk flow. These reverse axial gradients force the fluid downstream, and then back upstream towards the low pressure region. This phenomenon, shown in Figure 3.2 [63], leads to the creation of a central recirculation zone (CRZ).
In the context of combustion, the presence of the CRZ offers many advantages, such as recycling hot products back towards the base of the flame [63] which aids in ignition and overall stability. The turbulence created by swirling flows is beneficial for the mixing of fuel and air [63]. As a result, the length of the flame decreases and lean blow-out limit increases. Recirculation also leads to longer residence times for exhaust products which induce a thorough burnout of the fuel and reduce UHC, CO and PM emissions [63]. These advantages have led to the use of swirling flows in many combustion applications such as diesel engines, gas turbines, and boilers. One disadvantage which makes implementing swirling flows in combustion applications difficult is the promotion of combustion instabilities [64], which will be discussed later in detail.

Figure 3.2: Central recirculation zone formed in swirling flows [63].

Swirling flows are quantified by a non-dimensional number, $S$, which is referred to as the swirl number. The swirl number is the ratio between the angular momentum ($G_\phi$) and axial momentum ($G_x$) of the flows as shown in Equation 3.1. The variable $r$ represents the radius of the plane in which swirling flows exist. The angular and axial momentum terms are conserved and are defined in Equation 3.2 and Equation 3.3, respectively. The terms $U$ and $W$ represent the axial and tangential velocity respectively, and $p$ represents the static pressure along any cross-sectional plane in the flow field. Under cold flow conditions, the lowest value of $S$ which will induce a negative axial pressure gradient thus creating a CRZ, is between $0.5 \leq S \leq 0.6$ [63, 65].
3.2.2 Movable Block Swirl Generator

The creation of swirl in the combustion chamber was accomplished through a movable block [66] type swirl generator depicted in Figure 3.3 [49]. Primary air enters through the air box, after which it goes through a block assembly. The block assembly is composed of two different types of blocks, ones which are fixed and others which are movable. By varying the opening size of these blocks, the amount of swirl can be varied as shown in Figure 3.4. When the tangential passages are closed and the radial passages are open, no axial momentum is imparted on the flow, thus the swirl number is equal to zero. On the other hand, the maximum capable swirl number for this specific configuration is achieved if the tangential passages are open and the radial passages are closed. Swirl numbers which lie between the two extremes mentioned are also possible by adjusting the amount which each passage is open. Table 3.1 shows the geometric specifications corresponding to the swirl generator used for the combustion chamber.

\[
S = \frac{G_\theta}{G_x r} \quad \text{Equation 3.1}
\]

\[
G_\theta = \int_0^R (Wr)\rho U 2\pi r \, dr = \text{const.} \quad \text{Equation 3.2}
\]

\[
G_x = \int_0^R U\rho U 2\pi r \, dr + \int_0^R p 2\pi r \, dr = \text{const.} \quad \text{Equation 3.3}
\]

Figure 3.3: Details of swirl generator, nozzle and burner inlet geometries (in mm) [49].
Figure 3.4: Geometry of movable block type swirl generator [49].

Table 3.1: Geometric specifications for movable block type swirl generator [49].

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Design Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$n$</td>
<td>Number of swirl blocks</td>
<td>8</td>
</tr>
<tr>
<td>$R$</td>
<td>Swirl generator exit radius</td>
<td>76.2 mm</td>
</tr>
<tr>
<td>$R_h$</td>
<td>Swirl generator inner radius</td>
<td>9.53 mm</td>
</tr>
<tr>
<td>$B$</td>
<td>Depth of swirl blocks</td>
<td>38.1 mm</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>Fixed swirl block angle</td>
<td>60°</td>
</tr>
<tr>
<td>$\xi$</td>
<td>Adjustable swirl block angle</td>
<td>-</td>
</tr>
<tr>
<td>$\xi_m$</td>
<td>Maximum opening angle</td>
<td>12°</td>
</tr>
</tbody>
</table>

The theoretical swirl number for this configuration was estimated using the correlation in Equation 3.4 [65-67], with values ranging from 0 to 5.41. Although this expression does give a general ideal of the swirl number, it is derived using ideal conditions, which make it susceptible to error. A detailed discussion can be found elsewhere [37].

\[
S \sim \frac{2\pi}{n\xi_m} \sin \alpha \frac{\cos \alpha \left[ 1 + \tan \alpha \tan \left( \frac{\xi}{2} \right) \right]}{\left[ 1 - \cos \alpha \left( 1 + \tan \alpha \tan \left( \frac{\xi}{2} \right) \right) \right]^{\frac{1}{2}}} \frac{r}{2B} \left[ 1 - \left( \frac{R_h}{2B} \right)^2 \right]
\]

Equation 3.4
3.3 Fuel Atomizing Nozzle

Atomization of the fuel was accomplished through a stainless steel internal mix, air-blast nozzle (P.N. JX6BPL11, BEX Engineering Ltd.) with a 6 in. extension (P.N. 1/4”JX6BPL11, BEX Engineering Ltd.). The tip of the nozzle was placed in close proximity to the pilot flame, at the centerline of the burner in order to assist in ignition. Figure 3.5 shows the nozzle tip assembly which features two components, a liquid (P.N. JL40100, BEX Engineering Ltd.) and air (P.N. JPG60, BEX Engineering Ltd.) cap. Fuel enters the mixing chamber through an orifice in the liquid cap which is 1.0 mm in diameter. Compressed air is brought into the mixing chamber through the air cap, where it mixes with the fuel and forces this mixture through six evenly spaced orifices. The final fuel spray features a hollow cone pattern with six individual fuel-air jets, which are 65 degrees apart. A hollow cone spray is more conducive to allowing the hot recycled gases to move upstream, because of the lack of axial momentum which would penetrate through the CRZ. The total angle of the spray is approximately 65°.

Figure 3.5: Internal mix nozzle tip assembly [68].

3.4 Pilot Flame System

The pilot flame is composed of a premixed methane and oxygen flame torch (P.N. 110-406, Contenti). The central orifice of the pilot tip is surrounded by small openings which result in a larger flame diameter. The torch, which has a tubular shape, is inserted at the top part of the diffuser, right below the flange, and is sealed using a compression fitting. The tip of the torch remains in place and does not impinge into the flame. The energy input corresponding to the maximum methane flow rate of 0.88 L/min, corresponding to 0.5 kW of energy [49]. The pilot
flame system is operational throughout the duration of a test in order to stabilize combustion and provide ignition during periods of flame blowout.

With regards to obstructions, the pilot flame adds momentum which can hinder the flame structure as seen in Figure 3.6. Therefore, an alignment procedure was undertaken, where the combustion chamber was operated on EtOH, and the nozzle rotated until this obstruction disappeared [49]. This configuration ensured adequate ignition of all the fuel jets. The nozzle was then fixed in position using set screws.

![Figure 3.6: Pilot flame alignment configuration [49].](image)

3.5 Diffuser, Viewport, and Exhaust Sections

The diffuser features a conical shape with an expansion ratio of about 1.7 and a half angle of 35 degrees [49]. With this expansion ratio along with the swirl induced into the system, the flow can be classified as being highly confined [69].

After the diffuser section is the viewport. The main function of the viewport is to provide a visual of the flame. Instabilities and poor ignition characteristics of bio-oil increase the susceptibility of blow out, especially at extreme conditions. Although a blowout can also be tracked using temperatures inside the combustion chamber, the view of the flame serves many other functions. The nozzle is aligned by visually ensuring the conditions discussed in Section 3.4 are satisfied. Another function is to track, visually, the combustion quality. The viewport features two ports, one in which a quartz window (P.N. 1357T36, McMaster-Carr) is attached for viewing, and another port which allows access into the combustion chamber for maintenance purposes.
The last component of the combustion chamber assembly is the exhaust section. The exit of the exhaust section had to replicate the original combustion chamber which forms a seal with the ceramic plate on the engine block. The top of the exhaust section is a flange which adapts to the viewport section. The bottom was designed in such a way that the hot exhaust gases are forced through the annular heat exchangers to transfer heat to the working fluid and the pass into the exhaust outlet as shown in Figure 3.7. As discussed in Section 2.2.6, the hot combustion products have to flow through the hot end heat exchanger slots which then transfer the heat into the working fluid. This section had to both affix to the burner assembly and to the WhisperGen base.

Figure 3.7: Exhaust section of the burner assembly.
Chapter 4

Experimental Methodology

4.1 Control Systems Modification

The control systems rely on the signal/data interaction between the combustion chamber components to operate. Since the old combustion chamber is replaced, these controls had to be modified in order to operate the engine with bio-oil. As discussed in Section 2.2.3, there are five combustion chamber components which interact with the combustion chamber. The modifications to these control systems will be discussed in this section. These include the exhaust temperature sensor, oxygen sensor, glow plug, fuel pump, and flame rod. Details of the function of these components have been discussed in Section 2.2.3.

4.1.1 Exhaust Temperature Sensor

The exhaust temperature sensor features a K-type thermocouple which monitors the temperature of the exhaust exiting the original combustion chamber, before it enters the heat exchanger. It is attached to the combustion chamber with a ¼” NPT fitting. This signal plays a key role in engine operation as it is used by the control systems for several purposes. The value of the exhaust temperature is used to determine when to crank the engine during the heat up stage, when to shut off the glow plug and the amount of air required through the system during
the ignition phase. The reading of the signal is verified before every start to ensure correct operation. If this reading is incorrect, the engine would not pass the pre-check, and hence not start. Since the old combustion chamber was replaced, this signal had to be present as engine operation without it would not be possible. A 0.25 in. NPT port was added in the outlet of the new exhaust section where a new K-type thermocouple (P.N. TC-K-NPT-G-72, Omega) would be placed and its signal sent to the control systems. Preliminary testing determined that the exhaust temperature at the new location was similar to that with the old configuration and would not create any errors in the system.

4.1.2 Oxygen Sensor

The original oxygen sensor of the WhisperGen outputs the fuel-air equivalence ratio to the control systems and is housed above the exhaust heat exchanger. The sensor features two zirconium dioxide (ZrO\(_2\)) discs which a small hermetically sealed chamber in between. One of the ZrO\(_2\) discs acts as a reversible oxygen pump, which is used to fill and empty the sample chamber. The second disc then measures the ratio of the partial pressures and generates a signal which is read by the engine control systems. In order to obtain the required operating temperatures of 700 °C for the ZrO\(_2\) to operate as an oxygen pump, a heating element is used. Before the fuel pump is initiated, air is passed at a high flow rate through the combustion chamber to the outlet. At this point, the control systems verify that the oxygen sensor reading is approximately 13 to 25 % vol. If this reading is outside these limits, the controller will produce a fault and the engine will not start. If the reading falls between these limits, the sensor is recalibrated to 21 % vol. of oxygen, corresponding to the oxygen content in air. During operation, the engine uses this value to set the fuel-air equivalence ratio. If at any time during operation, the signal was outside a certain error, the engine would stop.

The exhaust heat exchanger of the original configuration, which housed the oxygen sensor, would no longer be used because of its narrow passages and low thermal capacity. Therefore, in order to avoid any faults the oxygen sensor signal had to be modified. The signal on this specific oxygen sensor is sinusoidal with voltage and frequency values which correspond to a certain fuel-air equivalence ratio. The signal output was through the engines LabVIEW program using a data acquisition (DAQ) card (P.N. 780116-01, National Instruments). The
specific amplitude and frequency which corresponds to an oxygen content of 21 % vol. in the air was 3.4 V and 1950 Hz respectively.

4.1.3 Glow Plug

The glow plug was connected to the evaporative burner of the old configuration, which would no longer be used, thus the signal had to be overridden. The absence of the glow plug during the start-up phase would lead to faults which would not allow the engine to start. When the start signal is received by the engine’s controller, a sequence is initiated to power the bus so that the isolate relay can be closed. Part of this sequence is a test to ensure that the glow plug is functional. This test requires that the bus voltage, when the glow plug relay is initiated, drop to less than 7 V. In order to override this test without a glow plug, the isolator relay was turned off during the start-up phase, which in turn decreased the bus voltage to less than 7 V. This procedure superseded the error and the engine was able to enter the subsequent steps in the start-up mode. This was implemented in the procedure for testing the engine with the new configuration.

4.1.4 Fuel Pump

The fuel system of the WhisperGen is composed of a 12 V fuel pump (P.N. ELPU30033, Total Power) and lines which connect to the evaporative burner. The pump operates on a PWM signal which is received from the control systems to deliver the required fuel flow rate. This fuel pump is only capable of delivering a maximum flow rate of 19 mL/min of diesel. As discussed later, this maximum flow rate is not sufficient for the required energy input, therefore this fuel pump could not be implemented into the new configuration. Also, the fuel lines connected to the pump are 0.125 in. in diameter, which would not be suitable for bio-oil. Therefore, a new fuel pump would be required which is capable of delivering higher flow rates, and the signal of the old fuel pump would have to be modified. A peristaltic pump (P.N. EW-07550-30, Cole Parmer) is used as the new fuel pump. Peristaltic pumps are highly accurate and can be used with a variety of liquids. Previous work on modifying the fuel system [31] showed that the original fuel pump would have to remain connected to the control systems whereas the fuel lines could be changed to another pump. The original fuel pump was disconnected from the system and
operated in an isolated closed loop setup. The fuel, diesel, is pumped from and to the same reservoir with the connections still present to the control systems.

4.1.5 Flame Rod

The flame rod signal indicates the presence of a flame which the control systems require for sensing ignition and a stable flame during operation. If at any time, during which the fuel pump is on, the signal indicates no flame, the engine would stop. Since the flame rod was attached to evaporative burner, the signal had to be modified. After operating the engine with the old configuration, it was found that the flame rod signal, which varies from 0 to 10 µA, could be linearly related to the exhaust temperature signal as shown in Figure 4.1. The relationship between the exhaust temperature and flame rod current is shown in Equation 4.1, where $I_{FID}$ is the flame rod current, and $T_{Exhaust}$ is the exhaust temperature at the outlet.

This 0 to 10 µA corresponds to a voltage ranging from 0 to -30 $V_{DC}$. Since the DAQ card available was only able to output -10 to 10 $V_{DC}$, an amplification circuit was designed. In order to amplify the signal to the required output, an operational amplifier (op-amp) in a non-inverting configuration was employed. An op-amp is able to amplify, using a power source, the voltage across its input terminals. This value of amplification can be determined by the configuration and attached circuit components, namely resistors, shown in Equation 4.2. In this equation, $V_{IN}$ represents the input voltage and $V_{OUT}$ is the output voltage. The amplification value of 3 was required for the signal. The circuit is shown in Figure 4.2, where a DC power supply (P.N. PS-32LAB, Pyramid) was used as the power source for the amplifier. The input voltage from the DAQ card, via LabVIEW, controlled the output dummy signal to the WhisperGen’s control systems, based on the relationship shown in Equation 4.3. This relationship was programmed into the engine’s LabVIEW program. The exhaust temperature was obtained from the engine’s control systems and the LabVIEW program calculated the input voltage to the op-amp. This sequence of the LabVIEW program was automatically initiated with the start of fuel flow.
\[ I_{FID} = 0.0764T_{Exhaust} - 2 \quad (T_{Fuel\,start} \leq T_{Exhaust} \leq 155) \]
\[ I_{FID} = 10 \quad (T_{E} > 155) \]  
\text{Equation 4.1}

\[ V_{IN} = V_{OUT} \left( 1 + \frac{R_2}{R_1} \right) \]  
\text{Equation 4.2}

\[ V_{IN} = 0.0764T_{Exhaust} - 2 \quad (T_{Fuel\,start} \leq T_{Exhaust} \leq 155) \]
\[ V_{IN} = 10 \quad (T_{Exhaust} > 155) \]  
\text{Equation 4.3}

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{exhaust_vs_fid_signal}
\caption{Exhaust Temperature vs. FID Signal.}
\end{figure}
4.2 Engine Installation

Figure 4.3 shows the experimental setup of the WhisperGen micro-CHP system. The setup includes the new combustion chamber, a Stirling engine, an alternator, an electronics enclosure which houses the control systems, two shell and tube heat exchangers and other components associated with the original WhisperGen and those which have been included to ensure operation with the new modifications. Figure 4.4 shows a schematic of these components and their interaction. It also provides information on the placement of various sensors, diagnostics, and the flow of coolant and exhaust. Descriptions of the temperature nomenclature used in the next chapter with respect to their placement are listed in Table 2.1.

Air is drawn through the system through a stack fan (P.N. VB-003, Spencer Turbine) located at the end of the system. This configuration ensures the system is always under negative pressure of approximately 150 to 350 Pa depending on the operating conditions. The pressure inside the chamber is measured using a manometer (P.N. 3974K2, McMaster-Carr). The maximum air flow rate achievable by the system is about 450 L/min. The stack fan is controlled manually with a variable voltage output transformer (P.N. 6994K17, McMaster-Carr). The air inlet is extended towards the combustion chamber to recover some of the heat which is lost, thus

Figure 4.2: Signal amplification output circuit schematic.
providing a source of preheat for the incoming air. A J-type thermocouple (P.N. 9251T91, Omega) is placed before the air box to measure the temperature of the intake air.

The engine’s original fuel pump was replaced with a peristaltic pump (P.N. EW-07550-30, Cole Parmer), which are highly accurate and can be used with a variety of liquids. The pump’s shaft speed, which varies from 0 to 600 RPM, is linearly proportional to the fuel flow rate. The peristaltic pump has a computer-compatible drive, which is controlled by a DC voltage signal through the relationship in Equation 4.4. $V_{\text{FUEL}}$ stands for the DC voltage output by the DAQ card and $N$ is the required rotation for the peristaltic pump. This relationship was programmed into LabVIEW, and the required pump speed as the input variable. The pump speed was entered beforehand so as to automate the process.

$$V_{\text{FUEL}} = \frac{N_{\text{PUMP}}}{60} \quad \text{Equation 4.4}$$

The nozzle featured two connections, one for the fuel and the other for the atomizing air. The atomizing air was supplied with the building compressed air supply. It was filtered and regulated through a pressure gauge and rotameter before entering the nozzle. A pressure gauge and a J-type thermocouple (P.N. JQ316SS-116G-12, Omega) were attached to the fuel inlet. The pressure gauge indicated the amount of back pressure which existed in the fuel line, and the temperature was used to determine fuel boiling.
The new combustion chamber is placed on top of the Stirling engine and sealed off with a high temperature sealant (P.N. 7356A33, McMaster-Carr). A K-type thermocouple (P.N. KQ316SS-18G-12, Omega) with a 12 in. extension was connected to the exhaust section, and the probe placed near the hot end heat exchangers of the Stirling engine. At the interface between the hot end heat exchangers and the engine block, there was another K-type thermocouple which came with the original engine. A thermistor monitored the temperature of the coolant as it exited the engine block.

The exhaust is then passed through a shell and tube heat exchanger (P.N. SK317HU, Seakamp) which is able to capture the majority of the heat energy. The cooled exhaust temperature is measured with using a J-type thermocouple (P.N. 9251T91, Omega). The heat exchanger is followed immediately by a water trap which holds the water condensate from the exhaust. Before the water enters the heat exchangers, it is passed through a rust and sediment water filter to prevent fouling of the heat exchanger which can inhibit the heat transfer characteristics. After exiting the heat exchanger and the water traps, the exhaust is then put through the stack fan and into the fume hood.
Another heat exchanger is employed to capture the heat from the coolant which is being passed through the engine block. The coolant in this case is pumped through the system at a flow rate of approximately 6 – 11 L/min. J-type thermocouples (P.N. TC-J-NPT-G-72, Omega) are fitted at the inlet and outlet of the heat exchanger’s water connections, and the flow rate is tracked using rotameter (P.N. FL7303, Omega). These temperatures are then used in the post-processing of the data for efficiency calculations. The electricity produced by the alternator was stored in a 216 Ah 12 V deep cycle battery (P.N. SRM-4D, Interstate Batteries). A 1500 W inverter (P.N. 975-0166-01-01, Xantrex) was used to convert the DC electricity into standard AC electricity for use with auxiliary devices.

Samples for emissions measurements are extracted from the exhaust outlet and passed through a condenser where the majority of the water content is condensed. A portion of this sample is diverted to the oxygen sensor before it enters the condenser. The sample is then passed through to the diagnostics area where it is passed through a heated filter and heated lines before it enters the diagnostics at approximately 160 °C.

Table 4.1: Description of temperature nomenclature.

<table>
<thead>
<tr>
<th>Description</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{Heat Exchangers}}$</td>
<td>Above hot end heat exchangers</td>
</tr>
<tr>
<td>$T_{\text{Interface}}$</td>
<td>Interface between ceramic and engine block</td>
</tr>
<tr>
<td>$T_{\text{Coolant}}$</td>
<td>Coolant exiting the engine block</td>
</tr>
<tr>
<td>$T_{\text{Exhaust}}$</td>
<td>Exhaust leaving the engine</td>
</tr>
<tr>
<td>$T_{\text{Cooled Exhaust}}$</td>
<td>Exhaust leaving the heat exchanger</td>
</tr>
</tbody>
</table>
4.3 Energy Balance

To evaluate the performance of the system, a control volume analysis was used to create a simple thermodynamic model. Figure 4.5 shows the inputs and outputs of this system. Fuel and air enter the combustion chamber and release heat which is used by the Stirling engine to produce electricity and heat which is captured by the coolant. The remaining energy is then passed through a heat exchanger which captures more heat. The energy which is not converted into electricity or captured by the heat exchanger is considered lost out of the system as waste heat, either through the exhaust or the combustion chamber.

An energy balance can be applied to the control volume, identifying the inputs and the outputs, as shown in Equation 4.5. The terms on the left hand side represent the energy inputs and those on the right hand side are energy outputs. The heat loss term, $\sum \dot{Q}_L$, is calculated using the energy balance and knowing that the exhaust exiting the heat exchanger is cooled to ambient temperature, thus having a value of zero in the equation. Using the energy balance above, the
energy efficiency of the system can be calculated resulting in Equation 4.6. The energy efficiency is the fraction of the input fuel energy which is used to create both power and heat. This can be further divided into the power and thermal efficiency, where the power efficiency is the fraction of the input fuel energy which is used to create power and the fraction which is used to create heat is the thermal efficiency.

![Thermodynamic model of CHP system.](image)

**Figure 4.5: Thermodynamic model of CHP system.**

\[
m_f h_f + m_a h_a + m_p h_p = W_{\text{net}} + Q_{R1} + Q_{R2} + \sum Q_L + m_w h_w
\]

**Equation 4.5**
The thermal output, via the heat exchangers, was calculated using Equation 4.6, where \( \dot{m}_{water} \) represents the mass flow rate of water, \( c_p \) is the specific heat of water, and \( \Delta T \) represents the temperature difference of the water entering and exiting the heat exchanger.

\[
\eta_{energy} = \frac{W_{net} + \dot{Q}_{R1} + \dot{Q}_{R2}}{\dot{m}_f h_f + \dot{m}_a h_a + \dot{m}_p h_p}
\]

Equation 4.6

\[
\eta_{energy} = \frac{W_{net}}{\dot{m}_f h_f + \dot{m}_a h_a + \dot{m}_p h_p} + \frac{\dot{Q}_{R1} + \dot{Q}_{R2}}{\dot{m}_f h_f + \dot{m}_a h_a + \dot{m}_p h_p}
\]

\[
\eta_{energy} = \eta_{power} + \eta_{thermal}
\]

During operation, significant heat loss was observed since the combustion chamber was not optimized. This resulted in a significant decrease in the power efficiency. In commercial applications, heat loss through the optimized combustion chamber will be negligible as it is with the original combustion chamber of the WhisperGen. In order to evaluate the performance between bio-oil and diesel for an optimized system, a new efficiency calculation is proposed as shown in Equation 4.8. This efficiency, referred to as the modified power efficiency, is the fraction of the energy input, neglecting heat loss through the combustion chamber, which is converted into power. In other words, this is the fraction of the thermal, power and exhaust energy which contributes to the generation of power. Therefore, if heat loss through the combustion chamber was eliminated while the remaining system stayed as is, then this would be the expected power efficiency of the system.

\[
\eta_{power, modified} = \frac{W_{net}}{(\dot{m}_f h_f + \dot{m}_a h_a) - \sum \dot{Q}_L}
\]

Equation 4.8
4.4 Gas Phase Emissions Measurements

4.4.1 Detailed Exhaust Gas Pollutant, and CO₂ Emissions

A Nicolet 380 Fourier Transform Infrared Spectrometer (FTIR) is used to measure CO and NO₃ concentrations in the exhaust. Gas samples are analysed inside the gas cell, which has a path length of 2 m, and a volume of 0.19 L. The basic principle of operation is the comparison of the absorption in the mid infrared region (500 to 4000 cm⁻¹) against known calibrated standards [70]. Each spectrum is an average of 24 successive scans of the gas sample over a one minute time period with a resolution of 1 cm⁻¹. More details regarding the FTIR operation and calibration can be found elsewhere [31].

The calibration standards composed of known species concentrations are used with the partial least squares model to calibrate the instrument. Table 4.2 shows the detection limits and root mean square error (RMSE) for the FTIR measurement species. Detection limits are found by determining the concentration which results in a signal to noise ratio of 4. The RMSE is the error associated between the predicted and actual concentration in a calibration mixture. Two calibration ranges were chosen for CO, due to a wide range which could be expected during testing. The high calibration was primarily used for start-up conditions, and operation with bio-oil. The lower calibration range was used for steady state operation with diesel. During operation, exhaust is continuously drawn into the gas cell using a vacuum pump, at a pressure of 86.3 kPa and a flowrate of 10.3 L/min. The temperature of the sample is kept between 115 to 120 °C by passing it through heated lines. When taking a spectrum, the gas cell is filled about 50 times resulting in average values for the species concentration.

<table>
<thead>
<tr>
<th>Species</th>
<th>Units</th>
<th>Detection Limits</th>
<th>RMSE</th>
</tr>
</thead>
<tbody>
<tr>
<td>CO (low)</td>
<td>PPM</td>
<td>10-250</td>
<td>15.6</td>
</tr>
<tr>
<td>CO (high)</td>
<td>PPM</td>
<td>250-1500</td>
<td>25.5</td>
</tr>
<tr>
<td>CO₂</td>
<td>%</td>
<td>2-15</td>
<td>0.19</td>
</tr>
<tr>
<td>NOₓ</td>
<td>PPM</td>
<td>10-300</td>
<td>6.3</td>
</tr>
<tr>
<td>H₂O</td>
<td>%</td>
<td>0.5-15</td>
<td>0.12</td>
</tr>
</tbody>
</table>
4.4.2 Oxygen Concentration and Equivalence Ratio

A zirconia (ZrO$_2$) oxygen sensor (P.N. OXY6200, Engine Control Systems Ltd.) was used to monitor the oxygen content of the exhaust in units of % vol. The sensor outputs a voltage, ranging from 0 to 5 V$_{DC}$, which is correlated to the oxygen content by the relationship shown in Equation 4.9. In this equation, $V_{OUT}$ represents the output voltage, $V_{CAL}$ is the calibration voltage and % $O_{2,CAL}$ is the calibration oxygen content at which $V_{CAL}$ was specified. The % $O_{2,CAL}$ is assumed to be 21 % vol., representing the oxygen content in air. This relationship was programmed into the engine’s LabVIEW program for monitoring the real time oxygen content. The voltage from the oxygen sensor was input into the DAQ card and used for the calculation. Before each test, the sensor would be operated with air, at a sampling rate of 1.8 L/min, and the calibration voltage term, $V_{CAL}$, adjusted. The sensor employs a high temperature element which could potentially oxidize the exhaust species, and thus was not placed in line with the FTIR exhaust sampling line.

\[ \% O_2 = \left( \frac{V_{O2}}{V_{CAL}} - 1 \right) \left( \% O_{2,CAL} \right) \]  \hspace{1cm} \text{Equation 4.9}

The real time oxygen content during tests was used to calculate the fuel-air equivalence ratio, assuming complete combustion. Since the CO and CO$_2$ values were not continuously monitored, it was difficult to include the accurate values in the calculations. The highest amount of CO obtained during tests was approximately 0.17 % vol. of the total exhaust, implying a very high combustion efficiency, which in turn justifies the back calculation. This calculation was also verified by using the monitored mass flow rates of fuel and air.

4.5 Engine Monitoring Programs

As mentioned in the previous sections, data acquisition and controls manipulation was accomplished through a LabVIEW program along with DAQ cards. Thermocouples were set-up in various locations as shown in Figure 4.4 to monitor the performance of the engine through the LabVIEW program interface shown in Figure 4.6. Temperature measurements of the combustion chamber were tracked to ensure safe operation within material limitations. The oxygen sensor
was also connected to the LabVIEW program in order to track the dynamic fuel-air equivalence ratio through the oxygen content. The primary air flow rate through the combustion chamber was tracked with a flow meter (P.N. 8689 MPNH-SSS-133-DC24-AIR, Eldridge Products). The flow rate was automatically logged by the EPI software.

The WhisperGen comes with its own monitoring software, Micromon, as shown in Figure 4.7. This software is used to track the operation of the engine. The data recorded includes power output, battery voltage, alternator current output, glow plug operation, and stages of operation among the many other parameters. The data from this program was monitored and logged into the LabVIEW program to create a central control program and eliminate the extra windows. This also provides the ability to perform real-time performance calculations during operation. A high rate of sampling was chosen for all data transfer in order to capture details of the engine’s transient behavior.

![LabVIEW controls program](image)

**Figure 4.6: LabVIEW controls program.**
4.6 Operating Procedure

Since significant changes were made to the combustion chamber, its control systems and experimental plan was revised in order to adequately optimize the performance of the engine with bio-oil. Diesel was used as the baseline for comparison, as this was the engine’s original fuel. All parameters will be monitored for performance evaluation, and the quality of combustion will be evaluated based on emissions measurements.

The engine was operated in the maintenance mode, which overrides the battery management function and produces the maximum amount of power. The heat management mode could not be chosen due to settings of the control parameters, such as a set coolant temperature. The inability to satisfy these control parameters resulted in an error. Tests were commenced with a battery charge of 50% to ensure that the control systems do not initiate the clamp element if the battery charge increases to 80% or above during charging. As mentioned before, the activation of the clamp element diverts the power produced by the engine to the resistive element instead of the battery. This would result in an inaccurate energy balance as the power output from the engine does not take into account the power provided to the engine components. If the clamp element was activated during the tests, a 500 W load was applied using a portable work light, leading to its deactivation. If the engine control systems detected a fault, the engine goes into
shut down mode. Care and attention had to be given at all time to ensure correct operation. To avoid any faults during testing, several different methods of running the engine were investigated.

Before discussing the operational procedure, one key issue to address was the energy input into the system. The original configuration required a fuel energy input of approximately 9.45 kW at steady state operation. This input would not be sufficient for the new configuration. Since the total length of the combustion chamber was approximately 0.5 m, a large amount of heat loss was expected. In order to obtain adequate operating temperatures inside the combustion chamber, more specifically near the hot end heat exchangers, a variety of energy inputs were investigated. Extreme test conditions confirmed that the maximum safe operating energy input would be approximately 15 kW on a LHV basis for an extended period in order to evaluate the performance of the engine. During operation with higher energy contents, the stainless steel of the burner turned red hot, proving not safe. Insulation was applied to the outer surface of the combustion chamber in the hopes that a lower energy input would result in the same temperature, but this again resulted in the combustion chamber turning red. Initial testing on the combustion chamber was done with EtOH because of its high volatility and no residue.

Operation with diesel was performed in the following manner. The start signal was sent to the control systems of the engine which prompted the LabVIEW program to start the peristaltic pump, thus initiating the flow of fuel. For the first 5 minutes, the input into the combustion chamber was 10 kW. This would later be increased to 15 kW and would stay consistent throughout the remaining operation time. Since the combustion chamber started from an ambient temperature, ignition was initially difficult even with the presence of a pilot flame. It was found that during first minute of starting, some of the fuel would ignite while the rest would spray onto the combustion chamber walls and the hot end heat exchangers of the engine. This was due to temperatures being below those necessary for auto-ignition. However, as temperatures increased, the fuel which was deposited on the surfaces would vaporize and auto-ignite leading to large acoustic fluctuations in the combustion chamber, otherwise known as combustion instabilities.

During the first five minutes, the engine’s control systems go through the preheat sequence where the glow plug is operated to heat the combustion chamber. At the five minute
mark, the control systems would initiate the original fuel pump. The engine is cranked at the seven minute mark, after two minutes of flame stabilization. The engine reaches steady state after approximately 40 minutes of operation. The engine is allowed to continue running for five more minutes after which a stop signal is sent to the control systems, which stops the peristaltic pump. The air flow rate is then decreased for about 30 seconds, so as to permit the working fluid to capture the heat inside the combustion chamber. The air flow rate is then switched to the maximum in order to cool the system and capture the remaining heat through the heat exchangers.

Blending bio-oil with EtOH has many benefits, including increased stability and volatility, and a decrease in viscosity [37]. A variety of bio-oil/EtOH blends were tested for operation. It was found that an 80 % vol. blend of bio-oil (with 20 % vol. EtOH) would be the maximum amount which would ensure adequate operation and combustion. The term “bio-oil/EtOH blend” from this point forwards implies an 80 % vol. bio-oil and 20 % vol. EtOH blend.

Operation with the bio-oil/EtOH blend was different from diesel because of bio-oil’s poor ignition characteristics. The combustion chamber was initially fuelled by EtOH, 10 kW energy input for the first five minutes, and 15 kW after. At the nine minute mark, the fuel reservoir was switched to the bio-oil/EtOH blend. Due to the distance between the peristaltic pump and the nozzle, it takes approximately one minute for the bio-oil/EtOH blend to reach the nozzle. When the bio-oil/EtOH blend is observed to be entering the nozzle, the peristaltic pump speed is increased. The bio-oil/EtOH blend fuelled the engine until the 45 minute mark, at which point a stop signal was sent to the control systems causing the peristaltic pump to stop. The combustion chamber and engine were then cooled. After the official testing and data acquisition ended, the combustion chamber was immediately operated with EtOH to clean the lines and burn off any deposits which had accumulated. Manual cleaning of the combustion chamber was also performed after every test to ensure accurate emissions measurements.

The fuel flow rates were calibrated before the tests and verified after the completion of the test. The fuel flow rate error of the peristaltic pump in the operating range for tests was about ± 0.25%. The difference in the fuel flow rates before and after tests was observed to be 0.1
mL/min for diesel and 0.2 mL/min for the bio-oil/EtOH blend. This corresponded to an error of approximately 0.4% for both diesel and the bio-oil/EtOH blend.

The oxygen sensor was monitored continuously throughout testing, and the air flow rate was adjusted to keep the operating point consistent. Emissions measurements were performed every minute to capture the transient trends which exist from start-up to steady state.
Chapter 5

Results and Discussion

5.1 Fuel Analysis

The properties of the bio-oil and diesel fuel used in this study are specified in Table 5.1. A more complete fuel specification for diesel is provided in Appendix B. The nitrogen content in bio-oil will contribute to higher NO\textsubscript{x} emissions than diesel which has no nitrogen content. This is due to the conversion of fuel bound nitrogen to NO\textsubscript{x}, where with diesel NO\textsubscript{x} formation will occur via the thermal mechanism. The presence of solids in bio-oil poses concern for clogging the nozzle passages, whereas with diesel this is not an issue due to the lack of solids. A higher ash content in bio-oil indicates a higher particulate loading in the exhaust than diesel [37]. A significantly higher water content in bio-oil signifies a major sensitivity to parameters which affect ignition quality, such as the pilot flame and primary air preheat [37]. The discrepancy between the heating values imples a higher mass flow rate for bio-oil, even with the addition of EtOH, in order to obtain the same energy throughput. Viscosity is significantly higher for bio-oil than for diesel. This along with the higher mass flow rates play an important role in determining atomization quality. However, given that the temperature of the fuel exiting the nozzle is about 80 °C, a significant decrease in the viscosity is noticed. The viscosity of diesel does not show a strong dependence on temperature and thus it can be assumed that the fuels exiting the nozzle have a similar viscosity [37].
Table 5.1: Comparison of fuel properties between bio-oil and diesel.

<table>
<thead>
<tr>
<th>Properties</th>
<th>Units</th>
<th>Bio-oil</th>
<th>Diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>C-H-O-N</td>
<td>wt. % (dry)</td>
<td>40.8-7.67-51.5-0.1</td>
<td>86-14-0-0</td>
</tr>
<tr>
<td>Solids</td>
<td>wt. %</td>
<td>0.06</td>
<td>0.000</td>
</tr>
<tr>
<td>Ash</td>
<td>wt. %</td>
<td>0.23</td>
<td>0.000</td>
</tr>
<tr>
<td>Water</td>
<td>wt. %</td>
<td>26.3</td>
<td>0.017</td>
</tr>
<tr>
<td>HHV</td>
<td>MJ/kg</td>
<td>16.8</td>
<td>45.6</td>
</tr>
<tr>
<td>LHV</td>
<td>MJ/kg</td>
<td>15.3</td>
<td>42.8</td>
</tr>
<tr>
<td>Density</td>
<td>kg/m³</td>
<td>1130</td>
<td>828.3</td>
</tr>
<tr>
<td>Viscosity @ 40 °C</td>
<td>cSt</td>
<td>40.09</td>
<td>1.93</td>
</tr>
<tr>
<td>Viscosity @ 80 °C</td>
<td>cSt</td>
<td>4.069*</td>
<td>-</td>
</tr>
</tbody>
</table>

*Viscosity of Bio-oil/EtOH blend

![Figure 5.1: TG curves for bio-oil/EtOH blend and diesel.](image)

The thermo-gravimetric (TG) behavior of the bio-oil/EtOH blend and diesel is shown in Figure 5.1. Diesel shows a smooth curve, which implies a continuous evaporation process [37]. Diesel is fully distillable by 200 °C with a residue of 0.02 wt. %, thus it can be assumed to be in
vapor phase above this temperature. On the other hand, bio-oil/EtOH blend results in a residue of about 40 wt. %, even with temperatures as high as 600 °C. The curve for the bio-oil/EtOH blend is lower than diesel up to 100 °C, thus showing bio-oil’s reduced low temperature volatility [37]. This low temperature volatility can assist in flame ignition and stabilization with the rapid evaporation of the lower molecular weight compounds present in bio-oil [37]. However, if high fuel temperatures are experienced, this can lead to flash induced instabilities caused by fuel boiling [37]. The lower temperature volatiles in diesel have a higher heating value, thus higher localized heat release rates, and likely play a role in better ignition and stability [37].

5.2 Combustion Chamber Operating Parameters

The average steady state operating parameters of the combustion chamber are shown in Table 5.2. These parameters were consistent for the multiple tests performed in order to evaluate reproducibility. Due to the lower calorific value of the bio-oil blend, the fuel flow rate is approximately twice as much as diesel. Both fuels were operated at a swirl number of 5.41. The fuel-air equivalence ratio was significantly higher than planned. The primary air and atomizing air flow rates could not be increased further due to the onset of large acoustic fluctuation in the combustion chamber which led to an unstable flame. These instabilities also led to uneven heat transfer rates which effect the operation of the engine in a negative manner. Operation of the combustion chamber with the presence of instabilities can also lead to flame blowouts. The atomizing air flow rate is related to the droplet size. Higher droplet sizes require a longer burnout time or longer flame residence times, which is not available due to the confined combustion chamber geometry. The flame residence time for this configuration is approximately 2 to 4 ms [37]. Low atomizing air flow rate also results in reduced turbulence levels which decrease the mixing rates thus increasing CO and UHC emissions [37]. Keeping these parameters consistent, the results obtained in the following sections were reproducible within a range of approximately 5 %. The slight inconsistencies are primarily due to variations in thermal output caused by the fluctuations in the building’s water flow rate and temperature.
Table 5.2: Combustion chamber operating conditions for each fuel.

<table>
<thead>
<tr>
<th></th>
<th>Units</th>
<th>Bio-oil/EtOH blend</th>
<th>Diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy Input at Steady State (LHV)</td>
<td>kW</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>Energy Input at Steady State (HHV)</td>
<td>kW</td>
<td>16.6</td>
<td>16.1</td>
</tr>
<tr>
<td>Fuel Flow at Steady State</td>
<td>mL/min</td>
<td>51.1</td>
<td>25.4</td>
</tr>
<tr>
<td>Swirl Number</td>
<td></td>
<td>5.41</td>
<td>5.41</td>
</tr>
<tr>
<td>Primary Air Flow</td>
<td>SLPM</td>
<td>348</td>
<td>337</td>
</tr>
<tr>
<td>Intake Air Temperature</td>
<td>°C</td>
<td>110</td>
<td>105</td>
</tr>
<tr>
<td>Exhaust Oxygen Content</td>
<td>%</td>
<td>4.6</td>
<td>4.6</td>
</tr>
<tr>
<td>Equivalence Ratio</td>
<td></td>
<td>0.75</td>
<td>0.75</td>
</tr>
<tr>
<td>Atomizing Air Flow</td>
<td>SLPM</td>
<td>14.2</td>
<td>15.3</td>
</tr>
</tbody>
</table>

5.3 Energy Efficiencies

Although the energy input into the system was consistent for both diesel and the bio-oil/EtOH blend, there was a significant difference in the steady state averaged temperatures as shown in Table 5.3. The difference in temperatures between the bio-oil/EtOH blend and diesel is primarily due to the magnitude of heat lost through the combustion chamber walls. Heat loss through the combustion chamber walls occurs through convection and radiation. Surface temperatures were similar for both the bio-oil/EtOH blend and diesel, which is indicative of the combustion chamber’s thermal inertia. In addition, near flame temperatures are also similar for both fuels signifying comparable heat transfer due to convection. But radiation heat transfer is a primary component of the thermal energy released by diesel. This radiation heat transfer stems from both the hot gasses and the soot particles within the diffusion flame [71]. The bio-oil/EtOH blend flame did not exhibit the same luminosity as the diesel flame, thus making convection the primary source of heat transfer. Based on the energy balance, the majority of losses, shown in Figure 5.2, were through the combustion chamber walls as the exhaust is cooled to near ambient temperatures. Higher fuel flow rates and lower atomizing air for the bio-oil/EtOH blend increased the flame length which is responsible for the slightly higher temperatures at the hot end heat exchangers.
Table 5.3: Steady state operating temperatures for the combustion chamber and engine.

<table>
<thead>
<tr>
<th>Units</th>
<th>Bio-oil</th>
<th>Diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{Heat Exchangers}}$ °C</td>
<td>739</td>
<td>709</td>
</tr>
<tr>
<td>$T_{\text{Interface}}$ °C</td>
<td>423</td>
<td>386</td>
</tr>
<tr>
<td>$T_{\text{Coolant}}$ °C</td>
<td>38.4</td>
<td>37</td>
</tr>
<tr>
<td>$T_{\text{Exhaust}}$ °C</td>
<td>462</td>
<td>425</td>
</tr>
<tr>
<td>$T_{\text{Cooled Exhaust}}$ °C</td>
<td>33.9</td>
<td>29</td>
</tr>
<tr>
<td>$T_{\text{Surface}}$ °C</td>
<td>407</td>
<td>414</td>
</tr>
</tbody>
</table>

Figure 5.2: Energy loss during testing for diesel and bio-oil/EtOH blend.

Average steady state performance of the engine with the bio-oil/EtOH blend and diesel is outlined in Table 5.6. Power output for the bio-oil/EtOH blend is significantly higher than that of diesel. Recall Equation 2.1, which states that the work output is proportional to the difference in the hot and cold end. The higher work output for the bio-oil/EtOH blend is due to the fact that the temperature differential between the hot and cold end temperatures is greater than diesel. The amount of heat captured through the coolant was greater for the bio-oil/EtOH blend than diesel.
due to the higher hot end heat exchanger temperature. The thermal output by heat exchangers can vary based on fluctuations in the building water supply flowrate and temperature.

The thermal output from the exhaust is significantly higher for the bio-oil/EtOH blend, even though the exhaust is cooled to relatively similar temperatures, and below the water saturation temperature of engine exhaust which in this case is was calculated to be about 40 to 50 °C. This larger extraction of energy is due to the difference in the amount of water present in the exhaust. Bio-oil’s inherent water content, and diesel’s lack of, allows for a greater amount of latent heat to be captured during the cooling of the exhaust to room temperature [37].

The LHV was used to calculate the power efficiencies due to the presence of water in the vapor form before and after the hot end heat exchangers. Thermal efficiencies were calculated based on the HHV due to the presence of water in the liquid phase. Figures 5.3 to 5.5 show the power, thermal and modified power efficiencies for diesel and the bio-oil/EtOH blend, respectively. These figures illustrate the transient nature of the efficiencies. These calculations do not include the power consumption of ancillary components of the engine.

Steady state operation with diesel resulted in power and thermal efficiencies of 2.5% and 61.9%, respectively. The corresponding steady state efficiencies for the bio-oil/EtOH blend were 3.4% and 72.4%, respectively. Lower efficiencies for operation with diesel are attributed to the higher heat losses incurred through the combustion chamber, and the lower latent heat transfer in the exhaust heat exchanger. The low power efficiencies for both fuels are primarily due to the choice in working fluid and its pressure. As discussed in Section 2.1.2, the choice of working fluid and its pressure can be very important in determining the power output of the engine. Nitrogen’s low thermal conductivity and high viscosity makes it inferior to fluids such as hydrogen or helium in order to obtain a high power density and efficiency [17]. A high working fluid pressure, in the range of 20-25 MPa is required for high power efficiencies, whereas the working fluid in the WhisperGen is pressurized to 2.8 MPa [11]. The modified efficiencies are slightly higher than the power efficiencies due to the exclusion of the heat loss through the combustion chamber walls. From Figure 5.5, it is observed that the modified efficiencies for both diesel and the bio-oil/EtOH blend are comparable. This shows that if the combustion chamber were to be optimized, such that heat loss was negligible, the performance of the engine with
diesel and bio-oil is comparable. More performance data for diesel and the bio-oil/EtOH blend can be found in Appendix C and D, respectively.

One issue to note is the high amount of variability which is seen in the power output during operation with diesel as shown in Figure C.3. This was due the large variability of the temperatures caused by fluctuating heat release rates. This led to varying heat transfer rates into the working fluid. The variability in the region of fluctuating power output, when calculated statistically is 3 °C, whereas with the bio-oil/EtOH blend it is 0.4 °C.

Table 5.4: Steady state performance of the engine.

<table>
<thead>
<tr>
<th></th>
<th>Units</th>
<th>Bio-oil</th>
<th>Diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power Output</td>
<td>W</td>
<td>534</td>
<td>396</td>
</tr>
<tr>
<td>Thermal Output (Coolant)</td>
<td>W</td>
<td>5693</td>
<td>5211</td>
</tr>
<tr>
<td>Thermal Output (Exhaust)</td>
<td>W</td>
<td>6655</td>
<td>5032</td>
</tr>
<tr>
<td>LHV Power Efficiency</td>
<td>%</td>
<td>3.4</td>
<td>2.5</td>
</tr>
<tr>
<td>HHV Thermal Efficiency</td>
<td>%</td>
<td>72.4</td>
<td>61.9</td>
</tr>
<tr>
<td>Modified Efficiency</td>
<td>%</td>
<td>4.1</td>
<td>3.7</td>
</tr>
</tbody>
</table>

Figure 5.3: LHV power efficiencies.
Figure 5.4: HHV thermal efficiencies.

Figure 5.5: Modified power efficiencies.
5.4 Exhaust Species Emissions

Emissions data was taken every minute during the operation of the peristaltic pump. Figure 5.6 shows the various exhaust species concentrations for operation with diesel. As mentioned previously, testing is initiated with all components at ambient temperature. Initiating the fuel flow, at ambient conditions, results in a significant amount of fuel which is not ignited and sprayed instead to the walls of the combustion chamber and on the hot end heat exchangers. Hence high CO levels are expected during this stage as the fuel partially burns. As temperatures begin to rise, the ignition improves and the CO levels drop to a steady state value of about 158 ppm. The rise in temperature also brings about the formation of NO\textsubscript{x} to a steady state value of 42 ppm.

![Graph showing CO and NO\textsubscript{x} emissions over time](image)

**Figure 5.6: Diesel exhaust species emissions.**

Emissions for the bio-oil/EtOH blend are shown in Figure 5.7. Again, cold start with EtOH proceeded in a similar manner to diesel with some of the fuel spraying on to the combustion chamber walls and the hot end heat exchangers. A sudden spike is seen in both CO and NO\textsubscript{x} with the introduction of the Bio-oil/EtOH blend at the 10 minute mark. During steady state operation, the CO emissions were approximately 1550 ppm, and the NO\textsubscript{x} emissions were
around 115 ppm. The FTIR’s detection limit for CO is 1500 ppm, and so the values which lay outside this limit were extrapolated.

Table 5.5 summarizes the steady state emissions of diesel and the bio-oil/EtOH blend. High CO emissions are attributed to the operating conditions and design of the combustion chamber. Operating conditions include the fuel-air equivalence ratio and atomizing air. The primary air is a key source of the turbulence levels created in order to obtain thorough mixing of the fuel and air [37]. Primary air also provides the oxygen for oxidation of CO to CO\(_2\) for more complete combustion. In this study, the operation was with a lower primary air flow rate or a lower fuel-air equivalence ratio than planned, due to the onset of combustion instabilities, which led to the increase in CO emissions. Lower atomizing air flow rates lead to larger droplet sizes which require more time for complete burnout. This also leads to less turbulence which decreases the mixing quality. However, both these factors contribute to the decrease in NO\(_x\) formation, which is again due to the low mixing quality and the presence of localized regions in the flame which are deprived of oxygen [37]. A benefit to low atomizing air is that the jets become longer, which in this case implies higher temperatures in the vicinity of the hot end heat exchangers. The absence of air preheating to higher temperatures was another source of high CO emissions. This increases the flame temperature and allows for improved oxidation of the fuel and a reduction in reaction quenching [37]. In small scale combustion applications (10-15 kW) of bio-oil, CO emissions are generally high. Smaller combustion chambers lead to smaller hot zone residence times which increase CO. A lack of insulation in the combustion chamber decreases temperatures which lead to less effective CO oxidation. Because of diesel’s high volatility, operation with lower primary air and lower atomizing air did not pose a problem in terms of CO and NO\(_x\) emissions as they were relatively low.

The higher NO\(_x\) formation in the bio-oil/EtOH blend is primarily due the inherent nitrogen content which exists in the fuel, the mechanism known as fuel bound NO\(_x\) formation. Although temperatures inside the combustion chamber continue to increase after switching the fuel from EtOH to the bio-oil/EtOH blend, NO\(_x\) emissions remains relatively consistent. This shows the independence of NO\(_x\) formation with respect to temperature. Since flame temperatures are higher for hydrocarbon fuels than bio-oil [72], and the lack of nitrogen in the fuel, diesel had significantly lower NO\(_x\) emissions, which are attributed to the thermal NO\(_x\) mechanism.
Figure 5.7: Bio-oil/EtOH blend emissions (after 10 minutes).

Table 5.5: Steady state CO and NOx emissions.

<table>
<thead>
<tr>
<th>Units</th>
<th>Bio-oil</th>
<th>Diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>CO ppm</td>
<td>1550</td>
<td>158</td>
</tr>
<tr>
<td>NOx ppm</td>
<td>118</td>
<td>44</td>
</tr>
</tbody>
</table>

5.5 Old and New System Comparison

Emissions and efficiency testing on the original engine was performed in a separate study [31], as reported in Section 2.2.7. Steady state performance and emissions for the old and new system, both fuelled by diesel, are shown in Table 5.6. The results for the old configuration were obtained by operating the engine with the heat management mode whereas this study employed the maintenance mode.

The fuel energy input was considerably lower in the old system. The extra fuel energy input in the new configuration was due the heat loss incurred through the combustion chamber.
walls. Power output for the new system was significantly lower with the new configuration due to several reasons. In the old system, the flame rests very close to the hot end heat exchangers. With the inclusion of preheated air and the increased radiation heat transfer, temperatures in that vicinity are significantly higher. Another reason for the decreased power output is the sealing between the ceramic plate and the combustion chamber. The old combustion chamber was designed to adapt onto the ceramic plate, between which was a rope seal. Although care was given towards the sealing of the new combustion chamber with the ceramic plate, it was not as good. This seal forces the exhaust into the hot end heat exchangers where the heat would be transferred into the working fluid, whereas there would be a decrease in the amount of exhaust energy which would be used in producing power. However, as mentioned before, more energy input results in more power output. A test was performed with a 22 kW fuel energy input which resulted in a steady state power output of approximately 850 W. Temperatures at the hot end heat exchanger were approximately 850 °C.

As expected, power efficiencies were significantly higher due to the higher power output and negligible heat loss. The cooled exhaust of the original configuration was at a temperature of approximately 66 °C, which implies that water was present in the vapour phase. Therefore, its efficiencies were calculated based on the LHV. One the other hand, due to the lower temperature of the cooled exhaust with the new configuration, the efficiency was calculated based on the HHV. The proportion of the fuel energy which was converted to thermal energy in the new system was less than that with the new system due to the heat losses. The modified power efficiency of the new configuration, which neglects the heat loss through the combustion chamber walls, was significantly lower than the power efficiency of the old configuration. This is attributed to the lower power output with the new configuration.

Table 5.7 summarizes the temperatures of the old and new configuration. The temperatures at the hot end heat exchanger are estimated to be greater than 1400 °C [31] due to the fuel lean premixed conditions and the inclusion of preheated air. These high temperatures lead to an increased interface and exhaust temperatures. The discrepancy in the interface and exhaust temperatures between the two configurations is not significant. This is due to the amount of heat which is absorbed into the working fluid. For a given temperature at the heat exchangers, the exhaust temperature dictates the amount of heat which is captured by the working fluid. If the
exhaust temperature is high, then less heat is transferred to the working fluid. If the exhaust temperature is lower, this implies that more heat is transferred to the working fluid. Considering the large difference in the hot end heat exchanger temperatures, the difference between the interface and exhaust temperatures in the old and new system are not the same. This implies that more heat was transferred to the working fluid with the old configuration. Since the engine was operating under the heat management mode, the coolant temperatures were higher due to the set point required in that mode of operation. Higher coolant temperatures were achievable due to higher heat absorbance from both the engine block and the exhaust heat exchanger.

Table 5.6: Comparison of engine performance parameters with old and new system [31].

<table>
<thead>
<tr>
<th></th>
<th>Units</th>
<th>Old</th>
<th>New</th>
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<tbody>
<tr>
<td>Energy Input LHV</td>
<td>kW</td>
<td>9.45</td>
<td>15</td>
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<tr>
<td>Power Output</td>
<td>W</td>
<td>1141</td>
<td>396</td>
</tr>
<tr>
<td>Thermal Output</td>
<td>W</td>
<td>6906</td>
<td>10243</td>
</tr>
<tr>
<td>Power Efficiency</td>
<td>%</td>
<td>11.3</td>
<td>2.5</td>
</tr>
<tr>
<td>Thermal Efficiency</td>
<td>%</td>
<td>73.3</td>
<td>61.9</td>
</tr>
</tbody>
</table>

Table 5.7: Comparison of temperatures with old and new system [31].

<table>
<thead>
<tr>
<th></th>
<th>Units</th>
<th>Old</th>
<th>New</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{Heat Exchangers}}$</td>
<td>°C</td>
<td>&gt;1400*</td>
<td>709</td>
</tr>
<tr>
<td>$T_{\text{Interface}}$</td>
<td>°C</td>
<td>431</td>
<td>386</td>
</tr>
<tr>
<td>$T_{\text{Coolant}}$</td>
<td>°C</td>
<td>66</td>
<td>37</td>
</tr>
<tr>
<td>$T_{\text{Exhaust}}$</td>
<td>°C</td>
<td>480</td>
<td>425</td>
</tr>
<tr>
<td>$T_{\text{Cooled Exhaust}}$</td>
<td>°C</td>
<td>72</td>
<td>29</td>
</tr>
</tbody>
</table>

*Estimated

Table 5.8 summarizes the steady state emissions for the old and new configuration. Due to the premixed mode of combustion and higher temperatures in the old configuration, CO emissions were lower. This allowed for better mixing of the fuel and air, and higher oxidation rates resulting in more complete combustion. For the new configuration, the reasons for a relatively high CO have been discussed. Higher NOx emissions for the old configuration are attributed to the difference in combustion chamber temperatures. Temperatures in the old
Combustion chamber are significantly higher than those in the new one due to the lower of heat loss and the inclusion of preheat.

**Table 5.8: Comparison of emissions with old and new system [31].**

<table>
<thead>
<tr>
<th></th>
<th>Units</th>
<th>Old</th>
<th>New</th>
</tr>
</thead>
<tbody>
<tr>
<td>CO Emissions</td>
<td>ppm</td>
<td>95</td>
<td>143*</td>
</tr>
<tr>
<td>NO\textsubscript{x} Emissions</td>
<td>ppm</td>
<td>80</td>
<td>40*</td>
</tr>
</tbody>
</table>

*Values adjusted to wet basis

### 5.6 Endurance Test

Testing with the bio-oil/EtOH blend was only done for a small time period of 45 minutes in order to evaluate the steady state performance. For commercial applications, engines need to operate for thousands of hours before any maintenance is required. For this reason, an endurance test was performed to determine the length of operation with the bio-oil/EtOH blend. The test was started similarly to the other tests with EtOH, 10 kW for the first five minutes, and 15 kW up to the 13 minute mark when the bio-oil/EtOH blend was introduced.

Figure 5.8 shows the power output for the engine, which peaks at approximately 700 W, and then starts to decrease. Noticeable signs of heat exchanger clogging occur during the 47 minute mark when the fan power is increased in order to maintain the fuel-air equivalence ratio constant. During operation with diesel, as the fan air speed is increased, the air flow rate reaches steady values. Increases in the fan speed with the operation of bio-oil result in a sharp increase in flow rate after which it steadily decreases as shown in Figure 5.9, which shows signs of clogging. Each discontinuity in the primary air flow rate represents a manual increase in the air flow rate. The power does start to increase at the 67 minute mark for a few minutes due to the rapid increase in the air flow rate followed by a decrease due to the reduction in primary air flow rate. The decrease in power output is due to the lack of air, which transfers heat via convection, to the hot end heat exchangers and then to the working fluid. At the 85 minute mark, the fan power input is 100%, but the flow rate continues to decrease along with the power output until approximately the 97 minute mark, when the engine is not able to produce any more power. No
stop signal was sent to the engine until power output was very close to zero, and it continued in normal operation up to the 97 minute mark.

Figure 5.8: Power output for Bio-oil/EtOH blend endurance test.

Figure 5.9: Primary air flow rate for Bio-oil/EtOH blend endurance test.
The size of the slots which surround the hot end heat exchangers are approximately 3 mm by 7 mm. These small passages are susceptible to clogging with non-distillable fuels such as bio-oil. Due to poor atomization, large heavy droplets are able to escape the main combustion region and deposit on the walls of the combustion chamber, the hot end heat exchanger or the annular slots. These heavier fuel drops can visibly be seen dropping to these surfaces during testing. The annular heat exchanger slots are the only avenue for the exhaust to exit the main combustion chamber to the outlet. Once these passages begin to clog, the amount of air flow for a given fan speed decreases. Keeping the fuel flow rate consistent, this increases the fuel-air equivalence ratio. As the fuel-air equivalence ratio increase, the rate at which these deposits accumulate also increases, thus accelerating the clogging process. The implications of heat exchanger fouling were discussed in Section 2.1.5. They include the reduction of power and thermal efficiencies, and an increase in emissions due to the increase in the fuel-air equivalence ratio. CO emissions were approximately 1550 ppm, as in the previous case, but started to increase after the 70 minute mark.

Figure 5.10 shows the hot end heat exchangers before and after the long test. It is observed that the majority of the passages are blocked with fuel residue and ash. Ash is seen more clearly in Figure 5.11 as the white spots where the flame was present. This is a before and after picture of the view port section from the inside. As noticed, after the test there is a significant amount of fuel residue and the ash. Ash formation at those locations is due to the burnout of fuel with the presence of the flame. The presence of ash on the hot end heat exchangers is from two sources. One of the sources is from the ash which is formed near the diffuser and viewport section, and is sent downstream. The second source is from burning droplets which ignite the fuel residue on top of the hot end heat exchangers. More data on the tests in presented in Appendix E.
Figure 5.10: Hot end heat exchanger before (left) and after (right).

Figure 5.11: Deposits inside the combustion chamber, before (left) and after (right).
5.7 Combustion Stability

As mentioned in previous discussions, combustion instability was an underlying issue which had a significant effect on the operating conditions, primarily the primary and atomizing air flow rates. Combustion instabilities are caused by unsteady heat release causing acoustic fluctuations which can have a detrimental effect on the operating parameters and thus the combustion quality [64]. Combustion instabilities exist primarily in premixed combustion applications, but they can also occur in non-premixed combustion applications. Lean operation in both applications results in increased sensitivity of the flame to these oscillations. Combustion instabilities are a very complicated phenomenon which involves the interaction between many parameters. Therefore, reasons for its occurrence in this combustion chamber can only be speculated.

There are two primary reasons which can lead to these instabilities in non-premixed combustion applications. One is the formation of vortices within the combustion chamber which capture pockets of fuel-air mixtures and ignite in another region of the combustion chamber leading to acoustic fluctuations [73-75]. Swirling flows alter the flow and combustion dynamics in a combustion chamber which can also lead to these instabilities [76]. In addition, the hot end heat exchangers through which the exhaust must flow in order to exit the combustion chamber introduced flow constrictions at the exit, which can have a significant effect upstream of swirling flows [77-78]. Another reason is the oscillations in the fuel-air equivalence ratio which are due to the acoustic fluctuations in the air and fuel lines leading to variable heat release [74, 79]. These oscillations are self-sustaining and in a closed feedback loop as shown in Figure 5.12 [64].

![Figure 5.12: Coupling between combustion and acoustic processes [64].](image-url)
The initiation of combustion instabilities in this study was observed to occur for several reasons. As mentioned earlier, during cold start, some of the fuel fails to ignite and instead deposits on the combustion chamber walls and hot end heat exchangers. When temperatures begin to increase, this fuel ignites, thus creating multiple flame fronts which can initiate these instabilities. Another means for these instabilities to occur is a mismatch between the atomizing air and the primary air. The mismatch between the swirling flow and the atomizing air leads to the variation of the CRZ, and creation of localized regions where the flow is sluggish and disorganized [64]. The acoustic fluctuations, which are a result of the unsteady heat release, travel upstream to the nozzle, as evident by the fluctuation in the fuel pressure. This leads to an unsteady variable fuel flow rates and thus a fluctuating release of heat which feeds back into the loop.

Figure 5.13 shows the fluctuation in the oxygen sensor values during a period of combustion instability. From this data, the frequency of oscillation was approximately 0.4 Hz, and the amplitude of pressure was approximately 23 Pa determined from the manometer. In gas turbine engines, frequencies and amplitudes are orders of magnitude higher due to high pressure operation. However, for naturally aspirated combustion chambers, even the slightest acoustic fluctuations have a significant effect. The effects of the instabilities on the emissions can be seen in Figure 5.14, with the spikes in CO emissions. When the pressure is greatest during the acoustic fluctuations, the primary air flow rate is also restricted, and so the majority of the fuel partially ignites. This partial ignition of the fuel is due to the variation in the pilot flame length. The acoustic oscillations also affect the flow rates of the methane and oxygen in the pilot torch thus reducing the length of the flame. The issue of combustion instability in this study was resolved by decreasing the atomizing and intake air to values which complement each other to create a stable flame. It is postulated that the combination of atomizing air and intake air introduces flow instabilities which lead to these combustion instabilities. A large amount of time was spent in improving the operating parameters but it was not possible.

It is worth noting that combustion instability is also an issue with the old combustion chamber of the WhisperGen at certain fuel-air equivalence ratios. During these instabilities, a howling sound would emanate from the exhaust pipe. The manufacturer recommends that the system be immediately turned off and the mixture setting changed.
Figure 5.13: Oxygen sensor response during combustion instability.

Figure 5.14: CO concentration during combustion instability.
Chapter 6

Conclusions and Recommendations

6.1 Conclusions

The WhisperGen DC micro-CHP, a commercial Stirling engine fuelled by diesel, was modified to accommodate fast pyrolysis liquid, or bio-oil, as a fuel. The original system featured an evaporative burner which would not be suitable with nondistillable fuels such as bio-oil. For this reason, a pilot stabilized swirl combustion chamber was designed to replace the original burner. The control systems which were crucial to the operation of the engine with the old combustion chamber were also modified. Dummy signals were sent to the control systems to ensure operation with the new combustion chamber. The control manipulation and data acquisition were accomplished through a LabVIEW program. Extensive testing was performed with the new engine configuration with ethanol (EtOH) to ensure correct operation. A comparative efficiency and emissions study was performed with diesel and an 80 % vol. bio-oil and 20 % vol. EtOH blend (bio-oil/EtOH blend). The engine was operated for 45 minutes at which point it reached steady state operation. An endurance test was also performed to determine the length of operation with the bio-oil/EtOH blend.

The engine was operated in the maintenance mode, which overrides the battery management function and produces the maximum amount of power. Under this operation mode,
the thermal and electrical efficiencies for diesel were 2.5% and 61.9% respectively. With the original configuration, the corresponding efficiencies were 12.1% and 73.3%, respectively [31]. This decrease in efficiency was attributed to the heat loss incurred through the non-optimized combustion chamber walls and the distance from the flame to the hot end heat exchangers. In the old combustion chamber, the flame rests on top of the hot end heat exchangers thus leading to increased heat transfer, both radiation and convection, than with the new configuration. There was also a significant reduction in power output, from the rated 1 kW of the original engine. During operation with the new configuration fuelled by diesel, the power output was approximately 389 W, even with the inclusion of more fuel energy. This decrease in power is attributed to the exhaust gas bypassing the hot end heat exchangers, and heat loss through the combustion chamber walls leading to a decrease in the hot end temperatures.

Comparing the performance of diesel with the bio-oil/EtOH blend, efficiencies were higher at 3.4% power and 72.4 % thermal for the bio-oil/EtOH. Lower efficiencies for diesel are attributed to the greater loss of heat through the combustion chamber walls via radiation. A higher thermal efficiency with the bio-oil/EtOH blend is attributed to the higher thermal output, primarily from the exhaust. This larger extraction of energy is due to the difference in the amount of water present in the exhaust. Bio-oil’s inherent water content, and diesel’s lack of, allows for a greater amount of latent heat to be captured during the cooling of the exhaust to room temperature [37].

In commercial applications, heat loss through the optimized combustion chamber will be negligible as it is with the original combustion chamber of the WhisperGen. In order to evaluate the performance between bio-oil and diesel for an optimized system, a modified power efficiency, which neglects the heat losses through the combustion chamber walls, is proposed. This efficiency calculation is the fraction of the energy input, neglecting heat loss through the combustion chamber, which is converted into power. The modified power efficiency for diesel and the bio-oil/EtOH blend were 3.7% and 4.1%, respectively. Therefore, if heat loss through the combustion chamber was eliminated while the remaining system stayed as is, then the power efficiency of diesel and the bio-oil/EtOH blend are comparable.

A FTIR was used to measure CO and NOx emissions in the exhaust. CO emissions for the old configuration were lower than those of the new configuration due to the mode of combustion
and increased temperatures. The old configuration featured premixed combustion leading to better mixing between the fuel and air, whereas, due to the low atomizing and primary air, mixing was not as thorough in the new configuration. Better mixing leads to more complete combustion and thus lower CO emissions. Higher temperatures in the old configuration led to higher CO oxidation rates. NO\textsubscript{x} emissions, formed by the thermal mechanism, were lower in the new configuration. This is due to the lower temperatures inside the new combustion chamber due to the heat loss through the combustion chamber walls. During cold start conditions, CO emissions were high due to the temperatures being below the auto-ignition temperature.

CO emissions for the bio-oil/EtOH blend were very high at steady state values of 1550 ppm. This was due to poor atomization and a high fuel-air equivalence ratio, leading to larger droplets, and poor mixing and spray quality. CO emissions for diesel at steady state were 140 ppm. Diesel’s significantly lower CO emissions are attributed to its fully distillable nature and better atomization. NO\textsubscript{x} emissions were higher for bio-oil than for diesel. Even though the flame temperature of bio-oil is lower than that of diesel, there is inherent nitrogen content in the fuel which leads to formation of fuel bound NO\textsubscript{x}.

Extended operation with the bio-oil/EtOH blend had to be stopped after 97 minutes due to the clogging of the hot end heat exchangers. This clogging restricted the amount of primary air flow which could enter the combustion chamber. Decreases in the air flow rate, with the same fan power input, were signs of clogging. This led to an increase in the fuel-air equivalence ratio, until the mixture did not have enough oxygen to oxidize, leading to a blowout. As the clogging intensified, the power output began to decrease due to the decrease in exhaust flowing through the hot end heat exchanger. Since the primary source of heat transfer to the hot end heat exchanger is convection, the reduction in the exhaust flow rate decreases the heat transfer rate into the working fluid.

This study shows that the use of bio-oil for micro-CHP applications using a Stirling engine is possible, although there are many design challenges present. Using the experience gained from this study and others, some design principles are recommended for bio-oil use in Stirling engines in the next section. These design challenges can be solved given the current state of innovation with bio-oil combustion research and the interest in micro-CHP applications. It is shown in this study, through the modified efficiency, that the efficiencies of bio-oil and diesel
are comparable given that the combustion chamber heat loss is negligible. Although the emissions were relatively high for bio-oil, they can be decreased by introducing preheat and applying a refractory lining to the combustion chamber walls. Given the issues currently faced with remote power generation using diesel generations, bio-oil is a definite contender because of the benefits, both economically and in terms of sustainability.

6.2 Recommendations

Based on the design principles presented in Section 2.3.4, a new bio-oil combustion chamber and engine should be designed for commercialization. In order for a bio-oil fuelled Stirling engine to survive commercially, several requirements have to be fulfilled. Maintenance and reliability are the most important issues which should be addressed. The time periods between required maintenance checks should be maximized.

Preheating and Insulation

The poor ignition quality of bio-oil can be mitigated using a preheating system which can be accomplished using an economizer or a heat exchanger between the intake and exhaust similar to the one employed in the WhisperGen. A refractory lining should also be applied to the interior walls of the combustion chamber because it will aide in increasing localized temperatures. Higher temperatures inside the combustion chamber due to preheating and refractory lead to higher efficiencies and higher oxidation rates resulting in lower CO emissions. The combustion chamber should be started with another alcohol based fuel, which increases the temperatures inside for improved bio-oil combustion.

Working fluid

In order to improve the power efficiency, a suitable working fluid has to be chosen. For applications where high power density is required, hydrogen is the ideal choice. Helium also offers a much better alternative and has been utilized in previous investigations [21-22, 24-28]. High pressure must also be used in order to obtain improved efficiencies. Studies have indicated that using helium or hydrogen over nitrogen is able to improve efficiencies by 15 to 20% [16].
For practical purposes, a helium supply should be kept in close proximity to the engines location in order to facilitate recharging.

*Heat exchangers*

The WhisperGen featured annular finned hot end heat exchangers which, due to the small fin spacing, introduced problems for use with bio-oil. Another alternative are tubular heat exchangers which are primarily used for high power density applications [16]. In applications of solid biomass with Stirling engines, tubular heat exchangers have been used [21-22, 24-28]. Narrow passages have to be avoided in all components of the combustion chamber and engine for two reasons. Firstly, they cause a potential for clogging which limits the operation time of the engine. Secondly, they introduce large pressure drops which can be lead to instabilities in combustion.

Although deposits can be formed on these heat exchangers, automated cleaning technologies have been developed to mitigate this [22]. Deposits on these heat exchangers have the ability to decrease the heat transfer and thus decrease the power output. This cleaning technology features nozzles which blast compressed air at high pressure to rid the tubes of deposits, which increases the periods between required maintenance.

*Fuel Delivery System*

The fuel delivery system must be entirely composed of stainless steel due to the acidic nature of bio-oil. A peristaltic pump was employed in this study due to accuracy for measurement purposes, but any pump which is compatible with viscous fluids can be used. There must be two reservoirs, one for the starting fuel, which in this case was EtOH, and the other for a bio-oil/EtOH blend. The fuel system should also include hot filtration, which shows improved combustion characteristics and removes particulates. These improvements include shorter ignition delay, shorter burn times, and reduced coking [80].

*Control and Operation Strategy*

The control and operating strategy can be very similar to the WhisperGen. The engine should be started with another fuel, preferably an alcohol, for mixing purposes. This will heat the burner and the preheating system in order to provide improved conditions for ignition when bio-
oil is commenced. The fuel flow should also be intermittently switched to EtOH to prevent any issues with clogging and residue build-up which EtOH is able to mitigate. Variation in bio-oil properties should be kept to a minimum.

A flame sensor, similar to the flame rod, should be used. In the event of a blowout, the fuel flow must immediately be stopped, the pilot reignited, and the system flushed with EtOH. Allowing the system to cool down without the operation of EtOH could have devastating effects on the whole system, which would require manual cleaning. Pressure transducers should also be connected to the system to recognize signs of combustion instabilities. The fuel pump, intake air, and atomizing air should all be automated and controlled using a central program.
References


[50] G. D. Roy, Combustion processes in propulsion: control, noise, and pulse detonation,


Appendix A

WhisperGen Circuit Diagram
Figure A.1: WhisperGen controller circuit diagram [33].
Appendix B

Fuel Specifications
Table B.1: Fuel properties for diesel.

<table>
<thead>
<tr>
<th>Analysis</th>
<th>Method</th>
<th>Unit</th>
<th>Minimum</th>
<th>Maximum</th>
<th>Results</th>
</tr>
</thead>
<tbody>
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<td>Ash</td>
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<td>wt. %</td>
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<td>0.000</td>
<td></td>
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<tr>
<td>Determination of C, H and N</td>
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<td></td>
<td></td>
<td></td>
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<tr>
<td>70%</td>
<td></td>
<td>°C</td>
<td>260.1</td>
<td></td>
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</tr>
<tr>
<td>80%</td>
<td></td>
<td>°C</td>
<td>273.0</td>
<td></td>
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</tr>
<tr>
<td>90%</td>
<td></td>
<td>°C</td>
<td>282</td>
<td>338</td>
<td>290.5</td>
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<tr>
<td>End point</td>
<td></td>
<td>°C</td>
<td>317.8</td>
<td></td>
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<tr>
<td>Residue</td>
<td></td>
<td>%</td>
<td>0.4</td>
<td></td>
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<tr>
<td>Loss</td>
<td></td>
<td>%</td>
<td>0.6</td>
<td></td>
<td></td>
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<tr>
<td>Density at 15°C</td>
<td>ASTM D4052</td>
<td>kg/m³</td>
<td>828.3</td>
<td></td>
<td></td>
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<tr>
<td>Heat of combustion, net at 25°C</td>
<td>ASTM D4809</td>
<td>MJ/kg</td>
<td>42.792</td>
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<tr>
<td>Total sulphur by ultraviolet fluorescence</td>
<td>ASTM 5453</td>
<td>ppm (µg/g)</td>
<td>500</td>
<td>91</td>
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<tr>
<td>Kinematic viscosity at 40°C</td>
<td>ASTM D445</td>
<td>mm²/s (cSt)</td>
<td>1.9</td>
<td>4.1</td>
<td>1.930</td>
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<tr>
<td>Water</td>
<td>ASTM D6304A</td>
<td>mg/kg</td>
<td>170</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Water and sediment</td>
<td>ASTM D1796</td>
<td>% vol.</td>
<td>0.05</td>
<td>0.000</td>
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</tr>
</tbody>
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Appendix C

Engine Performance with Diesel
Figure C.1: Primary Air Flowrate.

Figure C.2: Combustion Chamber and Engine Temperatures.
Figure C.3: Power Output.

Figure C.4: Thermal Output.
Figure C.5: LHV Efficiencies.

Figure C.6: HHV Efficiencies.
Figure C.7: Modified Efficiencies.

Figure C.8: Fuel-air Equivalence Ratio.
Figure C.9: Thermal Output from Coolant and Exhaust.

Figure C.10: Energy Loss.
Appendix D

Engine Performance with Bio-oil/EtOH Blend
Figure D.1: Primary Air Flowrate.

Figure D.2: Combustion Chamber and Engine Temperatures.
Figure D.3: Power Output.

Figure D.4: Thermal Output.
Figure D.5: LHV Efficiencies.

Figure D.6: HHV Efficiencies.
Figure D.7: Modified Efficiencies.

Figure D.8: Fuel-air Equivalence Ratio.
Figure D.9: Thermal Output from Coolant and Exhaust.

Figure D.10: Energy Loss.
Appendix E

Engine Performance with Bio-oil/EtOH Blend
(Endurance Test)
Figure E.1: Primary Air Flowrate.

Figure E.2: Combustion Chamber and Engine Temperatures.
Figure E.3: Thermal Output.