THE PERFORMANCE AND EMISSIONS CHARACTERISTICS OF
HEAVY FUELS IN A SMALL, SPARK IGNITION ENGINE

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THE PERFORMANCE AND EMISSIONS CHARACTERISTICS OF
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ABSTRACT

THE PERFORMANCE AND EMISSIONS CHARACTERISTICS OF HEAVY FUELS IN A SMALL, SPARK IGNITION ENGINE

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This thesis research was conducted in pursuit of the DoD’s plan for the universal use of a heavy, low volatility hydrocarbon fuel, and the increased interest in bio-derived fuels for small Unmanned Aircraft Systems (UAS’s). Currently a majority of small UAS’s use small spark ignition engines for their high power densities. Typically, these systems use commercial off-the-shelf power plants that are not optimized for fuel efficiency. Increased fuel efficiency is being pursued alongside the ability to utilize military heavy fuels. A test stand using a 33.5 cc four-stroke, spark ignition, air-cooled, single cylinder engine was constructed. Research was conducted to establish the feasibility of converting the existing system to utilize JP-8 with the stock mechanical carburetion. The stock carburetion had difficulty maintaining a consistent air/fuel ratio across the entire engine operating range. To resolve this, an electronic fuel injection
system was developed to gain greater control over fuel mixture. An air-assisted electronic fuel injector was sourced from a scooter and adapted to work with the 33.5cc four-stroke engine. An aluminum injector mount was designed and machined and electronic controls were employed. Sensors on the valvetrain and crankshaft were developed as control signals for the injection system. The injector was characterized for flow rates and droplet size.

The test stand consisted of a small dynamometer coupled to the engine. Servo throttle actuation was designed and throttle position was monitored with a throttle position sensor. The air-assisted injector was supplied with regulated shop air, and the fuel pressurized using regulated nitrogen. A fuel flowmeter and mass air flowmeter monitored equivalence ratio. Work was done to facilitate smooth measurement of unsteady air flow intrinsic to single-cylinder engines.

Performance testing showed a decrease in brake specific fuel consumption (BSFC) while utilizing the injection system for the baseline fuel (Avgas 100LL), as greater mixture control (closer to stoichiometric) was realized. The engine was started using gasoline. Heavy fuel testing showed the ability to achieve required torque values at certain engine speeds. JP-8 was tested on the carbureted engine and fuel injected engine, showing a decrease in BSFC over baseline (carbureted avgas) with the carburetor and a further decrease in BSFC for the injected system.

Biofuels that were tested were plant-based Camelina (carbureted and injected) and a UDRI grown and extracted algae-based fatty acid methyl ester (FAME) biofuel blended with D2 diesel in a 20% algae/80% diesel blend. Performance results for the Camelina showed a decrease in BSFC for the carbureted engine and the largest decrease of all the
test fuels for the injected Camelina fuel. The algae blend showed less decrease in BSFC than the 100% diesel fuel.

Emissions data were recorded as well. The injection system demonstrated less CO emissions for the injected fuels over the carbureted fuels due to closer to stoichiometric mixtures. Similarly, unburned hydrocarbon emissions decreased when injection was employed. NOx emissions were higher for the fuel injected engine, as peak NOx emissions will typically occur at slightly lean conditions and the injected fuels were closer to peak NOx emission conditions.
ACKNOWLEDGEMENTS

I would like to thank my advisor, Dr. Sukh Sidhu, for all of his support during this research. Without his support, the channel between UDRI and AFRL’s Small Engine Research Lab may not have been opened. Dr. Sidhu was always willing to let me steer the research in whatever direction I wanted to accomplish our goals. Dr. Fred Schauer was the original contact and his interest and enthusiasm was essential to making this research happen. Captain Cary Wilson was the original operator and researcher for the small engine test stand. His cooperative work with me was instrumental in getting the new test stand and injection system up and running. Almost all of the data was gathered with two operators: Cary and I. Paul Litke of AFRL’s SERL and Dr. John Hoke of Innovative Scientific Solutions (ISSI) were always available to advise me about the test stand and experimental setup and helped make sure the stand was up to safety standards. Jacob Baranski wrote the injector control, spark control, and servo control code, and often aided with electronics troubleshooting. Dave Burris wrote the LabView code and user interface for the test stand, and was always fast and effective at adding features that we needed. Furthermore in SERL, Adam Brown, Rich Ryman, Justin Goffena, and D-Bay’s Curt Rice were always available to help with equipment questions and educating me about operating machine tools.
From UDRI, Ben Naguy worked as the student tech to the test stand and helped build and added to the test stand whenever needed. UDRI’s Linda Shafer and Rich Striebich analyzed biofuels, and provided me with all the needed fuel specs. Liliana Martinez and Moshan Kahandawala were my contacts at UDRI, and Sally Homsy made the algae biodiesel in UDRI’s lab.
# TABLE OF CONTENTS

ABSTRACT ................................................................................................................................. iii
ACKNOWLEDGEMENTS .............................................................................................................. vi
TABLE OF CONTENTS ............................................................................................................... viii
LIST OF FIGURES ....................................................................................................................... xi
LIST OF TABLES .......................................................................................................................... xv

CHAPTER 1 – INTRODUCTION ................................................................................................. 1
  1.1 - Overview ....................................................................................................................... 1
  1.2 - Research Objectives .................................................................................................... 6
  1.3 - Organization ............................................................................................................... 8

CHAPTER 2 – BACKGROUND ................................................................................................. 9
  2.1 - Internal Combustion Engines .................................................................................... 9
  2.2 - Two-Stroke Spark Ignition Engines .......................................................................... 10
  2.3 - Four-Stroke Spark Ignition Engines ......................................................................... 11
  2.4 - Compression Ignition Engines .................................................................................. 13
  2.5 – Research Engine Selection ....................................................................................... 13
  2.6 - Heat Management ..................................................................................................... 14
  2.7 - Octane and Knock ..................................................................................................... 15
  2.8 - Fuel Delivery ............................................................................................................. 20
  2.9 - Fuels .......................................................................................................................... 23
  2.10 - Ground-Based Engines versus Aircraft Engines .................................................... 26
  2.11 - Heavy Fuels in SI Engines ..................................................................................... 29
  2.12 - Cold Starting Heavy Fuels ..................................................................................... 30

CHAPTER 3 – EXPERIMENTAL SETUP ............................................................................... 33

CHAPTER 4 – EXPERIMENTAL RESULTS ........................................................................... 60
  4.1 - Performance Results ................................................................................................. 60
  4.2 - Emissions Results ..................................................................................................... 71
<table>
<thead>
<tr>
<th>Section</th>
<th>Title</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.3</td>
<td>Uncertainty</td>
</tr>
<tr>
<td>5</td>
<td>CHAPTER 5 - CONCLUSIONS</td>
</tr>
<tr>
<td>6</td>
<td>CHAPTER 6 – FUTURE WORK</td>
</tr>
<tr>
<td>88</td>
<td>BIBLIOGRAPHY</td>
</tr>
<tr>
<td>91</td>
<td>APPENDIX A</td>
</tr>
<tr>
<td>91</td>
<td>Contour Plot MatLab Code</td>
</tr>
<tr>
<td>96</td>
<td>Spark and Servo Control Program</td>
</tr>
<tr>
<td>102</td>
<td>Injector Control Program</td>
</tr>
<tr>
<td>108</td>
<td>APPENDIX B</td>
</tr>
<tr>
<td>108</td>
<td>Performance Analysis Terms</td>
</tr>
<tr>
<td>110</td>
<td>APPENDIX C</td>
</tr>
<tr>
<td>110</td>
<td>Max Machinery Fuel Flowmeter</td>
</tr>
<tr>
<td>111</td>
<td>TSI 4021 Mass Air Flowmeter</td>
</tr>
<tr>
<td>112</td>
<td>Magtrol ED-715-6N</td>
</tr>
<tr>
<td>113</td>
<td>Electric Starter Mount Plate</td>
</tr>
<tr>
<td>114</td>
<td>Electric Starter Crankshaft Coupler</td>
</tr>
<tr>
<td>115</td>
<td>Aluminum Injector Manifold</td>
</tr>
<tr>
<td>116</td>
<td>APPENDIX D – ASM 2011 PAPER</td>
</tr>
<tr>
<td>116</td>
<td>Abstract</td>
</tr>
<tr>
<td>117</td>
<td>Nomenclature</td>
</tr>
<tr>
<td>117</td>
<td>I. Introduction</td>
</tr>
<tr>
<td>118</td>
<td>II. Experimental Setup</td>
</tr>
<tr>
<td>119</td>
<td>III. Experimental Data - Performance</td>
</tr>
<tr>
<td>121</td>
<td>IV. Experimental Data – Emissions</td>
</tr>
<tr>
<td>123</td>
<td>V. Uncertainty</td>
</tr>
<tr>
<td>124</td>
<td>VI. Fuel Injection</td>
</tr>
<tr>
<td>125</td>
<td>VII. Algae Based Biodiesel</td>
</tr>
<tr>
<td>125</td>
<td>VIII. Conclusions</td>
</tr>
<tr>
<td>126</td>
<td>APPENDIX E – JPC 2011 PAPER</td>
</tr>
<tr>
<td>126</td>
<td>Abstract</td>
</tr>
<tr>
<td>127</td>
<td>I. Introduction</td>
</tr>
<tr>
<td>127</td>
<td>II. Experimental Setup</td>
</tr>
<tr>
<td>131</td>
<td>III. Fuel Performance Characterization</td>
</tr>
</tbody>
</table>
IV. Emissions Characterization ............................................................................................. 133
V. Uncertainty ....................................................................................................................... 134
VI. Conclusions ..................................................................................................................... 136
VII. Recommendations and Future Work ............................................................................. 136
LIST OF FIGURES

Figure 1: Small engine and turbo-shaft power density versus BSFC .............................................. 2
Figure 2: Silver Fox UAV................................................................................................................ 3
Figure 3: Fuji-Imvac BF-34EI ......................................................................................................... 4
Figure 4: Pressure profiles for different magnitudes of knock (Ferguson & Kirkpatrick, 2001)... 17
Figure 5: Piston damage as a result of prolonged knock ............................................................... 17
Figure 6: Stock spark timing for Fuji engine ................................................................................. 19
Figure 7: Magtrol ED-715-6N Hysteresis Dynamometer .............................................................. 34
Figure 8: Lovejoy LF2 couplers and sacrificial bearing................................................................. 35
Figure 9: High speed video screenshot of coupler wobble ............................................................ 36
Figure 10: Oil line re-routing and catch can .................................................................................. 37
Figure 11: Pressurized fuel tank .................................................................................................... 38
Figure 12: Fuel flowmeter, filter, pneumatic valve, and plumbing ............................................... 38
Figure 13: FEMA starting system.................................................................................................. 39
Figure 14: Fuji engine pull-starter ................................................................................................. 40
Figure 15: TSI mass air flowmeter ................................................................................................ 40
Figure 16: Air damper drum with polyurethane foam inserts......................................................... 41
Figure 17: Air damper with foam sections..................................................................................... 41
Figure 18: High speed screenshot of air flow oscillations (left plot in blue) and smoothed flow (right plot in blue) ........................................................................................................ 42
Figure 19: Injector mounted on engine .......................................................................................... 43
Figure 20: Example of scooter injector signals.............................................................................. 44
Figure 21: Solid-state injector relays ............................................................................................. 44
Figure 22: Injector phasing sensor ................................................................................................. 45
Figure 23: Encoder, encoder pickup, and CPS .............................................................................. 46
Figure 24: Aluminum injector manifold ........................................................................................ 46
Figure 25: Orbital fuel injector flow rates ..................................................................................... 49
Figure 26: Schematic of Malvern Series 2600 ............................................................................... 50
Figure 27: Droplet size results for air-assisted injector ................................................................. 51
Figure 28: Manual throttle control bracket .................................................................................... 51
Figure 29: Throttle control servo on carbureted setup and injector manifold .............................. 52
Figure 30: Fuji throttle servo hysteresis ........................................................................................ 53
Figure 31: CTS 500 Series throttle position sensor and throttle hysteresis with TPS ................. 53
Figure 32: Inlet air heater for cold weather testing ........................................................................ 54
Figure 33: Small engine LabView control program ........................................................................ 56
Figure 34: PIC18 microprocessor chip .......................................................................................... 57
Figure 35: Enerac 700 emissions unit and exhaust pickup tube .................................................... 57
Figure 36: Oil trap and exhaust sample flowpath ......................................................................... 58
Figure 37: Propeller load points and carbureted actual load points achieved ............................... 61
Figure 38: Carbureted BSFC and phi versus engine speed for 17x10 and 18x12 props  
(respectively) ................................................................................................................................. 62
Figure 39: Equivalence Ratio Carbureted versus PFI (Avgas) ........................................................ 63
Figure 40: 17x10 injected BSFC versus engine speed for all test fuels + carbureted avgas and JP- 
8 ..................................................................................................................................................... 64
Figure 41: 18x12 injected BSFC versus engine speed for all test fuels + carbureted avgas and JP-
8 ..................................................................................................................................................... 65
Figure 42: Carbureted avgas BSFC versus torque versus RPM contour plot with propeller load 
lines .................................................................................................................................................. 66
Figure 43: Injected avgas BSFC versus torque versus RPM contour plot with propeller load lines .......................................................................................................................................................................................... 66

Figure 44: Injected JP-8 BSFC versus torque versus RPM contour plot with propeller load lines67

Figure 45: Example of engine knock ............................................................................................................................. 68

Figure 46: 17x10 CO Emissions ........................................................................................................................................ 72

Figure 47: 18x12 CO Emissions ........................................................................................................................................ 73

Figure 48: 17x10 CO2 Emissions ........................................................................................................................................ 74

Figure 49: 18x12 CO2 Emissions ........................................................................................................................................ 74

Figure 50: 17x10 UHC Emissions ........................................................................................................................................ 76

Figure 51: 18x12 UHC Emissions ........................................................................................................................................ 77

Figure 52: 17x10 NOx Emissions ........................................................................................................................................ 78

Figure 53: 18x12 NOx Emissions ........................................................................................................................................ 78

Figure 54: Max Machinery Flowmeter ........................................................................................................................................ 110

Figure 55: TSI 4021 Mass Air Flowmeter ........................................................................................................................................ 111

Figure 56: Magtrol Dynamometer Specifications ........................................................................................................................................ 112

Figure 57: Electric Starter Mount Plate ........................................................................................................................................ 113

Figure 58: Electric Starter Crankshaft Coupler ........................................................................................................................................ 114

Figure 59: Aluminum Injector Manifold ........................................................................................................................................ 115

Figure 60: Small Propulsion Power Density vs. BSFC ........................................................................................................................................ 117

Figure 61: Dynamometer/engine coupler arrangement ........................................................................................................................................ 118

Figure 62: Inlet air heater and mass air flowmeter with drum air damper ........................................................................................................................................ 119

Figure 63: Crankcase vent re-routing and emissions pick up location ........................................................................................................................................ 119

Figure 64: Experimental versus theoretical engine loading ........................................................................................................................................ 120

Figure 65: 22x10, 4720 rpm, 73.9psi bmep cruise point in-cylinder pressure ........................................................................................................................................ 120

Figure 66: BSFC and phi vs rpm for 17x10 and 18x12 props (respectively) ........................................................................................................................................ 121

Figure 67: CO and CO2 Emissions vs bmep and phi ........................................................................................................................................ 122
Figure 68: 17x10 Hydrocarbons vs bmep and phi ................................................................. 122
Figure 69: 18x12 Hydrocarbons vs bmep and phi ................................................................. 123
Figure 70: NOx vs bmep and phi ......................................................................................... 123
Figure 71: Engine to Dyno Coupler and Cooling Air ............................................................ 128
Figure 72: Engine Test Stand ............................................................................................... 128
Figure 73: Mass Air Flowmeter ......................................................................................... 128
Figure 74: Pressurized Fuel Tank and Fuel Flowmeter ......................................................... 129
Figure 75: Injector Droplet Sizes ....................................................................................... 130
Figure 76: Injector Mount ................................................................................................. 130
Figure 77: PIC18 Microprocessor Chip and 36-Tooth Encoder ........................................... 130
Figure 78: Solid State Relays, Throttle Servo, and TPS ....................................................... 131
Figure 79: JP-8 Carburetion vs. PFI System ...................................................................... 131
Figure 80: Avgas and JP-8 Performance at Phi=1 ............................................................. 132
Figure 81: BSFC vs RPM for 18x12 and 17x10 props (respectively) ................................. 132
Figure 82: CO and CO2 Emissions vs BMEP for 18x12 (left) and 17x10 (right) ............... 133
Figure 83: UHC and NOx vs BMEP for 18x12 (left) and 17x10 (right) ............................... 134
LIST OF TABLES

Table 1: Test Fuel Properties ......................................................................................................... 26
Table 2: Propeller load points ........................................................................................................ 55
Table 3: BSFC % change versus carbureted avgas ................................................................. 69
Table 4: Instrumentation Measurement Error .............................................................................. 123
Table 5: Low Condition Uncertainty (Q=3.81cc/min, N=1500rpm) ........................................... 124
Table 6: High Condition Uncertainty (Q=15.2cc/min, N=7500rpm) .......................................... 124
Table 7: Instrumentation Measurement Error (2) ........................................................................ 135
Table 8: Uncertainty for Equivalence Ratio (Low and High) ................................................... 135
Table 9: Uncertainty in Torque Measurement ............................................................................. 136
CHAPTER 1 – INTRODUCTION

1.1 - Overview

The Department of Defense has a sustained interest in the universal use of heavy, low volatility hydrocarbon fuels in the pursuit of a single fuel concept. Logistically, a single fuel concept would eliminate the need to ship different types of fuels around the world. This would reduce costs and simplify the stocking of available fuel. This would also reduce the risk of an incorrect fuel causing damage to an engine or its associated systems. Heavy fuels are less volatile and therefore safer to transport and store.

Many Remotely Piloted Aircraft (RPA’s) use small, spark ignition, internal combustion engines for propulsion. These engines are designed to run on high octane, highly volatile fuels. Spark ignition engines are used because of their high power density (hp/lbm) compared to most compression ignition engines. High overall pressure ratio (OPR) gas turbine turbo-shaft engines are capable of achieving very high power densities, but not with the required brake specific fuel consumption (BSFC) levels that are able to be achieved with advanced piston-type, spark and compression ignition, internal combustion engines. Gas turbine engines can achieve very good BSFC at high throttle openings, but in cruise conditions, smaller throttle openings yield poor BSFC. Spark ignition engines tend to show higher power densities than compression ignition engines with relatively low BSFC. This is due to the lower compression ratio of spark ignition engines requiring less engine mass to cope with engine stresses. Figure 1 shows
the desired region of power density and BSFC for designing future RPA propulsion systems to operate in. Much focus is being placed on developing spark ignition engines that can produce lower BSFC, while further increasing power density. One way that this could be achieved is by utilizing fuels with higher energy content. Heavy fuels tend to have higher energy contents than the fuels traditionally used in spark ignition engines, but there are some attributes to these heavy fuels that need to be overcome to allow these fuels to run reliably in current spark ignition engine systems. The primary issues with heavy fuels are the relatively low (and unregulated) equivalent octane rating, and the difficulty in vaporizing the fuel to achieve satisfactory combustion efficiency due to lower vapor pressures than gasoline.

![Figure 1: Small engine and turbo-shaft power density versus BSFC](image-url)
The engine used in this study was the Fuji-Imvac BF-34EI (Figure 3). This engine is a 33.5cc, spark ignition, 4-stroke, single-cylinder, air-cooled, internal combustion engine that is used in the Silver Fox Unmanned Aerial System (UAS).

![Figure 2: Silver Fox UAV](image)

Engine weight was 5.9 lbs and bore and stroke were 39mm and 28mm, respectively. This engine utilized overhead valves driven by pushrods actuated via a camshaft. Mechanical carburetion comes with the commercial-off-the-shelf (COTS) engine as a way to meter fuel and air. The stock carburetor has two mixture adjustment needles: one for low speed operation and the other for high speed operation. The engine was factory-rated for gasoline and peak horsepower and torque were quoted at 2.0 hp and 1.45 lbf-ft, respectively. It is of value to note that these performance numbers were never witnessed in the lab. Horsepower numbers peaked at approximately 1.3-1.4 hp and torque peaked at approximately 1.0 lbf-ft.
This engine was designed for the RC hobby industry. The parent company of Fuji-Imvac is Fuji Heavy Industries, which also owns Subaru. Subaru makes a small engine line under the name Robin, which is based on the exact same engine, albeit with a different plastic engine cover, magneto ignition instead of electronic ignition, and a pull starter. These engines are sold for powering generators and for yard work applications. During the research process, it was discovered that these Robin engines could be purchased for a fraction of the cost of the Fuji-Imvac engines and equal or better performance achieved. The reason for this is a stricter quality check on Robin engines going out to be used for commercial applications compared to the Fuji-Imvac engines that were going out to be used on short RC aircraft flights.

The Fuji engine was also considered COTS propulsion system. This means it was available to a commercial market and not necessarily designed to the same life and quality requirements expected of systems developed exclusively for military applications. The reason COTS systems have been used for RPA’s is that RPA’s came into active use relatively quickly and the COTS propulsion systems were already available. This means
development time could be shortened by using COTS systems and platforms could be
fielded relatively quickly. Most of these systems were developed for commercially
available fuels and with less stringent power density requirements than would normally
be expected for military systems. Currently, much effort is being put into improving the
COTS propulsion systems or even developing ground-up military spec propulsions
systems to replace the COTS systems.

The Air Force uses Jet-Propellant 8 (JP-8), a heavy, kerosene-based fuel, to power
large and small gas turbines and for compression ignition engines in trucks. This research
will focus on JP-8 as the heavy fuel of choice to satisfy the single fuel concept focused on
by the military. Beyond testing JP-8, on the commercial side of propulsion research, other
heavy fuels are being developed. Bio-derived heavy fuels, often referred to as bio-diesel
or green diesel, are undergoing evaluation as a method for creating domestically sourced,
renewable fuels.

Domestically sourced fuels could relieve some dependence on foreign oil. Also,
bio-derived fuels could also have a positive environmental impact when compared to
petroleum-based fuels. Bio-derived fuels still produce carbon emissions when burned, but
the plants that the fuel is based on also absorb carbon while they grow. In this respect, the
net release of carbon from the start of the process (planting and growing the fuel) to the
end of the process (burning the fuel in an engine) can be far less than simply burning
petroleum-based fuels. This gathering of carbon in the growing process (or carbon
sequestration) has sparked the research of many different types of biofuels. This research
focused on a number of fuels that were already available on the commercial market, as
well as fuels that were not yet available and were in the research phases.
One fuel that was looked at was algae biodiesel. This fuel was produced by the University of Dayton Research Institute’s algae fuels research group. The algae were grown in-house and the extraction process was conducted in UDRI’s own labs. The test fuel quantities at the time of production were quite low, so the algae fuel was blended with commercially available D2 diesel to increase the test fuel volume to a useable amount. Some biodiesel fuels are available for the general public to buy. For this research, the algae fuel was blended in the same ratio as B20. B20 is the name given to a 20% blend of biodiesel with 80% petroleum-derived diesel (by volume).

Another tested fuel was HRJ8-Camelina biodiesel. This was a Camelina-derived hydrotreated renewable jet fuel that was provided by Wright Patterson Air Force Base’s (WPAFB) Air Force Research Lab (AFRL) fuels branch. Camelina is a member of the mustard family and a distant relative of Canola.

Finally, standard D2 diesel was tested. This fuel is commercially available and commonly used in compression ignition engines in trucks, farm equipment, construction equipment, and for industrial applications. Further detail about all of these fuels will be discussed in the fuels chapter.

1.2 - Research Objectives

The primary objective of this research is to evaluate the effects of utilizing alternative heavy fuels in the aforementioned spark ignition engine as the preliminary work toward utilizing a single fuel on all military platforms while reducing brake specific fuel consumption. Furthermore, this research will serve as a preliminary comparison of the experimental algae-based biodiesel with other available fuels. The primary objective
of this research can be split into more specific objectives to be accomplished throughout
the course of the research.

1. Research Objective Breakdown:

   a. Demonstrate that the engine can run on the test stand in stock
      configuration, with the stock fuel

   b. Demonstrate that the engine can run with the stock carburetor
      using the test fuels

   c. Compare the performance of the test fuels to the stock fuel on a
      BSFC basis using carburetion

   d. Compare the emissions characteristics of the test fuels to the stock
      fuel using the stock carburetor
         i. Include carbon dioxide (CO2), carbon monoxide (CO),
            oxides of nitrogen (NOx) and unburned hydrocarbon
            emissions (UHC’s)

   e. Source and test a suitable electronic fuel injector for the engine
      i. Develop electronic controls for the injection system
         ii. Characterize the injector
             1. Fuel flow rates at different pressures
             2. Droplet size measurements

   f. Demonstrate that the engine can run with the new injection system

   g. Compare the performance of the test fuels to the stock fuel on a
      BSFC basis using the electronic fuel injection
h. Compare the emissions characteristics of the test fuels to the stock fuel using the electronic fuel injection

2. Design and Fabrication
   a. Design and fabricate a manifold for mounting the injection system
      i. Include throttle control and throttle position feedback
   b. Design and fabricate a method for remote starting of the engine
   c. Design and plumb injector air and fuel supplies

1.3 - Organization

Chapter I served as an introduction to the motivation for this research and described the research objectives to be accomplished.

Chapter II will outline some basic fundamentals of internal combustion engines necessary for completion of this research and will discuss the results of relevant research papers in this field.

Chapter III will discuss the experimental setup of the test rig and associated hardware for completion of this work.

Chapter IV will discuss the results and analysis of the experiments.

Finally, Chapter V will summarize all of the work and provide suggestions towards future work that could be conducted to continue this research.
CHAPTER 2 – BACKGROUND

This chapter will discuss several fundamentals of internal combustion engines instrumental for the completion of this research, including the 4-stroke Otto cycle, octane rating, chemical properties of test fuels, and abnormal combustion (knock). Also, types of fuel metering systems will be discussed, including mechanical carburetion and electronic fuel injection. More depth will be covered on fuel injection, including types of injectors, injection control methods, and fuel spray characteristics. Next, types of engine emissions will be discussed and methods for gathering emissions data will be noted.

2.1 - Internal Combustion Engines

The internal combustion engine has been around for more than 100 years. It is currently the main source of propulsive power for automobiles and provides power for equipment including weed trimmers, chainsaws, backup power generators, boats, and small airplanes. The internal combustion engine is so widely used because of its high power density. Liquid hydrocarbon fuels have a volumetric energy density that can’t even be approached with current electric battery technology. Most internal combustion engines use the reciprocating piston-cylinder principle in which a piston fits tightly into a cylinder in which it oscillates. The piston is connected by a connecting rod to a crankshaft that translates the reciprocating motion of the piston into angular rotation of the crankshaft. Air and fuel are combusted together inside the combustion chamber (the volume
displaced by the piston inside the cylinder bore) to produce force on the piston, driving it down and transmitting power to the crankshaft.

The method for mixing the fuel and air, type of fuel, method of initiating combustion of the mixture, and control of the flow in and out of the combustion chamber are all factors that differentiate different types of internal combustion engines. The two most common types of internal combustion engine are the spark ignition engine and the compression ignition engine. Nikolaus Otto developed the first spark ignition engine in 1876 (Ferguson & Kirkpatrick, 2001). His engine required a spark to ignite. Otto cycle engines use an electric arc (or spark) to initiate the combustion of the fuel/air mixture, so they are often called spark ignition engines. The spark contains enough energy to start the chemical reaction between the oxygen in the air and the fuel. After the reaction has started, it spreads through the rest of the mixture and consumes all of the available fuel and oxygen (ideally); converting fuel and oxygen into useable heat and pressure, carbon dioxide, water, and some other unwanted emissions, like carbon monoxide, unburned hydrocarbons, or oxides of nitrogen. The heat and pressure are what are used to put work into the piston, and therefore the crankshaft. Within the category of spark ignition are two main types of engines: two-stroke and four-stroke.

2.2 - Two-Stroke Spark Ignition Engines

Two-stroke engines were developed in 1878 by Dugald Clerk. Two-stroke engines use two strokes of the piston (up and down) to complete an entire cycle (Heywood, 1988). A two-stroke engine has ports that allow air and fuel into the combustion chamber. As the piston moves down, it uncovers the ports. There is an intake port and exhaust port. Usually the exhaust port is uncovered slightly before the intake
port. As the spent exhaust rushes out of the exhaust port, fresh air and fuel enter the intake port; the downward motion of the piston causing a lower pressure in the combustion chamber than in the crankcase volume below the piston. The piston reaches bottom dead center (BDC) and starts to travel back up in the cylinder, covering both the intake and exhaust ports. Now the mixture is compressed by the piston as the fuel and air mixture has nowhere to go. Before the piston reaches top dead center (TDC) the spark plug fires and ignites the mixture. The mixture starts to burn and release heat and creates pressure in the clearance volume (volume between the piston and cylinder head). The peak pressure of the combustion event ideally takes place after top dead center (ATDC) and forces the piston to move down. This force is where the useful work of the engine comes from. As the piston moves past the exhaust and intake ports, the cycle starts over. Two-stroke engines can be very light, simple and power dense, but tend to be less efficient than other types of engines. This is because both intake and exhaust ports are open simultaneously, often allowing some unburned fuel to get through to the exhaust port, causing increased fuel consumption and has negative emissions effects. Two-stroke engines are often used for applications that need to have a high specific power (power to weight ratio), such as for weed trimmers and chainsaws.

2.3 - Four-Stroke Spark Ignition Engines

The four-stroke engine addresses some of the shortcomings of the two-stroke engine. Four-stroke engines usually have poppet valves that are actuated by a camshaft that runs at half of the crankshaft speed. The entire cycle is completed in four strokes of the piston, or two complete crankshaft revolutions. As the piston travels downward, the intake valve is opened and fuel and air enter the combustion chamber. As the piston nears
BDC, the intake valve closes and the piston moves upward, compressing the charge. Before TDC, the spark is fired and combustion takes place, forcing the piston downward. The high pressure gases expand, and work is extracted as the piston travels downward. On the last stroke, the exhaust valve opens, and the piston pushes spent gases out of the combustion chamber. Separating intake and exhaust into separate strokes prevents the four-stroke engine from wasting unburned fuel. The added valves and camshaft make for a heavier engine compared to a two-stroke. Four-stroke engines usually operate at a higher compression ratio than two-stroke engines, allowing for a higher thermal efficiency.

Otto cycle, or spark ignition engines use thermodynamic principles that assume that the combustion event is happening so fast that the piston does not have time to move, so the combustion is taking place in a constant volume. The compression ratio of an engine is the ratio of the volume of the combustion chamber at BDC to the volume of the combustion chamber at TDC. The thermal efficiency of constant volume combustion with a compression ratio, \( r \), and specific heat ratio of air \((cp/cv), \gamma \) is shown in Equation 1 (Ferguson & Kirkpatrick, 2001).

\[
\eta_{\text{thermal}} = 1 - r^{1-\gamma}
\]  

For instance, an engine with a compression ratio, \( r \), of 8:1 and air specific heat ratio, \( \gamma \), of 1.4, the maximum theoretical thermal efficiency would be 56.5%. An engine with a compression ratio of 12:1 and the same \( \gamma \) would have a max thermal efficiency of 63%. It is important to note that no IC engine would actually achieve such efficiencies; as there are many other losses to consider.

Spark ignition engines, until recently, mix the fuel and air coming into the engine before the intake valve is opened. The fuel is mixed upstream of the intake valve so it has
time to evaporate and become a gas. Liquid fuel will not burn with oxygen until it has become a gas. For this reason, smaller fuel droplet sizes are desirable, since they have a higher surface area to volume ratio and evaporate faster. Faster evaporation means that the chances of all of the fuel being burned and turned into useful heat are higher. Larger droplets sometimes don’t evaporate completely, and can be rejected from the engine through the exhaust, wasting fuel and causing undesirable unburned hydrocarbon emissions.

2.4 - Compression Ignition Engines

The other main type of IC engine is the compression ignition engine. The compression ignition (CI) engine was developed by Rudolph Diesel in 1897 (Ferguson & Kirkpatrick, 2001). For this reason, compression ignition engines are often referred to as Diesel engines. The engine was designed to utilize the direct injection of liquid fuel into the combustion chamber. CI engines do not use a spark to initiate combustion. Diesel engines use the high heat generated by compressing air to auto-ignite fuel as it is injected. Fuel is injected and immediately starts to burn, raising the pressure of the cylinder and moving the piston. Diesel engines generally run a higher compression ratio than SI engines. Diesel engines are generally heavier than SI engines; the engines must be built to withstand the higher stresses due to the higher compression ratios.

2.5 – Research Engine Selection

Many small UAS’s have utilized two-stroke engines because of their high peak power to weight ratio, which is important when every ounce of weight on a small aircraft must be accounted for. On the other hand, the higher efficiency of four-stroke engines holds more potential for long-cruise UAS’s, while still holding a large weight advantage
over compression ignition engines. For this reason, the four-stroke SI engine, splitting the middle ground between specific power and efficiency has been chosen for this engine research. Four-stroke SI engines are also purchased at a lower price point than CI engines that are available today. For the foreseeable future, until further market development is complete concerning the weight reduction of high compression CI engines to levels suitable for small UAS’s, the four-stroke SI engine will be the engine of choice when long cruise times are called for. In the coming sections, past research into the use of heavy, low octane fuels in four-stroke SI engines will be discussed, as these fuels contain more energy per volume and are more readily available for many military applications.

2.6 - Heat Management

IC engines create a lot of waste heat. If the engine is not cooled, heat could build to the point where the temperature surpasses the melting point of the engine materials. To manage engine heat, engines are cooled. The two main types of engine cooling are air-cooled and water-cooled. Air cooling utilizes high surface area fins, cast into the engine itself, to maximize convective heat transfer, caused by air flowing past the engine. On engine applications such as in motorcycles and aircraft, plenty of air is available to cool the engine solely based on the engine’s movement through the air. Stationary applications may require a cooling fan to move air over the engine’s cooling fins and maintain an acceptable operating temperature. The lubricating oil that is pumped through the engine also serves to cool the engine; some air-cooled engines have an oil radiator for heat rejection. The other type of engine cooling, water-cooling, utilizes passages through the engine block material, that carry pumped liquid, commonly water mixed with a glycol-based antifreeze solution, through the engine material, cooling vital engine parts. Heat is
then dissipated from the coolant via a high surface area heat exchanger called a radiator. Water has a much higher specific heat than air, allowing water-cooled engines to maintain consistent operating temperatures much more effectively, and support more peak power without engine damage. Air-cooling is a passive process, requiring no monitoring or sensors. It is also less complex and therefore less expensive. Water-cooling requires a thermostat, water pump, and radiator. It is more effective, but more complex, heavy, and expensive.

Single cylinder, air-cooled engines have a high cooling surface area to displaced volume ratio. This is good for engine life, but can have some adverse thermodynamic effects. One problem is that the high cooling surface area tends to cause heat loss through the cylinder walls. Having less heat around the walls inside the combustion chamber can cause the combustion reaction to slow as it approaches the cylinder walls, sometimes even stopping entirely from lack of energy. This is called flame quenching and has negative effects on fuel consumption rates and emissions. Water-cooled, multi-cylinder engines tend to be able to keep a constant temperature throughout the engine block, facilitating less flame quenching, while keeping the engine in a safe operating temperature.

2.7 - Octane and Knock

The octane rating of a fuel is one of the key characteristics to consider when choosing a fuel for an engine. The octane rating system was developed in 1927 (Wilson, 2010), and it uses two reference fuels, iso-octane and n-heptane, as a high and low reference fuel to compare other fuels to. The rating gives the user an idea of how likely the fuel is to self-ignite by means other than a spark. A high octane rating (as with iso-
octane) means that the fuel is less likely to auto-ignite or knock. A low octane rating (as with n-heptane) means the fuel will be more likely to auto-ignite or knock.

There are two different situations that can arise from a fuel that has a lower than desirable octane rating for an engine: pre-ignition and knock. Pre-ignition happens when an engine is run with either a fuel of too low octane rating or engine heat builds too much. In this situation, the fuel/air mixture ignites itself and starts to combust before the spark event. This causes in-cylinder temperatures to rise very quickly and it causes very unstable pressure to take place inside the cylinder. Pre-ignition can lead to a runaway situation in which pre-ignition causes in-cylinder temperatures to rise, which increases in-cylinder temperature further, and causes even more pre-ignition to happen, until eventually something fails. Knock happens when auto-ignition happens after the spark event. The spark occurs, causing pressure to rise in the entire combustion chamber as the flame front spreads toward the cylinder walls. Rising pressure causes other parts of the unburned mixture to auto-ignite. Unstable combustion pressure across the piston surface induces a characteristic “pinging” sound. Knock is less dangerous than pre-ignition, but it can still cause unsuitably high in-cylinder pressures, and eventually piston or valve failure. Peak pressure is higher when an engine is knocking, and in some cases can actually increase power to an extent, but at the cost of possible engine damage. When knock becomes intense, power falls off, as end gas (the unburned mixture ahead of the flame front) is consumed at a rate 5-25 faster than the flame front would spread (Heywood, 1988). This rapid consumption puts peak pressure earlier in the cycle than where it should be for maximum engine torque, causing low of power. Also, since peak pressure can be pushed forward into the compression stroke, peak temperatures can
rise rapidly. Figure 4 shows what slight and intense knock look like compared to a normal combustion pressure profile. Figure 5 shows piston damage that can result from prolonged engine knock. In this case, JP-8 fuel was knocking at a high load and high temperature for approximately 30 minutes before permanent piston damage occurred.

![Pressure profiles for different magnitudes of knock](image1)

**Figure 4: Pressure profiles for different magnitudes of knock (Ferguson & Kirkpatrick, 2001)**

![Piston damage](image2)

**Figure 5: Piston damage as a result of prolonged knock**

Iso-octane is assigned an octane rating of 100 and n-heptane is assigned an octane rating of 0. Fuels are assigned an octane rating based on their propensity to knock at the
same conditions as a mix of iso-octane and n-heptane. If fuel tests were to show that a fuel had an octane rating of 85, it would have the same propensity to knock as a mixture of 85% iso-octane and 15% n-heptane. There are two methods for testing the octane rating of a fuel: the research octane number (RON) method and the motor octane number (MON) method. The difference between the tests lies in the differing test conditions that a fuel is evaluated at. In the United States, an antiknock index (AKI) is used to assign a fuel’s octane rating at the pump. The AKI is the average of the RON and MON ratings.

Spark timing is of particular interest when discussing knock in engines. The ignition system senses where the crankshaft is and fires the spark slightly before top dead center (BTDC). Spark timing is reported in degrees BTDC. The reason the spark is fired BTDC is that it takes time for the spark to initiate combustion and then it takes time for the combustion to reach peak pressure where it can apply the most force to the piston. Ideal spark timing for maximum brake torque (MBT) will place peak pressure at about 16 degrees ATDC and 50% of charge mass burned at about 10 degrees ATDC (Heywood, 1988). Stock spark timing for the Fuji engine is shown in Figure 6. The spark timing is fully advanced by 4000 rpm. The slope of the advancing section of the plot provides a smooth transition from the less advanced low engine speed ignition timing, to the more advanced ignition timing at higher engine speeds.
Figure 6: Stock spark timing for Fuji engine

With a high octane fuel, risk of pre-ignition and knock are low, so the spark event can be placed earlier; sometimes as early as 40 degrees BTDC. One way to mitigate knock in engines is to move spark timing closer to TDC. This puts peak pressure further into the expansion stroke, and lowers the overall temperature and pressure, reducing knock. Intense knock increases BSFC because less power is produced for the same fuel flow. Decreased burn times move the peak in-cylinder pressure point closer to or prior to TDC, further from the MBT point which is after TDC. Moving spark timing closer to TDC (retarding timing) can bring BSFC back down to a point, but usually not back to the level of maximum spark advance with no knock. For this reason, engines that have a higher compression ratio generate more heat through compression and tend to require higher octane fuels to operate without knock. Lately, knock detection techniques have advanced so that lower octane pump fuels can be used in higher compression SI engines.
Under higher load conditions, which tend to induce more knock, the car’s ECU can monitor when knock is starting to occur and retard spark timing accordingly, this comes at the cost of some high load power compared to higher octane fuels, but considerable cost savings can be incurred from utilizing lower octane gasoline fuels.

Automobiles often use piezoelectric knock sensors on the engine block to detect frequencies associated with engine knock. In a lab it is common to utilize in-cylinder pressure transducers to monitor pressure oscillations to detect knock. A band-pass filter can be used to observe a specific knock frequency range; the amplitude of the detected frequencies can be associated with a knock pressure. The human ear can be a surprisingly sensitive knock detector (Heywood, 1988). A length of tubing connected to the engine cylinder head and leading out of a test cell to the operator can be an effective method of detecting knock.

2.8 - Fuel Delivery

For many years, the most common way to deliver fuel to a spark ignition engine was through the use of a mechanical carburetor. Carburetors use the principle that as air passes through a narrowed cross-section in a flowpath (a venturi) the air will accelerate and the pressure will drop. It is in this narrowed section that a fuel jet, at a slightly higher pressure than ambient, protrudes. The fuel is entrained into the flowing air at a lower pressure and breaks up into droplets that start to evaporate. A throttle plate (or butterfly valve) sits downstream of the venturi and controls the amount of air allowed into the engine. Without the butterfly valve, SI engines would run at maximize air flow. This is the case when the butterfly valve is opened all the way, termed wide open throttle (WOT). When the butterfly valve is partly opened, the flow is restricted, slowing air flow.
through the venturi and reducing the amount of fuel that is pulled into the flow. The pumping losses that SI engines experience are largely due to running at part throttle openings.

To control the air/fuel ratio in a carburetor, the fuel jets can be adjusted. These jets must be set and left alone during engine operation. Some carburetors employ more than one size jet that transition at certain air flows, allowing better fuel flow characteristics in their respective air flow ranges. Even so, carburetors often run different air/fuel ratios at different engine speeds solely based on the tuning limitations of the mechanical system.

The evolution of the mechanical carburetor is the fuel injector. Fuel injectors are typically electronically controlled with solenoid valves and usually operate with pressurized fuel. The position of the injector determines what type of fuel injection the engine uses. Port fuel injection positions the fuel injector between the butterfly valve that controls air flow (the throttle body) and the intake valve. In this way, fuel can be injected upstream of the intake valve and it has time to evaporate, making a mostly homogeneous mixture of air and fuel that enters the combustion chamber. Fuel injection allows for smaller droplet sizes and better atomization of fuel. It also allows for better control of the air/fuel ratio. This can lead to decreased emissions and improved power across the engine speed range.

Another type of fuel injection is direct fuel injection. Direct fuel injection involves the injection of fuel directly into the combustion chamber; the fuel does not mix with the air until it is actually inside the cylinder. Direct fuel injection offers some advantages over port fuel injection. The evaporation of the liquid fuel inside the
combustion chamber lowers the in-cylinder temperature and reduces knock. Also, piston
design can allow for a stratified charge in which the air/fuel mixture is richer near the
spark and leaner in the rest of the combustion chamber. It is difficult to achieve good
combustion in leaner mixtures, but if the rich mixture near the spark causes a high quality
flame front to propagate throughout the leaner parts of the mixture, a leaner overall
mixture can be used, and fuel consumption can decrease. Direct fuel injectors usually
utilize higher pressures than port fuel injection to overcome in-cylinder pressure and to
achieve small droplet sizes. Since the fuel does not have time to evaporate in the intake
tract as in port fuel injection, a small droplet size is vital to achieving evaporation of the
fuel. High pressures and extremely small orifice sizes help accomplish this. High pressure
also allows very good control of the injection events; some injectors even utilize multiple
short bursts of fuel. Port fuel injection suffers from a delay between injector inputs to
actual engine response. Some of the fuel wets the walls of the intake tract and then
evaporates and before it is drawn into the cylinder. In this way, the wetted walls act as a
damper and slow reactions of the engine. As more air is drawn into the engine, it may
take a number of cycles before the air/fuel ratio has adjusted to be correct. This means
that sudden large throttle openings, many port fuel injection systems will overfuel to
prevent leaning out of the mixture before enough fuel can evaporate from the walls of the
intake tract. Direct injection engines do not suffer from this delay. Direct injection can
also ease engine starting. Port fuel injected engines can go through up to 10 engine
revolutions before enough fuel evaporates from the wetted walls to get into the cylinder
to cause combustion (Zhao, Lai, & Harrington, 1999). This means that some fuel gets
through the exhaust valve while engine starting is being attempted, and can lead to poor
emissions characteristics for an engine during startup. Direct injection technology has been used on diesel engines longer than it has on SI engines, but the advantages of it are quickly being realized. One of the few disadvantages of direct injection has been carbon deposits. Research into the spray characteristics of direct injectors has shown fouling of the injector and spark plug, as over time the stratified charge inside the cylinder will cause deposits to slowly build (Zhao, Lai, & Harrington, 1999). This is an issue that is currently being addressed through research.

Air-assisted two-phase direct injectors utilize two solenoids and lower pressures than other direct injection systems. The first solenoid meters an amount of fuel into a pre-chamber and the second meters pressurized air into the pre-chamber with the fuel where it mixes and passes through a nozzle. Research shows these injectors produce a finely atomized spray with a narrow filled-in cone-like spray structure (Boretti, et al., 2007). More detail about air-assisted injector operation will be discussed in Chapter III.

**2.9 - Fuels**

Spark ignition engines typically run on high AKI fuels compared to compression ignition engines. Gasoline is the most common fuel with a high AKI. In the United States pump gasoline is usually sold at three AKI ratings: 87, 89, and 91-93, depending on the state. Gasoline is a mixture of hydrocarbons distilled from petroleum that consists of hydrocarbon chains in the 8 carbon range. The heat of combustion of gasoline (heat released when burned in the presence of oxygen) is approximately 20,640 Btu/lb (48 MJ/kg). It is important to note for all fuels that values are approximate. Fuels are manufactured and distilled at different locations from different materials and values are experimental or calculated based on averages. Different batches of the same fuel can have
varying average chemical formulas, densities, and energy contents. For general information on test fuels see Table 1.

Ethanol is added to pump gas in most of the United States; it is typically 10% ethanol by volume and called E10. Ethanol has a higher AKI than gasoline, but contains less energy. Some pumping stations offer E85 (85% ethanol), which has approximately 60-70% of the energy contained in gasoline, but is largely produced from domestic resources. It is sometimes used as a performance fuel due to a high AKI. For aircraft applications, where weight is of utmost importance, ethanol is usually not considered based on its low energy density. The heat of combustion of ethanol is about 12,900 Btu/lb (30 MJ/kg).

Jet Propellant 8 (JP-8) is a kerosene based fuel used to power gas turbine aircraft, tanks, ships, and ground vehicles in the Air Force. It is a hydrocarbon fuel that averages about 12 carbon atoms. It is denser than gasoline by about 15%. JP-8 has been described as having a foul smell and a greasy feel to it. JP-8 is a lower volatility fuel than gasoline, so it does not readily evaporate, for this reason it tends to build up on parts and surfaces, leaving an oily feel. The heat of combustion of JP-8 is about 18,620 Btu/lb (43.3 MJ/kg)(Stouffer, Pawlik, Justinger, Heyne, Zelina, & Ballal, 2007).

Pump diesel (D2 diesel) is used widely in ground vehicles. The trucking industry runs almost exclusively on diesel fuel. Other markets that use diesel fuel are industrial and farming equipment, large boat engines, and backup power generation. Diesel fuel can consist of anywhere from 15 to 21 carbon atoms per molecule on average (Enweremadu, Rutto, & Oladeji, 2011) and has a heat of combustion up to about 19,460 Btu/lb (45.3 MJ/kg). Diesel fuel usually costs more per gallon than gasoline, but the CI engines that
use diesel tend to be more efficient, making up for the cost. Also, diesel fuel is denser than gasoline, allowing more energy to fit into the same volume of fuel tank.

The University of Dayton Research Institute (UDRI) has grown many different strains of algae that are being used to produce algae biodiesel. Upon chemical analyses, it was found that the algae biodiesel has an oxygen group attached to its hydrocarbon chains. Testing also showed that, on average, the algae biodiesel has about 18 carbon atoms per molecule. Algae biodiesel has a density slightly higher than JP-8. Heat of combustion data was not available for this fuel at the time of testing, but it is important to note that due to the oxygen molecules in the fuel, the stoichiometric air/fuel ratio is about 12.4:1, which is about 83-86% of the stoichiometric air/fuel ratio of the other fuels. A very small quantity of algae biodiesel can be produced using the current extraction and processing techniques in the lab at UDRI. For this reason, to have enough fuel to work with and generate some useable data, the algae fuel was blended with D2 diesel. A 20% (by volume) blend was used. The reason for using a 20% blend is that some ground-based biodiesels that are available on the market for trucking companies is blending with D2 diesel in this same ratio (called B20). When blended with the diesel the stoichiometric (A/F) ratio becomes about 14:1, which is 93-97% of the (A/F)stoich ratio of the other fuels.

Camelina biodiesel comes from the Camelina Sativa plant which is a member of the mustard family. It can grow in relatively harsh conditions; withstanding cold springs and low rainfall. It is a relative of Canola as well and the meal left after oil extraction is a good source of nutrients for livestock. Camelina usually does not produce as much fuel per acre as Canola, but is more resistant to harsh conditions and uses very little moisture.
Camelina has an average chemical formula similar to JP-8, with slightly higher heat of combustion at 18,720 Btu/lb (43.5 MJ/kg). Camelina showed a slightly higher Cetane number than D2 diesel (58 versus about 40-50 for D2 diesel) (Rahmes, Kinder, & Henry, 2009). A low flash point (43 degrees C versus about 55 degrees C for D2 diesel) and lower boiling point (242 degrees C versus about 300 degrees C for D2 diesel) can aid in vaporization and ignition of the Camelina biofuel as well.

<table>
<thead>
<tr>
<th>Chemical Formula</th>
<th>Density (lb/gal)</th>
<th>(A/F)stoich</th>
<th>Heat of combustion (Btu/lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AVGAS (100LL)</td>
<td>C_{8.13}H_{15.34} 5.79 @299K</td>
<td>14.61</td>
<td>20,640</td>
</tr>
<tr>
<td>JP-8</td>
<td>C_{11.9}H_{22.8} 6.59 @299K</td>
<td>14.66</td>
<td>18,620</td>
</tr>
<tr>
<td>D2 Diesel</td>
<td>C_{14.4}H_{24.9} 7.13 @306K</td>
<td>14.38</td>
<td>19,460</td>
</tr>
<tr>
<td>Algae Biodiesel</td>
<td>C_{18}H_{33}O_{2} 6.68 @304K</td>
<td>12.39</td>
<td>-</td>
</tr>
<tr>
<td>Algae 20% Blend w/ Diesel</td>
<td>- 6.97 @304K</td>
<td>13.98</td>
<td>-</td>
</tr>
<tr>
<td>Camelina</td>
<td>C_{11.3}H_{24.5} 6.24 @306K</td>
<td>14.96</td>
<td>18,720</td>
</tr>
</tbody>
</table>

### 2.10 - Ground-Based Engines versus Aircraft Engines

When it comes to engines, there exists a different list of priorities depending on whether an engine is being considered for a ground-based vehicle or an aircraft (Rozenkranc & Ernst, 2003). Although reliability is a priority for all engines, reliability is the most important attribute for an aircraft engine. If the engine on a ground-based vehicle fails, the vehicle stops moving. This is a major inconvenience for the operator and could cost time and money, but usually does not mean loss of the entire system or injury. If the engine on an aircraft fails, the whole system could be lost, which in some cases includes a human operator. For aircraft, certification is usually necessary. Certification
ensures quality and a more solid knowledge base of field engines and failure points. Engine maturity can be a deciding factor when selecting aircraft engines. The use of newly developed engines on aircraft platforms is less common than in ground vehicles as reliability is so important. An aircraft has to be designed around a propulsion system, so mature engines are often used to minimize the risk of a redesign if failures or problems are experienced with a fledgling propulsion system. Manufacturing techniques and support are also extremely important in engine choice. Large companies with large engineering and support teams are usually preferred to smaller companies with fewer resources on hand. Many of these reasons contribute to the fact that experimental engines are more often used in ground vehicles, as the risks are less. This is not to say that UAV’s and other aircraft have not used and are not using experimental engines; there is simply more risk involved.

Some more characteristics of ideal aircraft engines are further discussed by Rozenkranc et al. Low system weight is important for aircraft propulsion systems. 0.45 hp per pound of system weight is desirable as a minimum. UAV’s especially are focused on range, or loiter time, and need low specific fuel consumptions. Less than 0.4 lb/hp/hr is desirable for spark ignition engines.

Ideally, it should be possible to cool the engine with as little drag as possible. Having smaller engine frontal areas and less cooling surface area is desirable in increasing propeller efficiency and range. Multi-fuel engines or engines that can use heavy fuels, heavy fuel engines (HFE), are preferred. Heavy fuels contain more energy per volume and are more widely available. The ability to run different fuels in the same
platform increases a UAV’s scope of use and value. This will be discussed further in the Heavy Fuels in SI Engines section of this chapter.

Turbocharging has been recognized as an important factor in maintaining engine power at higher altitudes, where air density drops. This allows for a smaller overall engine, as the engine displacement can be sized for adequate performance at sea level and the turbocharger will maintain that power. Turbocharging is usually not considered for small (<10hp) engines as turbocharging is not efficient at these sizes. Boundary layer losses inside the turbocharger at the required high impeller speeds make turbocharging less efficient. Also, most small UAV’s are not flying high altitude missions; they suffer less significant losses at <5000ft altitude.

Fuel injection aids in starting and allows for better combustion efficiency. Also, it can help reduce problems with icing. Complete electronic control and systems health monitoring is expected of modern engines. Finally, mass production and good maintainability characteristics of engine systems mean higher quality systems and less frequent downtime for the UAV systems.

Many of these qualities are also important for ground-based engines, but a heavier focus is put on engine efficiency, driving dynamics, and power and torque characteristics. Aircraft engines spend much of their time at higher throttle openings than ground-based engines, which usually cruise at smaller throttle openings and spend more time stopping and accelerating. Engines that are designed for ground uses usually don’t make excellent aircraft engines and vice-versa.
2.11 - Heavy Fuels in SI Engines

Based on the goals described in the introduction, there is a recent focus to develop lightweight IC engines that can run on heavy fuels. In the case of the Air Force, gasoline is used for many small SI UAV engines, including the Fuji 34cc. The Air Force uses JP-8 for everything from trucks to turboprop aircraft, fighter jets, and large turbofan aircraft. The only vehicles that don’t use these heavy fuels are piston powered UAV’s. To remedy this, research is being conducted to allow these platforms to use JP-8, without having to switch engine platforms. To accomplish this, some modifications usually have to be made. Although it is possible to run heavy fuels and JP-8 through a carbureted system, fuel injection is ideal. Carburetors are designed for a certain fuel, and when other fuels are used in them, they tend to behave differently based on the fuels’ different flow characteristics, i.e. density and viscosity. The jets in most small SI engine carburetors are designed for gasoline. If a permanent switch to a heavy fuel was planned with a carburetor, different fuel jets might need to be sized.

With the advent of modern fuel injection systems, some small SI engines have been converted to use fuel injection. Fuel injection offers the ability to gain tighter control on injection rates and therefore maintain desired equivalence ratios more consistently. This can have performance and emissions benefits. Also, with heavy fuels, fuel injectors can provide smaller droplet sizes, aiding in the evaporation process. Since air-assisted injectors operate at lower pressures than other injectors, they were focused on for this research. The pre-mixing of fuel with air in the injector prechamber, coupled with the pressurized discharge of the locally rich mixture results in small droplet sizes.
(Geoffrey Cathcart, 2006). The particular type of air-assisted injector used for this research has been seen to achieve droplet sizes as small as 5-6 microns.

Experimental results of SI engine heavy fuel research show decreased full load torque compared to baseline avgas for JP5 heavy jet fuel. At the lowest and highest engine speeds, torque loss is less than at middle engine speeds. Higher BMEP values are seen at mid engine speeds, causing higher propensity to knock and spark timing adjustments. Operating at higher engine speeds can reduce knock as there is less time for knock to develop per cycle (Cathcart, Dickson, Tubb, & Schmidt).

2.12 - Cold Starting Heavy Fuels

Cold starting heavy fuels can prove to be quite difficult. Heavy fuels have a lower vapor pressure than gasoline and avgas. Because of this, getting them to evaporate into a combustible gas at ambient temperatures is quite difficult. Many methods have been tried to facilitate cold starting of JP-8 in small SI engines. Research has shown that it is difficult to start a cold SI engine on kerosene (similar properties to JP-8) even with small droplet sizes, without adding some energy in the form of heat to the system (Suhy, Evers, Morgan, & Wank, 1991). Suhy et al. used a vortex pneumatic atomizer to create a similar effect to an air-assisted injector. The vortex pneumatic atomizer was a prechamber device that used compressed air and an orifice array to further atomize fuel from the fuel injector before it entered the intake tract. To facilitate cold starting, Suhy et al. supplied pre-heated air to the vortex pneumatic atomizer to add enough energy to the kerosene to cause vaporization around the spark plug. Only approximately 115°C (239°F) was needed to start, which is less than is needed to cause vaporization of kerosene at ambient conditions. This lower temperature was attributed to further energy being added in the
compression process and some of this energy being stored in the intake manifold and engine. After a sufficient number of revolutions, enough energy was stored to cause vaporization. Their cold start procedure involved heating the atomizer air and slowly bringing up injector pulse width until starting occurred. This limited the chance of liquid fuel bridging the spark plug electrode gap and stopping a spark from happening.

Other methods for cold starting SI engines on heavy fuels focus on pre-heating the liquid fuel, but this can lead to coking of the fuel inside the heating circuit. Intake air can be pre-heated to aid in cold starting, but a relatively large and heavy intake air heat exchanger is needed. Some systems utilize a glow plug to vaporize heavy fuel for starting and then switched off (Hynes, 2004). All of these methods require the addition of heating hardware, which is a problem for small UAS’s which cannot spare extra weight and complexity.

High-power, high-energy spark ignition systems have been used to help with cold start in compression ignition engines, but problems were encountered with spark plug fouling. The high voltage and power required by the system ultimately made the ignition environment too harsh and the benefits small (Ward, 1990).

Another cold-start method used by Sonex Research in Maryland is the use of a battery-powered fuel vaporizer to start an engine that is specified to use kerosene only (Charles C. Failla, 1991). This unit runs just long enough to get the engine started and is then disengaged. The extra weight of such a unit would be a problem for UAS use. Another method for cold start mentioned in this research is the combination of direct injection and a hotter, high energy spark plug. This combination is only good for cold-
start though, as a hot spark plug could encourage engine knock as operating temperatures are reached.

The simplest method for achieving cold start is to use a dual-fuel system in which the engine is started on a more volatile starter fuel and then switched to heavy fuel for normal operation. This is not considered acceptable for many applications as a dual-fuel system would still require the logistics of stocking the starting fuel.

Cold starting heavy fuel SI engines simply and reliably is an ongoing issue that is getting significant research attention. As heavy fuel SI engine research has shown that decreased BSFC can be achieved with heavy fuels, especially at partial cruise loads, it is expected that more research will be conducted to solve this hurdle and allow for the reliable and consistent use of a single heavy fuel for UAS’s.
CHAPTER 3 – EXPERIMENTAL SETUP

The test stand was operated in the Small Engine Research Lab (SERL) test cell that incorporates fire suppression, room ventilation, CO detection, and a separate fuel storage room. This test stand was constructed on a steel, t-slot tabletop and an extruded aluminum frame. Engine mounting plates were designed in-house. The engine was mounted from the bottom of the oil sump. This allowed access to both sides of the engine crankshaft.

The dynamometer used for this testing was a Magtrol ED-715-6N hysteresis type dynamometer (Figure 7). Maximum torque for this dynamometer was 55 lbf-in (4.6 lbf-ft). The maximum continuous power was 4.0 hp and maximum peak power was 4.6 hp. Maximum rotor speed was 25,000 rpm. The maximum engine speed that was used in this research was 7500 rpm. A hysteresis dynamometer is a type of electromagnetic brake similar to an eddy current dynamometer. It has a steel rotor/shaft assembly that is spun by the engine. The steel rotor/shaft moves through an electromagnetic field generated by a coil around the rotor, but there is no physical contact. The dyno brakes the rotor by varying the magnetic field in the air gap between the rotor and coil and can provide braking force without any rotor speed (from 0 rpm). The dyno was air-cooled by an electric fan unit that piped air through the dyno rotor housing.
To couple the engine to the dynamometer, multiple coupling designs were tried. Since the engine spins a relatively high rpm and is a single cylinder four-stroke, couplers had to be able to handle a lot of unsteady torque and vibrations. Single cylinder engines, especially four-stroke single cylinder engines, generate heavy vibrations. This is because the rotating mass of one cylinder is not cancelled out by the rotating mass of other cylinders. The power stroke of the engine is only every other revolution, and it causes far more force than the other three strokes. For this reason, single cylinder engines tend to generate large amounts of vibrations that can be troublesome to the other hardware in an experimental setup. Different types of rubber couplers and bearing setups were tried, but many equipment failures occurred. Polymeric spider-type couplers that were rated for these engine speeds tended to not be able to handle the unsteady torque and larger rubber couplers that were designed to handle the torque usually were not rated for these rotational speeds. Eventually a Lovejoy rubber coupler was found (Lovejoy LF2) that was rated for these speeds (Figure 8).
Magtrol recommended the use of a sacrificial bearing to protect the rotor bearings in the dyno in the event of a coupler failure. A pillowblock bearing setup was designed (Figure 8) and an alignment tool was developed. Alignment was accomplished using a magnetic proximity sensor, a v-block and clamps, and a LabView program. The v-block and proximity sensor were mounted on the crankshaft of the engine and the whole assembly was spun around the input shaft to the dyno and distances were observed and recorded through the LabView program. Then, shims and adjustments were used to make the distances from the proximity sensor to the input shaft of the dyno equal on the top, bottom, left, and right sides of the shaft. This was a tedious process, but worked well enough to get shaft alignment within 2-3 thousands of an inch on all sides. The single bearing setup’s biggest problem is that certain engine speeds will cause a resonance in the shaft that started a wobble on the rubber coupler nearest the engine. This wobble caused a lot of vibration and was most likely the cause of most of the coupler failures experienced. In the future a two bearing setup will be used between the dyno and engine to eliminate this wobble. Figure 9 shows a screenshot of highspeed video that was taken to observe and verify the existence of coupler wobble with one of the older, smaller couplers installed. Coupler wobble stopped some data points from being taken and was worsened.
when injection timing problems were going through the trouble-shooting process. In the end, a majority of data points were able to be captured.

![High speed video screenshot of coupler wobble](image)

*Figure 9: High speed video screenshot of coupler wobble*

The Fuji engine had a wet sump oiling system, which means the oil drains down to a section under the crankcase before being pumped up around the engine. The stock setup had an external line that ran from the side of the crankcase, up to the carburetor top, and to the valve cover. This line was used to drive the diaphragm fuel pump in the carburetor. There was also a tee-d off line that was open to atmospheric pressure with an orifice insert. During testing, large amounts of oil were being lost out this line. This could have been due to oil splashing from the engine being hard mounted to the table. In a UAV the engine would be soft mounted on floating dampers, reducing oil splashing. To remedy this oil loss, the oil lines were re-routed based on the method used by the Army (Small Heavy Fuel Engine Research Team). A hole was drilled and tapped into the sump and a fitting installed, then the oil line was routed up to the valve cover. Next, a new atmospheric line was drilled and tapped into a separate part of the crankcase that
experienced much less oil splashing. A catch can was installed on the table at the end of this line to catch any oil that might have still made it through (Figure 10).

Figure 10: Oil line re-routing and catch can

The fuel system for the test stand consisted of a fuel tank (Figure 11), which was mounted higher than the test stand to allow gravity draining of the system, manual shutoff valves, two remote pneumatic shutoff valves controlled by air solenoids, ¼” stainless steel tubing, two fuel filters, a flowmeter, flowmeter bypass circuit, a nitrogen supply to pressure the system, and a three-way valve to let the operator choose between test fuels. A small separate tank (also under pressure) was used to hold starting avgas. A nitrogen pressure regulator was used to choose fuel pressures. The nitrogen tank was located outside the building. Fuel pressures were chosen based on fuel injector flow characterization tests (discussed later).
The fuel flowmeter selected for this testing was a Max Machinery 213 Positive Displacement Flowmeter (Figure 12). As fuel flows through the meter, pistons fill and empty, turning a small crankshaft. A known volume fills each piston stroke and crank rpm gives volumetric flow readings.

To remotely start the engine, a geared starting system was purchased. The system is designed for small RC engines. An adapter plate had to be designed for the gear assembly to mount to. This was designed and machined in-house (Figure 13).

A solid coupler was designed to couple the back side of the crankshaft to the starter assemble. The starter utilized a one-way needle bearing to allow it to power the engine electrically for starting, and then free-wheel as soon as engine rpm overcame the
started rpm. In this way, the only engine drag added to the system was the power required to free-wheel the crankshaft in the needle-bearing: an amount deemed to be negligible.

The geared starter assembly utilized a high rpm electric starter motor, geared down twice to produce sufficient starting torque. This gear reduction also limited actual starting rotational speed to about 600 rpm. This was sufficient for starting, but usually took longer than desired. Less gear reduction or more starter torque would be desired.

![Figure 13: FEMA starting system](image)

Later in testing, failures were experienced with the remote starting assemble due to the aluminum gears stripping some teeth. One of the pull start assemblies from the Subaru-Robin engine kits was also modified to fit the engine. In this way, the engine could be pull-started by one of the operators in the lab. This system worked well, as the operator was able to induce plenty of rotational speed to get good compression on the engine, even if pull-starting was less than ideal (Figure 14).
Air flow measurements were taken using a TSI 4021 hot wire type mass air flowmeter (Figure 15). This type of flowmeter works by passing electricity through a wire in the flow cross section, as air flow rates change, the sensor increases or decreases the current flowing through the wire to maintain a specific temperature based on a change in resistance in the wire. More air will increase heat transfer from the small wire and cause more current to flow. This current flow is translated to a voltage output signal from 0 to 5 volts.

Single cylinder engines, along with producing a lot of vibration, also produce a lot of unsteady airflow. The reason for this is that the engine is only taking air in during the intake stroke. This means that during the other three strokes (compression, expansion, and exhaust) the flowing air must stop against the closed intake valve rather abruptly. This causes air flow measurements to oscillate so much that it can be difficult to use the data. Ferguson suggests the use of an air damper to smooth the air flow. An air damper (also called a surge tank) that is at least 250 times the volumetric capacity of the engine
displacement is recommended. Initially, a 17.45 gallon plastic tank was used as an air damper. This damper was approximately 2000 times the displacement of the engine, which is far beyond the 250 times called out in the reference. This plastic box was lined with polyurethane foam and sectioned into “rooms” to damp out oscillations as air passed through the pours of the foam and also the rooms of the box. Air flow tests still showed significant air flow oscillations in the high speed data. So, a 55 gallon polyethylene drum was sourced as an air damper and operated without foam. This is approximately 6200 times the volume of the Fuji engine. Some flow oscillations were still seen in highspeed data. Polyurethane foam was placed inside the drum; around the perimeter of the drum and with two separate “walls” of foam that air had to pass through (Figure 16).

Figure 16: Air damper drum with polyurethane foam inserts

Figure 17: Air damper with foam sections
This highly porous foam absorbed some of the energy of the oscillations, as small movements in the pores of the foam absorbed oscillation energy. These walls are somewhat loose in the drum as they were able to flex back and forth slightly at the center point, further absorbing flow oscillation energy. After the foam was installed in the drum, the flow oscillations were smoothed (Figure 18).

The air-assisted injector used for this engine testing was an Orbital injector sourced from a 50cc Aprilia two-stroke direct injected scooter. Aprilia has solved many of the emissions issues associated with two-stroke engines by utilizing direct injection. Two-stroke engines send a lot of unburned hydrocarbons out the exhaust because both the intake and exhaust ports are open for a short time near BDC. By using direct injection from the cylinder head, the intake air contains no fuel, so any intake air that makes it out the open exhaust port causes no extra emissions. Fuel injection does not start until the ports are closed and compression has started.

The Orbital air-assisted injector uses two separate injection events to meter fuel and air into the engine (Figure 19). The two injection events are separated by a set amount of time and there is no overlap. The fuel-metering solenoid is triggered by an electrical signal, allowing pressurized fuel to enter a pre-chamber, before it closes. Then
after the separation time has passed, the fuel/air solenoid opens, allowing pressurized air to enter the pre-chamber with the already metered fuel. The pressurized air and fuel then exit the injector nozzle together as a rich air/fuel mixture. As the air and fuel leave the nozzle, the fuel is separated into small droplets. The air-assisted injector used shop air and pressurized fuel from the fuel system. The shop air was regulated using a standard shop air regulator and had a check valve installed to prevent any fuel from backing up through the system. Air pressure used for this testing was 75 psi. If fuel pressure was left on and the air pressure was removed, fuel would flow freely though the injector. The air/fuel prechamber needing air pressure on it to remain closed. To purge one fuel from the injector to make way for another fuel, this injector characteristic was utilized. Air pressure would be removed from the line and bled off, allowing the new fuel to flow and “clean out” the injector, and then air pressure would be restored, stopping flow.

Figure 19: Injector mounted on engine

To gain a working knowledge of the injector’s signals, the Aprilia scooter was connected to high speed data acquisition equipment and run at different engine speeds (Figure 20).
Figure 20: Example of scooter injector signals

The fuel metering solenoid is shown in red and the air and fuel injection solenoid is shown in blue. The plot is inverted from what would normally be seen, but the peak signal and hold (shown here as an oscillating signal or duty cycle) signal for each part of the injection can be seen. When the injector closes there is voltage in the opposite direction as the solenoid slams shut before it returns to no voltage.

Solid-state relays were chosen to control the air-assisted injector because they have faster reaction times than mechanical relays. Four Crydom D1D12 solid-state relays (capable of 12 amps) were used to control injector signals (Figure 21).

Figure 21: Solid-state injector relays
With overhead valves, it was relatively straightforward to place a magnetic proximity sensor into the valve cover to pick up the intake valve (Figure 22). This phasing sensor would allow the control program to know when the engine was on the intake stroke. It was difficult to find the exact proximity to the valve that would provide reliable signals (by moving the sensor in and out of the threads tapped into the valve cover). Spark plug noise was seen in the phasing sensor that caused many bad signal reads and misfires. These misfires were very harsh on the engine couplers. At one point, the phasing sensor was abandoned for a simplified control scheme, in which the injector simply fired once every other revolution, even if it was out of phase.

![Injector phasing sensor](image)

The stock engine came with a crankshaft position sensor (CPS). The CPS allowed the stock engine to know when the magnet on the crankshaft was passing the CPS (Figure 23). The spark signal for the stock engine was fired using this signal; a spark was delivered once per revolution. The spark that happened before TDC on the exhaust stroke served no function. On the research engine, the CPS was used for controlling spark timing and for injector sequencing and was relocated to the side of the crankshaft.

An encoder was designed and machined in-house that was used to give the control program finer time increments of crankshaft rotation. The encoder was made of
aluminum and had 36 teeth (Figure 23). A stationary magnetic sensor picked up the teeth as they passed it. Since there were a known number of encoder teeth, the control program could also count crank rotations. For the research engine, the encoder, CPS, and (originally) the phasing sensor all worked together to control spark and injector signals.

Figure 23: Encoder, encoder pickup, and CPS

The injector was mounted to the engine using an in-house designed injector manifold (Figure 24). The Orbital injector was rather large, making it difficult to package it in an efficient manner (some that would need to be addressed if ever a flight demonstrator was to be developed). For the purposes of this testing, the injector was mounted so that it was positioned pointed straight at the intake opening on the side of the engine.

Figure 24: Aluminum injector manifold
The air flow path was such that it had to make a 90 degree turn once it entered the top of the manifold through the throttle body butterfly, which was the stock carburetor, mounted on its side on the top of the manifold. Two iterations of the injector manifold were designed and machined. The bend in the manifold airflow path was likely a cause of loss of volumetric efficiency in engine performance over the stock carburetor setup. The injector manifold had threaded mounting holes for the injector, the carburetor throttle body, the throttle control hardware, and the manifold absolute pressure (MAP) line. The MAP line was tapped for a pipe thread, and was simply a hole drilled into the manifold block. A fitting was used to connect a 1/8” stainless steel line leading to a pressure transducer so manifold pressure could be recorded.

For testing, a low volumetric efficiency was observed while running on the air-assisted injector compared to volumetric efficiency data from the engine while running a carburetor. This was because air was being injected in sufficient quantities to require less air to be taken in through the MAF sensor on the air damper drum. Rather than trying to add a flowmeter on the pressurized airline (unsteady flow in the pressurized air) testing was completed to determine how much air the injector flowed at different engine speeds. This testing was completed with no liquid fuel flowing. To test this flow a backing plate was designed for the injector manifold to stop air from flowing through the manifold hole that normally mounts up to the engine and the throttle butterfly was opened to 100%. The MAF sensor was installed backwards in the air damper so it could read airflow coming out of the air damper in the reverse direction it normally flowed. The air damper was still used so that injector air flow would be steady. Air flow data was recorded and plotted. A linear equation was then assigned to the air flow data.
This air flow data did not take into account liquid fuel flow. To compensate for liquid fuel flow, it was assumed that total mass flow would be conserved and a correction equation was developed that took into account the density of the liquid fuel and the density of the air (Equation 2). This correction equation was applied to all air-assisted injector data to produce a more accurate air flow and therefore a more accurate equivalence ratio.

\[ \dot{V}_{\text{tot}} = \dot{V}_{\text{mat}} + (4.3197 \times \text{rpm} + 1631.7) - \dot{V}_{\text{fuel}} \left( \frac{p_{\text{fuel}}}{p_{\text{air}}} \right) \]  

(Equation 2)

To characterize the injector, injector flow rates were measured with both avgas and JP-8 (as a representative heavy fuel) and droplet sizes were captured.

To measure flow rates a small test stand was set up. This test stand included two tanks: compressed nitrogen to pressurize the test fuel tank and compressed air for the air injector supply. The injector was mounted on a stand and plumbed to the compressed bottles for fuel and air. The Max Machinery fuel flowmeter from the engine test stand was borrowed to allow measurement of fuel flow. To accomplish this, since data acquisition was not present at this separate test stand, a pulse counter was used. A Berkeley signal generator was used to control the injector relays. Based on the signals from the scooter test, peak times for both solenoids were set as constants, and the only signal that was varied was the fuel hold time. Engine rpm was set as a constant as well. As fuel hold times were varied from 0 to 20 ms, fuel flow rates were recorded from the pulse counter on the fuel flowmeter. Figure 25 shows the results of fuel flow measurements. Note that only fuel hold pulse width up to 8 ms are shown, since 8 ms is the maximum fuel hold pulse width utilized in testing. Flow rates were recorded at 80 psi, 90 psi, 100 psi, and 115 psi. It is interesting to note the difference in flow rates at the
lowest pressure (80 psi) between avgas and JP-8. This difference could be due to the different densities or viscosities of the fuels. During testing, the fuel flow rates will be recorded using the fuel flowmeter, so this difference won’t affect results; it will only cause different fuel hold pulse widths to be used for different fuels. A fuel pressure of 85 psi was used in testing based on the required fuel flowrates seen from carbureted engine testing.

![Injector Performance](image)

**Figure 25: Orbital fuel injector flow rates**

Literature on droplet sizes typical for carburetors is limited (Heywood, 1988). Heywood quotes droplet sizes in the 25 to 100 micron ranges, with larger droplets also produced. Heywood goes on to point out that droplets of more than 100 microns will not likely vaporize completely in an intake tract at any engine speed. With small, lower tech carburetors such as the unit used for this application, even larger droplet sizes are expected (>100 microns). If a droplet does not vaporize, it will likely impact an intake tract wall, especially if there are bends in the tract, where it will wait for a number of
engine cycles before it evaporates. All of these large droplets and wall wetting effects lead to poor combustion efficiency and transient response of the engine.

To measure droplet sizes, a Malvern Series 2600 droplet size analyzer was used. The Malvern measured droplet sizes using laser diffraction. The Malvern incorporated two units with a laser beam between them. The injector sprayed through the laser beam and the included software read out droplet sizes using the Sauter Mean Diameter (SMD).

\[
\text{SMD} = D[3, 2] = \frac{d_v^3}{d_s^2} = 6 \frac{V_p}{A_p}
\]  

Where:
- \( d_v \) = volume diameter for the whole spray
- \( d_s \) = surface diameter for the whole spray
- \( V_p \) = particle volume
- \( A_p \) = particle surface area

Results for injector droplet sizes showed slightly higher droplet sizes than reported in the literature. Droplet sizes reported in literature with the Orbital air-assisted injector are in the 5.0 micron SMD range (Cathcart, Dickson, Tubb, & Schmidt). This could be due to different test conditions (test conditions of literature results were unknown) or different test equipment. Tests still showed results in the sub-15 micron range, which was considered a large improvement in droplet size over what typical carburetors can produce. Cathcart et al. measured smaller droplet sizes for gasoline than
for heavy fuels, which is the opposite of what is shown in the figure. This could again be due to differing test conditions.

![Droplet Sizes](image)

**Figure 27: Droplet size results for air-assisted injector**

Throttle control was originally achieved through a manual throttle control bracket with a lever that could be set to the desired throttle position and clamped into place using a wing nut. This manual bracket employed a backing plate with a lever arm that was connected to the throttle linkage (Figure 28).

![Manual throttle control bracket](image)

**Figure 28: Manual throttle control bracket**

To establish remote control, the use of a RC industries sourced servo (Figure 29). This type of servo uses a pulse width modulation (PWM) control scheme. A 5 volt square wave is sent every 20 ms. Pulse width is modulated such that a 1 ms pulse corresponded
to a servo location of full extent in the counterclockwise direction, while a 2 ms pulse corresponded to a servo position of full extent in the clockwise direction. A 1.5 ms pulse corresponded to a neutral position. In terms of throttle control, a bracket and linkage were designed so the servo could control throttle position. The control circuit for the servo was set up so that 0 volts corresponded to 0% throttle, and 5 volts corresponded to full servo extent. By adjusting the maximum voltage the servo could be calibrated to allow 0-100% throttle butterfly angle.

Figure 29: Throttle control servo on carbureted setup and injector manifold

Considerable servo hysteresis was observed on the original cart setup (Figure 30). The blue line on the plot represents the throttle position when the control program was told to step up from 0% to 100% and the red line represents the throttle position stepping down from 100% to 0%. The hysteresis in throttle position was enough to warrant the implementation of a throttle position sensor so that the servo would not have to be relied upon for real throttle position feedback.
The throttle position sensor utilized for this testing was sourced from the small motorcycle market. It was a CTS Automotive Products 500 Series Single Ear Rotary Position Sensor (Figure 31). It gave position feedback as a percentage of excitation voltage (5 volts). This feedback voltage was translated to throttle position percentage by the control program. The addition of a throttle position sensor removed throttle hysteresis from the system, which can be seen in Figure 31.

During cold weather testing, an inlet air heater was used to bring inlet air temperatures closer to room temperature (approx. 75°F or 24°C). The test cell used outside air to ventilate via three large electric fans, and this air was quite cold in the
winter. Engine starting was difficult with such cold intake air; a heater was developed using two 750W electric strip heaters. The inlet air heater was used in an attempt to aid in cold-starting of JP-8 fuel without success. Engine knock was intense with high levels of intake air heating and cold-start was not achieved on heavy fuel with the heater.

![Inlet air heater for cold weather testing](image)

*Figure 32: Inlet air heater for cold weather testing*

To generate an engine map on a specific fuel, data points were taken for a range of throttle positions at one specific engine speed, then the engine speed was changed and all of the throttle positions were swept through again. Throttle position was adjusted 10% at a time. For some of the engine testing, it was not feasible to generate an entire engine map. This was because of lack of fuel, or lack of time. The engine run time needed to generate an entire map was on the order of 2-3 hours. With the coupler arrangement experiencing failures as often as every 30-60 minutes of run time, it was not feasible to take entire engine maps for every fuel. To minimize the number of data points required, a system of useful data points were needed. The Fuji engine uses a 17 inch propeller in the field. More specifically, it is a 17x10 inch propeller, which means it has a 17 inch diameter and a 10 inch pitch. This pitch is the number of forward inches the propeller would travel during one revolution if no slip through the air was achieved. Other
propeller’s that were being assessed by other research at the time of this research included an 18x12 inch prop and a 22x10 inch prop. For this testing, the 17x10 inch prop and 18x12 inch prop were used to generate load points that were or could be used in the field.

These load points were generated by using the actual cruise speeds that the UAV platform would see while in use and backing out thrusts for those cruise speeds with these propellers. Using those thrusts, torque values and engine speeds to create those thrusts were calculated. These engine speeds, thrust values, and torque values can be seen in Table 2. Takeoff points are shown highlighted in blue.

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Data acquisition for this research was accomplished with a National Instruments Fieldpoint. Signals were connected to the NI Fieldpoint’s different modules. An Ethernet cable connected the lab computer to the Fieldpoint. The Fieldpoint was located on the test stand with the engine, while the computer was located outside of the test cell and the Ethernet cable was run through the wall. LabView software was written in-house to interface with the Fieldpoint. The LabView software changed regularly as it was updated with new sensors and features. A recent screenshot of the LabView user interface can be
seen in Figure 33. The LabView gives the user control over the throttle position, injector timing, spark timing, sample rate, and data averaging times. Also, buttons controlled PIC chip power, cooling air, and ignition power.

![Small engine LabView control program](image)

**Figure 33: Small engine LabView control program**

PIC18 F452 microprocessor chips were used to control spark timing, servo position, and signals. There were two chips: the first for spark timing and servo control, the second for injector relay control.
Emissions data were gathered using an Enerac 700 portable emissions characterization unit. Engine exhaust was collected from the engine muffler by welding a stainless steel fitting onto the muffler housing. The pickup was positioned so that it protrudes into the muffler cavity and samples exhaust gases from the middle of the center cavity, away from the muffler exit to minimize ingestion of any ambient air that might backflow between pulses into the muffler (Figure 35).

During some heavy fuel testing, particularly while attempting cold-start with JP-8, liquid fuel was being ingested by the emissions pickup. This was bad for some of the sensors in the unit and caused them to need repair when the unit was sent in for calibration. A small oil trap was installed before the unit’s water separation filter to catch any large droplets of liquid fuel that might make it to the unit. The flow had to make a
turn in a tee fitting, causing larger droplets with more momentum to become trapped in the clear tubing (Figure 36). In later testing, water built up in the oil trap that smelled of fuel, but seemed to still contain mostly water. No further testing was completed on this water. The cylindrical filter before the water separator on the unit did tend to turn a dark color, but this was deemed normal upon consulting the manufacturer.

![Figure 36: Oil trap and exhaust sample flowpath](image)

Emissions data were gathered using the Enerac’s provided software. Sample rate and averaging period were programmable. 10 samples were taken at steady state conditions and averaged later. Data was gathered for carbon dioxide (CO2), carbon monoxide (CO), oxides of nitrogen, NOx, (NO and NO2) and unburned hydrocarbons (UHC’s). This data was saved to a text file by the program and imported into Excel by the operator.

For this testing, engine knock was detected by monitoring BSFC and audibly. During testing there was an operator inside the test cell wearing safety glasses and hearing protection (with radio communication to the other operator) and there was an operator running the LabView program to control the engine and take data. The operator in the test cell was responsible for starting the engine, setting up the fuel system valves, pressurizing the fuel, watching for any failures, observing engine running characteristics
(roughness, coupler wobbles), and listening for engine knock. Knock could be heard quite easily when wearing hear protection. Since most of the loud distortion of the engine noise was cancelled out by the hearing protection, subtle changes in the engine could be heard. When knock started to occur, a slight ticking noise could be heard in the engine this would start softly and increase with increasing knock intensity. The test cell operator could notify the control room operator when knock was observed. Also, the control room operator was watching and recording BSFC values. When BSFC values increased unexpectedly, it was likely that knock was starting to set in. The combination of increased BSFC and audible cues from the test cell operator were considered to be satisfactory for the scope of this testing. More quantifiable methods will be employed in the future, such as an in-cylinder pressure transducer with a band-pass filter.
CHAPTER 4 – EXPERIMENTAL RESULTS

4.1 - Performance Results

Once the test stand and equipment were operational, experimentation commenced. With the carbureted engine, an avgas baseline for performance and emissions was recorded. This was followed by JP-8 and Camelina biofuel. The engine hardware was then changed to the air-assisted injector setup and avgas, JP-8, D2 diesel, the algae/D2 diesel blend, and finally Camelina were tested for performance and emissions. There was considerable time (about 6 months) between the collecting of carbureted data and air-assisted injector data due to design, build, and test time for the injection system.

Cold-starting the engine with both the carburetor and air-assisted injector were attempted. Cold-start was not achieved with either method. For testing, the engine was started on gasoline and switched to the test fuel using a three-way valve. Then, after a calculated time to consume the starting fuel in the injector and short fuel line before the three-way valve, data was collected for the test fuel.

Carbureted data is valuable to analyze because it demonstrates the difficulty of maintaining a consistent equivalence ratio. The only way to run a carburetor in the field is to set the needle jets to a position that will develop an acceptable equivalence ratio to run well. In most cases, this will be in the more forgiving rich range (equivalence ratio greater than 1). Once the jets are set, they cannot be adjusted while the UAV is flying, as
the operator is separate from the UAV. It is more important to pick a carburetor setting that will be reliable than it is to pick a carburetor setting that is fuel efficient. Retrieving equipment is of the utmost priority because of sensor cost. For all carbureted testing, the carburetor was adjusted to the manufacturers recommended position of 1.75 turns out for the high speed needle and 1.33 turns out for the low speed needle.

Figure 37: Propeller load points and carbureted actual load points achieved

Figure 37 shows the ability of the engine to achieve the specified propeller load points. Load points are shown using engine speed and BMEP. The lined red points show the theoretical load points based on fielded cruise speeds, and the other points show actual results achieved through experimentation. This data was taken for a carbureted engine, and was completed to show that the experimental fuels could achieve the required BMEP’s to function at the cruise speeds selected.
In many cases, take off propeller load points were not recorded because of high levels of knock seen at those loadings. This is not to say that the engine would necessarily not be able to take off. For these takeoff points, maximum achievable torque before heavy knock was recorded.

![Figure 38: Carbureted BSFC and phi versus engine speed for 17x10 and 18x12 props (respectively)](image)

Figure 38 shows BSFC (left axis) and equivalence ratio (right axis) for the three test fuels used during carbureted testing. Here, “Bio” was the notation used for Camelina, as it was the only biofuel being tested at the time. BSFC was lowest for JP-8 at almost all engine speeds. Camelina showed intermediate BSFC compared to the other two. It was interesting to see the heavy fuels maintain a lower BSFC than the stock fuel. Furthermore, the equivalence ratios show that for these carburetor settings, avgas was usually quite rich (phi greater than 1). This is all promising data for heavy fuels. Perhaps the most important takeaway from these plots is the fact that an equivalence ratio of 1
was not able to be maintained while using a carburetor. A primary goal when developing a fuel injection system is to overcome the inability to maintain equivalence ratio.

Figure 39 shows the difference in equivalence ratios for the original carbureted setup with stock needle settings when running avgas compared to the new injection system. Equivalence ratios are much closer to 1.0 across all the load points due to the enhanced ability of the fuel injection system to meter different amounts of fuel at different engine speeds.

Figure 39: Equivalence Ratio Carbureted versus PFI (Avgas)

Figure 40 shows BSFC values for all of the different fuels with avgas and JP-8 from the carbureted data set added in for comparison. The lowering of BSFC between the carbureted data and PFI data is evident. This could be due to a number of factors. First, equivalence ratio was lower (close to 1), which meant most of the fuel was being burned and turned into heat, rather than just leaving as unburned hydrocarbons. Also, better atomization of the fuel could have been aiding in lower BSFC. Looking at the other fuels
running on PFI, Camelina showed improvement compared to carbureted data (refer back to Figure 38), where its lowest BSFC was approximately 1.0 lb/hr/hp. With PFI, Camelina showed BSFC less than 0.8 lb/hr/hp. D2 diesel and JP-8 showed similar BSFC levels across the engine speed range with the exception being the lower engine speed load points for the 18x12 prop where higher BSFC was observed for the D2 diesel compared to JP-8. The algae blend showed poor (as high as the stock carbureted engine BSFC at some points) performance until the higher engine speed load points where it came on par with most of the other fuels in the 0.8 to 1.0 lb/hr/hp range.

![Figure 40: 17x10 injected BSFC versus engine speed for all test fuels + carbureted avgas and JP-8](image-url)
Figure 41: 18x12 injected BSFC versus engine speed for all test fuels + carbureted avgas and JP-8

Figure 41 shows similar results to Figure 40. The 18x12 propeller spins slower to attain the same thrust level, also the more aggressive pitch “grabs” more air per revolution. This means the same torque will be seen at lower rpm. Higher engine loading will be experienced when using this prop.

Contour plots were created using Matlab code. This code interpolated between data points to create smoother lines. The contours show BSFC (this time presented versus torque) and overload are the two propeller load lines that were used. Data is presented for avgas and JP-8, since comprehensive 10% throttle increment maps were made for these two fuels.

Figure 42 shows BSFC contour plots generated using carburetion. The best BSFC’s are located between approximately 4000 and 6000 rpm at the highest torque
values. At the lowest torque values, BSFC is as much as 2 lb/hr/hp, and this BSFC range stretches to the higher rpm range. The load lines are placed on the contour plots to show what BSFC would be expected when running a carbureted version of this engine at the cruise speeds for these propellers.

Figure 42: Carbureted avgas BSFC versus torque versus RPM contour plot with propeller load lines

Figure 43: Injected avgas BSFC versus torque versus RPM contour plot with propeller load lines
Figure 44: Injected JP-8 BSFC versus torque versus RPM contour plot with propeller load lines

Figure 43 and Figure 44 show BSFC contour plots for the injected version of the engine for avgas and JP-8. When comparing the contour plots in Figure 42 and Figure 43 (for avgas), one can see that the engine spent more time (for the load lines) in the 0.9-1.3 lb/hr/hp range of BSFC for the injected engine to the carbureted engine’s 1.1-1.5 lb/hr/hp range.

An attempt at making a map of JP-8 with stock spark timing was made, but most of the map was knock-limited based on the highly advanced stock spark box. Figure 45 shows knock detected using an in-cylinder Optrand pressure transducer. This particular screenshot was taken for a load point from an even larger 22x10 inch prop, as a clear demonstration of engine knock. The red line shows unstable combustion using JP-8 and the blue line is stable combustion using iso-octane. The unstable combustion is knock, instead of pre-ignition, because it occurs after the spark event. Refer back to Figure 5 in
Chapter II for a picture of actual piston damage done to the Fuji engine as a result of prolonged engine knock.

Figure 45: Example of engine knock

Table 3 shows the BSFC difference (% change) when comparing the performance of the test fuels to the performance of the baseline fuel (avgas) using a carburetor. This table presents an easy way to compare the performances of the fuel quantitatively.
Table 3: BSFC % change versus carbureted avgas

<table>
<thead>
<tr>
<th>Cruise Velocity</th>
<th>JP-8 Carb</th>
<th>AVGAS PFI</th>
<th>JP8 PFI</th>
<th>D2 PFI</th>
<th>Algae PFI</th>
<th>Camelina Carb</th>
<th>Camelina PFI</th>
</tr>
</thead>
<tbody>
<tr>
<td>17x10</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>-24%</td>
<td>-32%</td>
<td>-31%</td>
<td>-29%</td>
<td>-17%</td>
<td>-8%</td>
<td>-47%</td>
</tr>
<tr>
<td>2</td>
<td>-12%</td>
<td>-25%</td>
<td>-18%</td>
<td>-24%</td>
<td>10%</td>
<td>-4%</td>
<td>-42%</td>
</tr>
<tr>
<td>3</td>
<td>-5%</td>
<td>-10%</td>
<td>-18%</td>
<td>-13%</td>
<td>Not Taken</td>
<td>-10%</td>
<td>Not Taken</td>
</tr>
<tr>
<td>4</td>
<td>-29%</td>
<td>-22%</td>
<td>-29%</td>
<td>-32%</td>
<td>-32%</td>
<td>-23%</td>
<td>-28%</td>
</tr>
<tr>
<td>5</td>
<td>-13%</td>
<td>-44%</td>
<td>-42%</td>
<td>-51%</td>
<td>-43%</td>
<td>-8%</td>
<td>-49%</td>
</tr>
<tr>
<td>AVG</td>
<td>-17%</td>
<td>-26%</td>
<td>-28%</td>
<td>-30%</td>
<td>-20%</td>
<td>-11%</td>
<td>-41%</td>
</tr>
</tbody>
</table>

| 18x12          |           |           |         |        |           |               |              |
| 1              | -24%      | -35%      | -43%    | -27%   | -33%      | -9%           | -54%         |
| 2              | 2%        | -29%      | -34%    | -24%   | -25%      | -9%           | -52%         |
| 3              | -19%      | -9%       | -22%    | -27%   | 6%        | -9%           | -38%         |
| 4              | -3%       | 16%       | 1%      | 2%     | Not Taken | 2%           | Not Taken    |
| 5              | -11%      | -19%      | -22%    | -25%   | -17%      | -6%           | -26%         |
| AVG            | -11%      | -15%      | -24%    | -20%   | -17%      | -6%           | -43%         |

When comparing the carbureted and injected load points, a few points showed an increase in BSFC for the fourth load point for the 18x12 prop. This could be due to destructive resonance in the intake manifold using the injection setup, as very little decrease in BSFC or sometimes even increased BSFC was seen at this load point.

Resonance deals with pressure waves in the intake manifold caused by the closing of the intake valve. When the intake valve closes, it stops air flow, causing a reflected wave that moves at approximately the speed of sound. This wave moves back upstream before reflecting off of something (possible the bend in the intake manifold) and reflecting back towards the intake valve. At different engine speeds, this wave will either be moving toward or away from the intake valve when the intake valve opens on the next intake stroke. This can cause increased power or decreased power, depending on the direction of the wave. It is possible that the intake resonance characteristics of the manifold could be causing a negative impact on BSFC compared to the straight through design of the carbureted setup at certain engine speeds and load points.
When switching the engine from avgas to JP-8 with carburetion a 17% and 11% average decrease in BSFC was shown across the load points for the two props (17x10 and 18x12 respectively). This is promising when considering the military goal of using JP-8 in UAV engines to reduce fuel consumption and maintain performance. This is without engine modification, but with avgas as a starting fuel (no cold start on JP-8).

When switching from carburetion to injection while still using avgas, an approximate 27% average decrease in BSFC was seen for the 17x10 prop and an approximate 15% average decrease in BSFC for the 18x12 prop. This decrease in BSFC proved the effectiveness of fuel injection at decreasing fuel consumption. This is also on a first iteration manifold setup, which means further performance increases could be sought in the future.

Taking the next step, when switching from carbureted JP-8 to fuel injected JP-8, the 17x10 propeller points for the injected JP-8 showed approximately 11% lower BSFC than when carbureted (compared to baseline). About 13% decrease for the 18x12 prop was shown as well.

Moving on to the other test fuels, D2 diesel (PFI only) showed very similar results to JP-8 on PFI. Performance gains were slightly less (about 4% less) for the 18x12 prop for diesel compared to JP-8, while about 2% better performance was seen for diesel than JP-8 for the 17x10 prop.

The algae fuel blend was only run with PFI, and it showed about 20% and 17% improvements over baseline for the two prop load lines. This is not as good as either JP-8 or 100% D2 diesel, but still shows improvement over carbureted avgas. In the case of the
17x10 inch prop load line, the injected avgas actually performed better than the algae blend by about 6%.

Camelina was tested on both the carbureted engine and the injected engine. The carbureted Camelina showed about 11% decrease in BSFC for the 17x10 prop and about a 6% decrease in BSFC for the 18x12 prop. In terms of carbureted fuels tested, the Camelina fell between avgas and JP-8 in BSFC for almost all of the test points. The PFI Camelina performance was better than all of the other fuels. Approximately 43% and 41% (18 and 17 inch props respectively) BSFC improvement was shown over carbureted avgas. The injected Camelina fuel showed a 31% further decrease in BSFC for the 17x10 prop and a 36% further decrease in BSFC for the 18x12 prop compared to carbureted Camelina.

4.2 - Emissions Results

Engine emissions were characterized using the Enerac 700 unit described in the Experimental Setup chapter. To gather emissions data, the engine was started on avgas, switched to the test fuel, an appropriate amount of time was timed to clear the fuel lines of any avgas, and then the load points were sought by adjusting injector pulse width and spark timing. Then when the correct torque was found, an emissions data file was captured. The Enerac software (Enercom) was set to capture snapshots when the save button was pressed (this software ran separately than the LabView software) and then the snapshots were processed and averaged in Excel. Finally, the averaged load point data was added to the main data file that was captured for the performance side of the data (to keep things together). This was a cumbersome process, but worked well until integrating emissions unit data into the LabView program can be accomplished at a later time.
Rich equivalence ratios tend to cause more CO production because there is not enough oxygen present to convert the carbon in the fuel to CO2 (Ferguson & Kirkpatrick, 2001). Figure 46 and Figure 47 show the CO emissions for both propellers versus BMEP. The carbureted fuel test points shows CO emissions up to 10-12ppm, with carbureted avgas producing the most CO over the range of BMEP’s in most cases. The data for the three fuels testes in warmer weather (D2 diesel, algae blend, and Camelina) shows much lower (less than 2 ppm) CO production. The carbureted fuels ran with very rich equivalence ratio, which seems to correlate well with the high levels of CO production.

![Figure 46: 17x10 CO Emissions](image-url)
CO2 emissions are shown in Figure 48 and Figure 49 for both propeller load lines. Lower amounts of CO2 production can be caused by lower combustion efficiency and rich equivalence ratios. As CO production goes up with richer equivalence ratios, less CO2 is produced. In both figures, the lowest percentage of CO2 production is shown for carbureted avgas, while the highest percentage of CO2 production is shown for Camelina and diesel, both using injection. All of the injected fuels showed more CO2 production, suggesting that higher combustion efficiency was achieved with a small droplet size and closer to stoichiometric air/fuel mixture. Amongst the carbureted fuels, JP-8 and Camelina showed similar CO2 emissions that were higher than those of the avgas fuel.
Figure 48: 17x10 CO2 Emissions

Figure 49: 18x12 CO2 Emissions
Unburned hydrocarbon (UHC) emissions are shown in Figure 50 and Figure 51 for both propeller load lines. UHC emissions result from unburned fuel getting through to the engine exhaust. Most of the major species of fuel in common hydrocarbon fuels are seen in UHC emissions, although some new species are created during the internal combustion process. This is the result of hydrocarbon fuel combustion not being carried out to completion, and intermediate species making it through (Ferguson & Kirkpatrick, 2001). The most common ways that UHC emissions are created are in small crevices in the engine, oil layers, deposits, flame quench regions, and exhaust valve leakage. Crevices cause small local flame quench regions when the flame cannot propagate into the crevice. Also, on air cooled engines, inconsistent and low temperature combustion chamber walls will quench the flame, resulting in UHC’s near the walls caused by incomplete burn.

For the 17x10 propeller, UHC emissions are higher for the carbureted fuels, with avgas showing by far the highest UHC emissions (close to 2000 ppm). JP-8 and Camelina came in the 500 ppm range. The D2 diesel demonstrated the lowest UHC emissions. The injected fuels tended to show lower UHC emissions. For UHC emissions it is important to again emphasize the temperature difference in test conditions between carbureted and injected fuels. Low test cell temperature could cause more wall quenching and higher UHC emissions.

The 18x12 prop shows extremely high UHC emissions for the avgas fuel (note the different y-axis scale). For this more highly loaded load line JP-8 and Camelina (carbureted) still show an order of magnitude increase in UHC emissions over the injected fuels (at higher ambient temperatures). D2 diesel again demonstrated the lowest
UHC emissions. Future emissions testing will employ the use of a smoke test. It is valuable to note that the test cell operator noted that D2 diesel showed a large amount of smoke emitting from the engine exhaust.

Figure 50: 17x10 UHC Emissions
Oxides of nitrogen (NOx) emissions are shown in Figure 52 and Figure 53 for both propeller load lines. NOx emissions consist of NO and NO2 and are very temperature dependent. Increased temperatures allow for the chemical reactions that form NO and NO2 to happen more readily. Peak NOx emissions will occur at slightly lean mixtures where temperature is highest. Equivalence ratios in the 0.9 to 1.0 range will yield the highest NOx emissions and NOx will drop off as richer mixtures are seen (Ferguson & Kirkpatrick, 2001). The figure for the 17x10 and 18x12 propeller load lines show how the lower equivalence ratio numbers of the injected system support this. The injected fuels all showed higher NOx emissions, as they were running much closer to an equivalence ratio of 1. The carbureted fuels, running very rich, produced much less NOx. Increased ambient temperatures could also have increased NOx emissions by further increasing reaction temperatures. The highest NOx emissions were observed for injected
Camelina and diesel and the lowest NOx emissions were observed for carbureted avgas (equivalence ratios close to 1.3).

Figure 52: 17x10 NOx Emissions

Figure 53: 18x12 NOx Emissions
4.3 - Uncertainty

As a final note in the collection of data, two separate uncertainty analyses were completed as part of separate papers written during the course of this research. Appendix D contains an uncertainty analysis of measured torque at the dynamometer and calculated BSFC, based on fuel flow uncertainty. Measured torque uncertainty is in the 1-6% range for both a low torque condition and a high torque condition on the engine. BSFC uncertainty ranged from 1-50% for a low condition and 1-40% for a high condition. Appendix E contains an uncertainty analysis of fuel flow and air flow. Uncertainties for both were below 1%.
CHAPTER 5 - CONCLUSIONS

The main goal of this research was to test the feasibility of utilizing heavy, petroleum and bio-based, fuels in a small spark ignition engine while maintaining cruise performance, increase control over engine equivalence ratio, and evaluate engine emissions. The results for this research verify that a port-fuel-injection system for a small SI engine can increase control over equivalence ratio. This increased control over equivalence ratio showed decreased BSFC for all of the test fuels compared to the stock fuel with a carburetor excluding a few points. When the baseline fuel, avgas, was tested with the PFI system, 26% and 15% (average across all of the points for the 17x10 and 18x12 propeller load lines respectively) decrease in BSFC was shown. The few increased (compared to carbureted avgas) BSFC’s are suspected to be due to resonance characteristics of the engine with the injector manifold. The air-assisted injector demonstrated the ability to meter fuel in the flow ranges needed for this small engine. The air-assisted injector demonstrated small droplet sizes in the sub 15 micron range. These small droplet sizes aided in achieving better BSFC values by allowing for more complete evaporation of the fuels. The small droplet sizes were expected to aid in cold-starting of the engine, but cold-start on heavy fuels was not achieved.

JP-8 was used as the base heavy fuel to verify that heavy fuels could be used to operate this engine platform based on the required cruise torques for the different propellers. Data showed that the JP-8 fuel could achieve the torque values needed to
satisfy the cruise speeds while consuming 17% and 11% less fuel. Fuel consumption was further decreased to 28% and 24% when JP-8 was run with the air-assisted injector. The commercially available heavy fuel, D2 diesel, showed similar decreases in BSFC compared to carbureted avgas (30% and 20%).

Two types of biofuels were tested: Camelina and algae-based biodiesel. The algae fuel was blended with D2 diesel because of limited availability of algae oil (20% algae fuel and 80% diesel ratio). The Camelina fuel was run on the carbureted engine and the injected engine, while the algae blend was only run in the injected engine. The algae blend did not show as much of a decrease in BSFC as straight D2 diesel. For the 17x10 prop, 9% less of a decrease was shown for the algae blend versus the regular diesel. For the other prop, 3% less of decrease was shown. Just switching the carbureted engine from avgas to Camelina showed 11% and 6% decreases in BSFC, while the injected Camelina showed 42% and 43% decrease in BSFC. The combination of the air-assisted fuel injector and the Camelina biofuel demonstrated the best BSFC numbers.

The Camelina biofuel’s superior performance in this application could stem from a number of the fuel’s chemical properties. The higher Cetane number caters to the forgiving nature of this lower compression, air-cooled engine from a knock perspective. A fast energy release from the Camelina biofuel was likely aiding in energy release. A low flash and lower boiling point than D2 diesel aid in vaporization and ignition of the Camelina biofuel as well. Increased vaporization and tendency to ignite likely aided in energy release and combustion efficiency.

Emissions results showed the superior performance of the air-assisted injector over the carburetor, largely because of the increased ability to control fuel flow across the
whole engine speed range. For the fuel injected system, carbon monoxide and unburned hydrocarbon emissions were down compared to the carbureted system. Producing less CO meant more CO2 was emitted. Bring the mixture closer to stoichiometric meant an increase in combustion temperatures and an increase in NOx production.

Carbureted avgas tended to produce the most CO emissions, and the least CO2 emissions. It far surpassed all of the other fuels in unburned hydrocarbon emissions except for at two points (where JP-8 produced more). Also, avgas produced the lowest levels of NOx, largely due to the very rich mixture produced from the carburetor.

Carbureted JP-8 produced less CO than carbureted avgas, and slightly more CO2. Unburned hydrocarbons were down from carbureted avgas, while NOx production was up.

The last carbureted fuel was Camelina, and it showed lower CO emissions than carbureted avgas and JP-8, and slightly higher CO2 emissions. UHC’s were lower than that of JP-8. NOx emissions were similar to JP-8. For the injected Camelina fuel CO was about the same as the other injected fuels and CO2 levels were amongst the highest shown, indicating more of the fuel was going through to a complete reaction. UHC’s were similar to the other biofuels, with one point being higher than the carbureted version. This could have something to do with test cell temperature conditions (much higher for the injected data). NOx levels for the Camelina were amongst the highest.

D2 diesel CO emissions were similar to the other injected fuels (Camelina and algae blend) and far lower than the carbureted fuels. CO2 emissions were about the same or slightly lower than injected Camelina. UHC emissions were the lowest of all the fuels.
NOx emissions were low at the highest and lowest BMEP’s for both propellers, but just as high as Camelina for the midrange BMEP’s. The algae blend showed similar CO production to the other injected fuels, with slightly reduced CO2 production compared to the 100% D2 diesel. UHC emissions were generally lower than Camelina and slightly higher than D2 diesel. NOx levels were generally down from the 100% diesel fuel.

The data shows that it is possible to achieve better performance through decreased BSFC by using petroleum-derived heavy fuels and bio-derived heavy fuels for the specified propeller load lines. Also, the employment of electronically controlled, air-assisted, port fuel injection shows decreased BSFC. The PFI system could better control air/fuel mixture than carburetion and delivered a small droplet size, which aided in evaporation of the fuel. Emissions data showed lower CO emissions for the heavy fuels than the stock fuel. CO2 emissions were increased, indicating more complete combustion. UHC emissions were decreased compared to carbureted avgas substantially. The biofuels all demonstrated better UHC emissions than JP-8, with the D2 diesel fuel emitting the least UHC’s. NOx emissions went up for the heavy fuels compared to carbureted avgas, and as equivalence ratios came down closer to stoichiometric with the injected fuels, NOx emissions continued to increase.

In short, it was found that an electronic fuel injection system was effective in meeting the equivalence ratio control research objective. Also, petroleum and bio-based heavy fuels can match or better the cruise fuel consumption of the original UAV system configuration.
Secondary goals that arose during the research were also accomplished. Increased air flow measurement accuracy was achieved through the use of a large volume air damper, in which the chamber was separated into sections with polyurethane foam, to smooth out air flow oscillations characteristic of single cylinder, four-stroke engines. Finally, the air-assisted injector fuel and air flow were characterized.
CHAPTER 6 – FUTURE WORK

An ambient temperature control system for the engine intake air will be developed to achieve similar intake air temperatures regardless of the test cell conditions based on the weather. Low intake air temperatures were seen in the winter and high intake air temperatures in the summer. As the carbureted data for this testing was taken in the Fall and Winter, much of the data was taken in the cold, while the opposite is true of the injected fuels (Summer testing). An intake air heat exchanger could be installed, or some form of temperature controlled chamber built around the air damper.

The injector manifold had a 90 degree turn in the flow path. This was to simplify packaging and machining. A manifold with a less restrictive flowpath will be designed, as peak power was down from the carbureted engine.

More test fuels will be introduced. As more time goes by, UDRI will be able to produce enough algae biodiesel to fuel the engine on a 100% algae biodiesel blend. Also, a Fischer-Tropsch process synthetic heavy fuel (Syntroleum S8) will be tested. As other fuels become available for testing, more fuels will be tested and more blends will be created.

The engine to dyno coupler arrangement on this engine experienced failures due to vibrations induced by the engine at certain engine speeds. A new coupler arrangement will be designed that will reduce this vibration and extend the coupler life. This will
enable the operators to take larger data sets without having to tear down the coupler arrangement and replace the rubber dampers that have failed.

The comparison between fuels for this research was done using propeller load lines to minimize the number of data points that needed to be recorded. This method was useful for doing a BSFC comparison between the fuels and comparing emissions data. This method did not test the peak torque capabilities of the test fuels. In the future, peak torque values will be sought. Also, more time will be spent optimizing spark timing for different fuels.

Cold-start was never achieved on the engine with heavy fuels. Different methods for achieving cold start are possible. One method that could be effective is heating the air-assisted injector air to close to the boiling point of the fuel, as the fuel and air mix in the injector, some of this heat will be transferred to the fuel. As it is injected into small droplets the fuel will be more likely to start evaporating before it even gets to the intake valve. Upon entering the cylinder, the compression of the piston could raise combustion chamber temperature above the boiling point of the fuel, evaporating the fuel. Heat could also be applied to the intake manifold, or even the fuel itself. Small droplet sizes alone do not seem to be enough to initiate a cold start.

Emissions data did not include a smoke number. The D2 diesel in particular did create a large amount of smoke in the exhaust. In the future, a method for quantifying this smoke will be employed. One way this can be done is to pull a sample of emissions through a white cloth or paper filter and assign a number based on the darkness of the paper according to a printed scale.
Other types of injectors might be employed on the engine. The Orbital injector performs well, but is heavy and large. These are not desirable qualities for a small UAV engine. There are small injectors being developed based on ink-jet printer technology for small engines. New manifolds would have to be designed and machined.


APPENDIX A

Contour Plot MatLab Code

% This program will import raw data from LabView and perform several tasks:
% [A] - Delete all Rows of data that have RPM<1000, Torque<=0,
m_dot<=0 and throttle setting<5
% [B] - Sort Data by Throttle (ascending) first, then by RPM
% (ascending) second
% [C] - Create separate data files for each throttle setting and
% delete any that have less than 10 data points
% [D] - Look at data with matching RPM and first filter any points
% that lie outside of the specified standard deviations and second average the
% remaining data points together to create one data point per RPM.
% [E] - Produce several plots

% NOTE: Raw Data file must have no titles and columns should be sorted as
% follows:

% [Throttle(%)] [RPM] [Torque(Ft-Lb)] [Power(hp)] [Fuel Mass Flow(lb/hr)] [BSFC].
% [BMEP(psi)] [Ambient Temp(F)] [EGT(F)] [Fuel Temp(F)] [T1(F)] [T2(F)]
% For Octane > From CC/min to lb/hr, Multiply by 0.092542562 (At 74 deg F)
% and 0.088932713 (@ 140 deg F)----------------- To calculate BMEP Multiply
% Torque by 73.764523893229736
%
% For Heptane > From CC/min to lb/hr, Multiply by 0.08626997 (@140 deg F),
% and 0.090135114 (@74 deg F)
%
clear all; close all; clc;
load data_ac.txt

% Engine Map
figure
steps=100;       %30 for heptane 15 fpr octane
x=tot2(:,4);y=tot2(:,21);z=tot2(:,11); %x=RPM, y=BMEP, z=BSFC

91
\[ \text{xi} = \{\min(x) - \{(\max(x) - \min(x)) / \text{steps}\} : \max(x) \}\}; \\
\text{yi} = \{\min(y) - \{(\max(y) - \min(y)) / \text{steps}\} : \max(y) \}\}; \quad \text{Set grid size} \\
[\text{XI}, \text{YI}] = \text{meshgrid(xi, yi)}; \quad \text{Create x, y grid} \\
\text{ZI} = \text{griddata(x, y, z, XI, YI)}; \quad \text{Interpolate z values to new grid} \\
\% \text{create Filled in color Map} \\
\text{colormap jet} \\
v = \{.3 : 1 : .8 \ 9 : 2 : 1.9 \ 2.0 : 1 : 5\}; \\
[C, h] = \text{contourf(XI, YI, ZI, v)}; \\
\text{clabel(C,h,'manual','fontsize',14,'rotation',0)}; \\
\text{colorbar} \\
\text{axis([-2500 7500 10 100])} \\
caxis([0.3 2]); \\
xlabel('RPM') \\
ylabel('BMEP (psi)') \\
title('Fuji Engine Map') \\
\text{legend('BSFC (lb/hr/\text{hp})')} \\
\% \text{create black and white single line map} \\
figure \\
[C, h] = \text{contour(XI, YI, ZI, v, 'k')}; \\
\text{clabel(C,h,'manual','fontsize',14,'rotation',0)}; \\
axis([2000 7500 10 70]) \\
xlabel('RPM') \\
ylabel('BMEP (psi)') \\
title('Fuji Engine Map (BMEP vs RPM vs BSFC)') \\
\% \text{Engine Map} \\
figure \\
\text{steps} = 200; \quad \% 30 for heptane 15 fpr octane \\
x = \text{tot2(:,6)}; y = \text{tot2(:,5)}; z = \text{tot2(:,15)}; \quad \% x = RPM, y = Torque, z = BSFC \\
\text{x} = \{\min(x) - \{(\max(x) - \min(x)) / \text{steps}\} : \max(x) \}\}; \\
\text{yi} = \{\min(y) - \{(\max(y) - \min(y)) / \text{steps}\} : \max(y) \}\}; \quad \% \text{Set grid size} \\
[\text{XI}, \text{YI}] = \text{meshgrid(xi, yi)}; \quad \% \text{Create x, y grid} \\
\text{ZI} = \text{griddata(x, y, z, XI, YI)}; \quad \% \text{Interpolate z values to new grid} \\
\% \text{create Filled in color Map} \\
\text{colormap jet} \\
v = \{.3 : 1 : .8 \ 9 : 2 : 1.9 \ 2.0 : 1 : 5\}; \\
[C, h] = \text{contourf(XI, YI, ZI, v)}; \\
\text{clabel(C,h,'manual','fontsize',14,'rotation',0)}; \\
\text{colorbar} \\
\text{axis([-2500 7500 .1 1])} \\
caxis([0.3 2]); \\
xlabel('RPM') \\
ylabel('Torque (ft-lb)') \\
title('Fuji Engine Map (Torque vs RPM vs BSFC)') \\
\% \text{PROP LOAD LINES}
prop18_12=[4055 .39;4233 .42;4607 .51;5094 .62;5527 .76]; %column 1 is RPM, column 2 is torque (ft-lb)
prop17_10=[4544 .39;4735 .41;5152 .48;5669 .58;6213 .71];

hold on
plot(prop17_10(:,1),prop17_10(:,2),'k--','linewidth',2.5)
plot(prop18_12(:,1),prop18_12(:,2),'k-.','linewidth',2.5)

prop_leg=legend('BSFC (lb/hr/hp)','17x10 Prop Cruise','18x12 Prop Cruise',...
  'location','northoutside',...
  'orientation','horizontal');
legend('boxoff')
set(prop_leg,'fontsize',9)

%Vol Eff Map
figure
steps=100;           %30 for heptane 15 fpr octane
x=tot2(:,4);y=tot2(:,6);z=tot2(:,19).*100; %x=RPM, y=thr, z=Vol Eff
xi=[min(x):((max(x)-min(x))/steps):max(x)]';
yi=[min(y):((max(y)-min(y))/steps):max(y)]'; %Set grid size
[XI,YI]=meshgrid(xi,yi); %Create x,y grid
ZI = griddata(x,y,z,XI,YI); %Interpolate z values to new grid
%create Filled in color Map
colormap jet
v=[30 35 40 45 50 55 60 62 64 66 68 70 72 74 76 78];
[C,h] = contourf(XI,YI,ZI,v);
clabel(C,h,'manual','fontsize',14,'rotation',0);
colorbar
axis([2500 7500 10 100])
caxis([35 80]);
xlabel('RPM')
ylabel('Throttle(%)')
title('Fuji Engine Map')
legend('Vol. Eff (%)')

%Phi Map
figure
steps=100;           %30 for heptane 15 fpr octane
x=tot2(:,4);y=tot2(:,3);z=tot2(:,13); %x=RPM, y=Torque, z=Phi
xi=[min(x):((max(x)-min(x))/steps):max(x)]';
yi=[min(y):((max(y)-min(y))/steps):max(y)]'; %Set grid size
[XI,YI]=meshgrid(xi,yi); %Create x,y grid
ZI = griddata(x,y,z,XI,YI); %Interpolate z values to new grid
%create Filled in color Map
colormap jet
v=[.9:.02:1.1];
[C,h] = contourf(XI,YI,ZI,v);
%clabel(C,h,'manual','fontsize',14,'rotation',0);
colorbar
axis([2500 7500 .1 1])
caxis([0.9 1.1]);
xlabel('RPM')
ylabel('Torque (ft-lb)')
title('Fuji Engine Map (Torque vs RPM vs PHI)')

%BSFC Vs Throttle
figure
steps=100;           %30 for heptane 15 fpr octane
x=tot2(:,4);y=tot2(:,6);z=tot2(:,11);  %x=RPM, y=BMEP, z=Phi
xi=[min(x):((max(x)-min(x))/steps):max(x)]';
yi=[min(y):((max(y)-min(y))/steps):max(y)]';     %Set grid size
[XI,YI]=meshgrid(xi,yi);    %Create x,y grid
ZI = griddata(x,y,z,XI,YI);     %Interpolate z values to new grid
%create Filled in color Map
colormap jet
v=[.3:.1:.8 .9:.2:1.9 2.0:1:5];
[C,h] = contourf(XI,YI,ZI,v);
%clabel(C,h,'manual','fontsize',14,'rotation',0);
colorbar
axis([2000 7500 10 100])
caxis([0.3 2]);
xlabel('RPM')
ylabel('BMEP (psi)')
title('Fuji Engine Map')
%
%
Add Points where engine knocks with Heptane to the map
hold on
knock=[3000 2700 2500 2900 3300 3000;69.41 73.03 71.18 71.85 59.31 66.02]';
plot(knock(:,1),knock(:,2),'.r')
Add JP8 - BMEP vs RPM w/ Propeller to map
prop=[2700 2970 3450 4440 5040 5400 5490 5520 5700;14.16278859 15.93313716
20.50653764 34.5247886 49.86481815 54.58574768 56.28233173 59.08538364]';
plot(prop(:,1),prop(:,2),'-b','linewidth',2.5)
legend('BSFC(lb/hr/hp)','Prop Load Profile')
run 'H:\PR\PRT\PR\T\SHFE\Carys Folder\THESIS RESEARCH\Fuji Data\7Dec
Octane Low Speed\BSFC Comparison\prop_points.m'
legend('BSFC','Bolly 17x10 C','T/O','APC 17x10 C','T/O','TF 18x12 C','T/O','APC 22x10 C','T/O')

%hold on
%legend(legendstuff(3:1),'T/O','APC 22x10 C','T/O')
figure
surf(XI,YI,ZI)
shading interp
}
Spark and Servo Control Program

#include <18F452.h>
#device adc=10
#fuses HS, NOBROWNOUT, STVREN, NOWDT, NOLVP
#use delay(clock=40000000)

//This program implements variable spark timing for the predator engine. It requires
//an input of 35 pulses per revolution with the 36th pulse being a blank (locator) "pulse"
//to calculate correct spark angle and time. It is currently set up to wait for
//the 32nd pulse of the 36 pulse hall effect speed sensor before it calculates the new
time/angle.
//It calculates and updates the time/angle before it gets to the
//min_spark_angle where it then uses the new time to fire the spark.
//Timer uses a 17.5us count.

//#priority EXT1, TIMER2, EXT

#define OUTPIN_SPARK_12 PIN_D0 //pin #19
#define THROTTLE_OUTPUT PIN_D1 //pin #20
#define CHOKE_OUTPUT PIN_D2 //pin #21
#define SPARK_ENABLE_PIN PIN_D7 //pin #30

//CONSTANT VARIABLES
#define rev_length_buffer 30000 //this MUST BE LARGER THAN rev_length_max!!!
#define rev_length_max 9600 //250 rpm; low speed limit

#define spark_angle_channel 0 //pin #2
#define THROTTLE_INPUT 1 //pin #3
#define CHOKE_INPUT 2 //pin #4

//#define spark_angle_min -50.0 //Spark angle in degrees ATDC at 0V. So spark will be
//#define spark_angle_max 10.0 //Spark angle in degrees ATDC at 5V.

#define tps_min 41 //"tps" == "time pulse stop"
#define tps_max 81
#define max_servo_time 800 // 20 ms period at 25 microsec clock

#define ADC_DELAY 100 //this is in microseconds

int16 servo_time, tps_1, tps_2, throttle, choke, skip_rev_length_update;
int16 time, encoder_count, start_flag;
int16 time_pulse_16, time_pulse_22, time_pulse_1, time_pulse_7;
int16 stoptime_12, sparktime_12; // stoptime_34, sparktime_34;
int16 stoptime_12_new, sparktime_12_new; // stoptime_34_new, sparktime_34_new;
int8 sparktime_12_ready, rev_length12_updated; // sparktime_34_ready, rev_length34_updated;

float rev_length12; // rev_length34;

float spark_angle_slope;

#define TIMER2
TIMER2_isr()
{
    time++;

    if (sparktime_12_ready == TRUE) { // receive new spark times if they are ready
        sparktime_12 = sparktime_12_new;
        sparktime_12_ready = FALSE;
    }

    if (rev_length12 <= rev_length_max) {
        if (input(SPARK_ENABLE_PIN)) {
            if ((time == sparktime_12) && (skip_rev_length_update == 0)) {
                output_low(OUTPIN_SPARK_12);
                // output_high(OUTPIN_FREQ);
            } else {
                output_high(OUTPIN_SPARK_12);
                // output_low(OUTPIN_FREQ);
            }
        } else {
            output_high(OUTPIN_SPARK_12);
            // output_low(OUTPIN_FREQ);
        }
    } else {
        output_high(OUTPIN_SPARK_12);
        // output_low(OUTPIN_FREQ);
    }

    if (time >= rev_length_buffer) { // reset time if engine is not spinning so that buffer does not overflow.
        time = rev_length_buffer;
    }
}
rev_length12 = rev_length_buffer;
skip_rev_length_update = 1;
// rev_length34 = rev_length_buffer;
}
servo_time++;
if (servo_time == 1)
{
    output_high(THROTTLE_OUTPUT);
    output_high(CHOKE_OUTPUT);
}
if (servo_time == tps_1) output_low(THROTTLE_OUTPUT);
if (servo_time == tps_2) output_low(CHOKE_OUTPUT);
if (servo_time == max_servo_time) {
    servo_time = 0;
tps_1 = throttle;
tps_2 = choke;
}

} //End of Timer Interrupt

//Pin RB0 (#33) Pulse interrupt 0
//crankshaft location pulse
#int_EXT
EXT_isr()
{
    rev_length12 = time;
    set_timer2(0);
time = 0;
skip_rev_length_update = 0;
// locate spark pulse with speed indicator
    // if (start_flag == 0){
    //     start_flag = 1;
    //     encoder_count = 0;
    // }  
    // else if ((start_flag == 1) && (encoder_count > 10)) {
    //     encoder_count = 0;
    // }
}
//End of crankshaft position pulse interrupt

//Pin RB1 (#34) Pulse interrupt 1
//hall effect speed sensor timing
pulse
//#int_EXT1
//EXT1_isr()
//
/*increment speed sensor count for each indicator
encoder_count++;  
//measure time between 21st and 27th pulse (6 pulses) on 36 position indicator
if (encoder_count == 16) {
    time_pulse_16 = time;
}
//wait until speed sensor's 27th pulse and update rev_length to calc new fire time
if (encoder_count == 22) {
    time_pulse_22 = time;
    // set_timer2(0);
    // time = 0;
    // rev_length12_updated = TRUE;
}
if (encoder_count == 29) {
    time_pulse_25 = time;
    set_timer2(0);
    time = 0;
    skip_rev_length_update = 0;
    // rev_length12_updated = TRUE;
}
// if (encoder_count == 1) time_pulse_1 = time;
//wait until speed sensor's 9th pulse and update rev_length to calc new fire time
// if (encoder_count == 7) {
//    time_pulse_7 = time;
//    rev_length34_updated = TRUE;
// }
*/
//}  
//End of hall effect speed sensor timing pulse
interrupt***********************************

void main()
{
    setup_adc_ports(A_ANALOG);
    setup_adc(ADC_CLOCK_DIV_64);
    setup_psp(PSP_DISABLED);
    setup_spi(SPI_SS_DISABLED);
    setup_wdt(WDT_OFF);
    setup_timer_0(RTCC_INTERNAL);
    setup_timer_1(T1_DISABLED);
    setup_timer_2(T2_DIV_BY_1,249,1); //Time is in 17.5 microsecond increments -
> (175*4/40000000)*1=0.0000175
    setup_timer_3(T3_DISABLED|T3_DIV_BY_1);
    setup_ccp1(CCP_OFF);
    setup_ccp2(CCP_OFF);
    disable_interrupts(INT_TIMER2); //Block interrupts momentarily
}
disable_interrupts(INT_EXT);
// disable_interrupts(INT_EXT1);
disable_interrupts(GLOBAL); //Disable Global interrupt
ext_int_edge(H_TO_L); //Look for positive edges on interrupt 0
// ex int_edge(1,L_TO_H); //Look for positive edges on interrupt 1

//Initialize variables******************************************************
time = 0;
// encoder_count = 0;
// start_flag = 0;
// time_pulse_16 = 0;
// time_pulse_22 = rev_length_max;
// time_pulse_1 = 0;
// time_pulse_7 = rev_length_max;

skip_rev_length_update = 1;

servo_time = 0;
tps_1 = tps_min;
tps_2 = tps_min;
throttle = tps_min;
choke = tps_min;

rev_length12 = rev_length_buffer; //Default to a high cycle time
// rev_length34 = time_pulse_7 - time_pulse_1;

spark_angle_slope = 0.0009775171;

set_adc_channel(spark_angle_channel);
delay_us(100);

sparktime_12 = rev_length12 * 0.16666666 * spark_angle_slope * read_adc();
sparktime_12_new = sparktime_12;
output_high(OUTPIN_SPARK_12);
// output_low(OUTPIN_FREQ);
rev_length12_updated = FALSE;
sparktime_12_ready = FALSE;
/*
sparktime_34 = rev_length34 + (rev_length34 * spark_angle_slope * read_adc());
sparktime_34_new = sparktime_34;
output_high(OUTPIN_SPARK_34);
rev_length34_updated = FALSE;
sparktime_34_ready = FALSE;
*/
//End initialize variables******************************************************

enable_interrupts(GLOBAL); //Enable Global interrupt
enable_interrupts(INT_EXT); //Begin watching crankshaft timing pulse
// enable_interrupts(INT_EXT1);  //Begin watching 25 tooth gear timing pulse
enable_interrupts(INT_TIMER2);  //Enable timing

while (1){

    set_adc_channel(spark_angle_channel);
delay_us(ADC_DELAY);

    if (skip_rev_length_update == 0) {
        //  while ((encoder_count != 23) && (skip_rev_length_update == 0)) {} // &&
        (rev_length12_updated == FALSE)) {}  //wait until speed sensor's 27th pulse and update
        rev_length to calc new fire time
        //  rev_length12 = time_pulse_22 - time_pulse_16;
        sparktime_12_new = rev_length12 * 0.16666666 * spark_angle_slope * read_adc();
        sparktime_12_ready = TRUE;
        rev_length12_updated = FALSE;
    }
    else delay_us(ADC_DELAY);
    /*
    while ((encoder_count != 7) && (rev_length34_updated == FALSE)) {}  //wait until speed
    sensor's 9th pulse and update rev_length to calc new fire time
    rev_length34 = time_pulse_7 - time_pulse_1;
    sparktime_34_new = time_pulse_7 + rev_length34 + (rev_length34 * spark_angle_slope *
    read_adc());
    sparktime_34_ready = TRUE;
    rev_length34_updated = FALSE;
    */

    //READ AND CONVERT THE THROTTLE VALUE
    set_adc_channel(THROTTLE_INPUT);
delay_us(ADC_DELAY);  //give the ADC time to switch
throttle = 0.0391 * read_adc() + 41.0; //1 to 2 ms(0.01955034 5 volts scale now 4.2 Volt)
    if (throttle <= tps_min) throttle = tps_min;
    if (throttle >= tps_max) throttle = tps_max;

    //READ AND CONVERT THE CHOKE VALUE
    set_adc_channel(CHOKE_INPUT);
delay_us(ADC_DELAY);  //give the ADC time to switch
choke = 0.0391 * read_adc() + 41.0; //1 to 2 ms(0.01955034 5 volts scale now 4.2 Volt)
    if (choke <= tps_min) choke = tps_min;
    if (choke >= tps_max) choke = tps_max;

}
Injector Control Program

#include <18F452.h>
device adc=10
#use delay(clock=40000000)
#fuses HS, NOBROWNOUT, STVREN, NOWDT, NOLVP
#include <math.h>
#define OUTPIN_FUEL_PEAK PIN_D0  //pin#19
#define OUTPIN_FUEL_HOLD PIN_D1  //pin#20
#define OUTPIN_AIR_PEAK PIN_D2  //pin#21
#define OUTPIN_AIR_HOLD PIN_D3  //pin#22
#define FUEL_ENABLE_PIN PIN_D7  //pin #30

//CONSTANT VARIABLES
#define time_buffer 40000
#define rev_length_max 20000 //250 RPM min
#define pw_channel 1  //desired fuel pulsewidth channel. Pin #3

#define fueltime_start 5 //time counts to wait to inject fuel after time is reset (must be greater than opening delay)

#define fuel_duration_peak_max 40.0 //time at peak voltage for fuel injector. 40.0 * 20.0 = 0.80ms
#define fuel_duration_hold_max 400.0 //time at hold voltage for fuel injector. 600.0 * 20.0 = 12.00ms

#define air_delay 60.0 //time between end of fuel hold voltage and beginning of air peak voltage. 60.0 * 20.0 = 1.20ms
#define air_duration_peak 40.0 //time at peak voltage for air injector. 40.0 * 20.0 = 0.80ms
#define air_duration_hold 100.0 //time at hold voltage for air injector. 100.0 * 20.0 = 2.00ms

int8  rev_length_updated, even, start_flag;
//int8  cam_flag, cam_located, start_flag;

int16 time, encoder_count, rev_length, start_rev_count;// turtle;
//int16 time_pulse_41, time_pulse_59;
int16 stoptime_peak, stoptime_hold, fueltime;
float stoptime_peak_new, stoptime_hold_new, fueltime_new;
float fuel_duration, fuel_duration_peak, fuel_duration_hold;
float pw_slope, pw_intercept;
int16 airtime, stoptime_air_peak, stoptime_air_hold;
float airtime_new, stoptime_air_peak_new, stoptime_air_hold_new;

#define TIMER2
TIMER2_isr()
{

}
time++;

if ((input(FUEL_ENABLE_PIN) == TRUE) && (rev_length < rev_length_max)) {
    if (time == fueltime) {
        output_high(OUTPIN_FUEL_PEAK);
        output_high(OUTPIN_FUEL_HOLD);
    }
    if (time == stoptime_peak) {
        output_low(OUTPIN_FUEL_PEAK);
    }
    if (time == stoptime_hold) {
        output_low(OUTPIN_FUEL_HOLD);
    }
    if (time == airtime) {
        output_high(OUTPIN_AIR_PEAK);
        output_high(OUTPIN_AIR_HOLD);
    }
    if (time == stoptime_air_peak) {
        output_low(OUTPIN_AIR_PEAK);
    }
    if (time == stoptime_air_hold) {
        output_low(OUTPIN_AIR_HOLD);
    }
    else {
        output_low(OUTPIN_FUEL_PEAK);
        output_low(OUTPIN_FUEL_HOLD);
        output_low(OUTPIN_AIR_PEAK);
        output_low(OUTPIN_AIR_HOLD);
    }
}
else {
    output_low(OUTPIN_FUEL_PEAK);
    output_low(OUTPIN_FUEL_HOLD);
    output_low(OUTPIN_AIR_PEAK);
    output_low(OUTPIN_AIR_HOLD);
}

if (time >= time_buffer) //reset time if engine is not spinning so that buffer does not
    overflow.
    time = rev_length_max;
    rev_length = rev_length_max;
    //start_flag = TRUE;
}

} //End of Timer Interrupt ****************************

//Pin RB0 (#33) Pulse interrupt 0
//crankshaft location pulse**************************************
#if EXT
EXT_isr()
{
    if (even == FALSE) {
        even = TRUE;
        set_timer2(0);
    }
}
time = 0;
}
// if (time > 700) {
//   if (even == FALSE) {
//     even = TRUE;
//     rev_length = time;
//     time = 0;
//   }
// }
else if (even == TRUE) {
  even = FALSE;
  rev_length = time;
}
// else if (time > 350) {
//   if (even == TRUE) {
//     even = FALSE;
//     rev_length = time;
//   }
// }
// time = 0;
}
// End of crankshaft position pulse interrupt

// Pin RB1 (#34) Pulse interrupt 1
// hall effect speed sensor timing pulse
/**/*.int_EXT1
EXT1_isr()
{
  // if (start_flag == TRUE)
    encoder_count++;
}
*/
// End of hall effect speed sensor timing pulse interrupt

// Pin RB2 (#35) Cam interrupt
// cam location pulse
/**
#int_EXT2
//EXT2_isr()
//{
//  if (start_flag == TRUE) {
//    encoder_count = 55;
//    start_flag = FALSE;
//  }
//  // cam_flag = TRUE;
//}*/
// End of cam position pulse interrupt

104
void main()
{
    setup_adc_ports(A_ANALOG);
    setup_adc(ADC_CLOCK_DIV_64);
    setup_psp(PSP_DISABLED);
    setup_spi(SPI_SS_DISABLED);
    setup_wdt(WDT_OFF);
    setup_timer_0(RTCC_INTERNAL);
    setup_timer_1(T1.Disabled);
    setup_timer_2(T2_DIV_BY_1,199,1);  //Time is in 16 microsecond increments - >{(160*4/40000000)*1}=0.0000160. Setting timer less than 159 is unstable.
    setup_timer_3(T3_DISABLED|T3_DIV_BY_1);
    setup_ccp1(CCP_OFF);
    setup_ccp2(CCP_OFF);
    disable_interauts(INT_TIMER2);  //Block interrupts momentarily
    disable_interauts(INT_EXT);
    // disable_interauts(INT_EXT1);
    // disable_interauts(INT_EXT2);
    disable_interauts(GLOBAL);  //Disable Global interrupt
    ext_int_edge(H_TO_L);   //Look for positive edges on interrupt 0
    // ext_int_edge(1,H_TO_L);  //Look for positive edges on interrupt 1
    // ext_int_edge(2,H_TO_L);  //Look for negative edges on interrupt 2

    //Initialize variables******************************************************
    time = rev_length_max;
    even = FALSE;
    start_flag = TRUE;
    rev_length = rev_length_max;
    // encoder_count = 0;

    pw_slope = fuel_duration_hold_max / 1023.0;

    set_adc_channel(pw_channel);
    delay_us(10);
    fuel_duration = pw_slope * read_adc();

    if (fuel_duration < 1) fuel_duration = 1;

    if (fuel_duration <= fuel_duration_peak_max) {
        fuel_duration_peak = fuel_duration;
        fuel_duration_hold = 1;
    } else {
        fuel_duration_peak = fuel_duration_peak_max;
        fuel_duration_hold = fuel_duration;
    }
if (fuel_duration_hold < 1) fuel_duration_hold = 1;
if (fuel_duration_hold > fuel_duration_hold_max) fuel_duration_hold =
fuel_duration_hold_max;

fueltime_new = fueltime_start;
if (fueltime_new < 1) fueltime_new = 1;
stoptime_peak_new = fueltime_new + fuel_duration_peak;
stoptime_hold_new = stoptime_peak_new + fuel_duration_hold;

airtime = stoptime_hold_new + air_delay;
stoptime_air_peak = airtime + air_duration_peak;
stoptime_air_hold = airtime + air_duration_hold;

output_low(OUTPIN_FUEL_PEAK);
output_low(OUTPIN_FUEL_HOLD);
output_low(OUTPIN_AIR_PEAK);
output_low(OUTPIN_AIR_HOLD);

//End initialize variables*****************************************************************************

enable_interrupts(GLOBAL);  //Enable Global interrupt
enable_interrupts(INT_EXT);  //Begin watching crankshaft timing pulse
// enable_interrupts(INT_EXT1);  //Begin watching 25 tooth gear timing pulse
// enable_interrupts(INT_EXT2);  //Begin watching cam sensor pulse
enable_interrupts(INT_TIMER2);  //Enable timing

while (1){

while (time != 0) {};

fueltime = fueltime_new;  //recieve new times
stoptime_peak = stoptime_peak_new;
stoptime_hold = stoptime_hold_new;
airtime = airtime_new;
stoptime_air_peak = stoptime_air_peak_new;
stoptime_air_hold = stoptime_air_hold_new;

fuel_duration = pw_slope * read_adc();

if (fuel_duration < 1) fuel_duration = 1;

fuel_duration_peak = fuel_duration;
if (fuel_duration_peak > fuel_duration_peak_max) fuel_duration_peak =
fuel_duration_peak_max;
fuel_duration_hold = fuel_duration;
if (fuel_duration_hold > fuel_duration_hold_max) fuel_duration_hold =
fuel_duration_hold_max;
fueltime_new = fueltime_start;

stoptime_peak_new = fueltime_new + fuel_duration_peak;
stoptime_hold_new = fueltime_new + fuel_duration_hold;

airtime_new = stoptime_hold_new + air_delay;
stoptime_air_peak_new = airtime + air_duration_peak;
stoptime_air_hold_new = airtime + air_duration_hold;

// while (even == TRUE) {};

// while (encoder_count != 72) {};
  // encoder_count = 0;
}
APPENDIX B

Performance Analysis Terms

Some commonly used methods for analyzing the performance of an engine are brake mean effective pressure (BMEP) and brake specific fuel consumption (BSFC). In both cases, the term “brake” means that the term is on a power basis, and the power is measured power, not theoretical power.

Power is measured using engine torque and engine speed. The torque (twisting force) an engine generates is measured using a load cell at a known distance from the center of rotation. A dynamometer tracks both engine speed and torque and uses Equation 3 to get a power value. Power is the ability to do work. Engine power is often reported in horsepower (hp) in U.S. units or kilowatts (kW) in SI units.

\[
P(hp) = \frac{\tau \times N}{5252} \tag{3a}
\]

Where:
- \(hp\) = engine power in horsepower (hp)
- \(\tau\) = engine torque (lbf-ft)
- \(N\) = engine speed (rev/min or rpm)

or:

\[
P(kW) = \frac{2\pi N \tau}{1000} \tag{3b}
\]

Where:
- \(kW\) = engine power in kilowatts (kW)
- \(N\) = engine speed (rev/sec)
- \(\tau\) = engine torque (N-m)
BMEP is a good way to normalize engine power for different displacements. In this way, one can compare different sized engines. BMEP is the average pressure that results in the same amount indicated or brake work produced by the engine (Ferguson & Kirkpatrick, 2001). Equation 4 shows BMEP on a torque basis (U.S. units).

\[
BMEP = \frac{4\pi \tau}{V_d} = \frac{4\pi \tau}{2.04ci} \left( \frac{12ln}{ft} \right)
\]  

(4)

Where:
Vd = displaced volume in cubic inches (ci)

Engine fuel consumption for different engines is normalized by using BSFC. BSFC is reported as fuel consumption (by mass) per time per power (lb/hr/hp or g/kWh) (Equation 5).

\[
BSFC = \frac{m_f}{P}
\]  

(5)
Appendix C

Max Machinery Fuel Flowmeter

Model 213 Piston Flow Meter
1 to 1800 cc/min

Specifications:
- Flow Range (at 1 psi):
- Maximum Operating Pressure: 70 bar or 210 bar (1000 or 3000 psi)
- Displacement: 0.89 cc/sec
- Weight: 0.6 kg
- Recommended Filtration: 10 micron
- Port Sizes: UP/NPT in. 5/8" or manifold mount
- Accuracy: ± 0.2% of reading with a linearized transmitter
- ± 0.5% of reading with a non-linearized transmitter
- Fluids: Most non-aggressive, organic liquids

Materials of Construction:
- Body: Stainless steel, type 303
- Piston: Nitride hardened stainless steel, type 303
- Crankshaft: Stainless steel
- Bearing: All ball bearings, 440C stainless steel
- O Rings: 'O' ring standard, PTFE-Premium elastomer

Dimensions:

Figure 54: Max Machinery Flowmeter
**TSI 4021 Mass Air Flowmeter**

### Model Designations

<table>
<thead>
<tr>
<th>Model</th>
<th>Gas Calibration</th>
<th>Flow Range</th>
<th>Inlet Adapter</th>
<th>Outlet Adapter</th>
</tr>
</thead>
<tbody>
<tr>
<td>40211</td>
<td>Air</td>
<td>0 to 300 standard L/min</td>
<td>22 mm ISO tapered Male</td>
<td>22 mm ISO tapered Male</td>
</tr>
<tr>
<td>40212</td>
<td>Oxygen</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>40241</td>
<td>Air</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>40242</td>
<td>Oxygen</td>
<td>0 to 300 standard L/min</td>
<td>0.75 inch (19.1 mm) straight</td>
<td>0.75 inch (19.1 mm) straight</td>
</tr>
<tr>
<td>40246</td>
<td>Nitrogen*</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>41211</td>
<td>Air</td>
<td>0.01 to 20 standard L/min</td>
<td>0.25 inch (6.4 mm) straight</td>
<td>0.25 inch (6.4 mm) straight</td>
</tr>
<tr>
<td>41212</td>
<td>Oxygen</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>41216</td>
<td>Nitrogen*</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>41221</td>
<td>Air</td>
<td>0.01 to 20 standard L/min</td>
<td>0.375 inch (9.53 mm) straight</td>
<td>0.375 inch (9.53 mm) straight</td>
</tr>
<tr>
<td>41222</td>
<td>Oxygen</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>41226</td>
<td>Nitrogen*</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*Nitrogen models are calibrated in air and a nitrogen correction is applied by the meter.*

**Figure 55: TSI 4021 Mass Air Flowmeter**
**Magtrol ED-715-6N**

<table>
<thead>
<tr>
<th>Model</th>
<th>Torque Measure Unit Code</th>
<th>Maximum Torque Range</th>
<th>Drag Torque at 1000 rpm</th>
<th>Nominal Input Inertia</th>
<th>Max. Power Ratings</th>
<th>Maximum Speed</th>
<th>Brake Cooling Method</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>lb-ft s²</td>
<td>kg-m²</td>
<td>5 minute</td>
<td>continuous</td>
<td>rpm</td>
</tr>
<tr>
<td>ED-715</td>
<td>0N</td>
<td>55.0 lb in</td>
<td>0.3 lb in</td>
<td>62.0 kg cm</td>
<td>0.36 kg cm</td>
<td>1.27 × 10⁻³</td>
<td>1.72 × 10⁻³</td>
</tr>
<tr>
<td></td>
<td>8N</td>
<td>6.20 N·m</td>
<td>0.035 N·m</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ED-815</td>
<td>0N</td>
<td>28.0 N·m</td>
<td>0.14 N·m</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>5N</td>
<td>250 lb in</td>
<td>1.2 lb in</td>
<td>280 kg cm</td>
<td>1.4 kg cm</td>
<td>9.51 × 10⁻³</td>
<td>1.30 × 10⁻²</td>
</tr>
<tr>
<td></td>
<td>8N</td>
<td>28.0 N·m</td>
<td>0.14 N·m</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

* The maximum speed will depend on what type of keyway (if any) is used on the shaft. Unless specified, the dynamometer shaft will be made without a keyway.

** 5 Volt Output
Electric Starter Mount Plate

Figure 57: Electric Starter Mount Plate
Electric Starter Crankshaft Coupler

Figure 58: Electric Starter Crankshaft Coupler
Aluminum Injector Manifold

Figure 59: Aluminum Injector Manifold
The Performance and Emissions Effects of Utilizing Heavy Fuels and Biodiesel in a Small Spark Ignition Internal Combustion Engine

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Innovative Scientific Solutions Inc., Dayton, OH, 45440

Abstract

The Department of Defense has a sustained interest in the universal use of heavy, low volatility hydrocarbon fuels for safety and logistical reasons. There is also an increased environmental interest in lower emission, domestically grown biofuels. Many Remotely Piloted Aircraft (RPA) use small internal combustion engines for propulsion that are designed to run on high octane, highly volatile fuels. This paper discusses the experimental results of running a 4-stroke, spark ignition, 33.5 cc internal combustion engine with Avgas, JP-8 and Camelina Biofuel. Carburetor settings were unaltered and data is shown for two sets of engine loading points derived from torque requirements for cruise conditions with different fielded propeller load lines. Results show that for cruise points required torque output could be achieved with heavy fuels. Brake specific fuel consumption (bsfc) was lower for the heavy fuels, with JP-8 showing the lowest specific fuel consumption. Emissions data for the heavy fuels (compared to Avgas) shows lower CO and hydrocarbon emissions and higher CO2 and NOx emissions. If cold starting, fuel metering, and knocking issues can be resolved, preliminary results indicate heavy fuels can meet power requirements while lowering fuel consumption and CO/hydrocarbon emissions.

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Nomenclature

bmep – brake mean effective pressure (lbf/in²)
bsfc – brake specific fuel consumption (lb/hr/hp)
COTS – Commercial-off-the-shelf
RPA – Remotely Piloted Aircraft
UAS – Unmanned Aircraft Systems

I. Introduction

Current small-scale propulsion and power systems, characteristic of those utilized in USAF Groups I-IV Unmanned Aircraft Systems (UAS), typically use commercial-off-the-shelf (COTS) propulsion solutions with some modifications which are not optimized for DoD operations. There have been recent trends in the aerospace community to improve the metrics of power density and specific fuel consumption for these small-scale systems (less than 750 kW). As shown in Error! Reference source not found., higher power densities are more achievable with small, high-OPR turbine engines, but not necessarily with the fuel efficiency (specific fuel consumption) of more advanced Otto/Diesel-cycle-based internal combustion (IC) engines. While IC engines have shown improvements in fuel efficiency, they have had limitations to fundamentally advance the power density figure of merit. At the lower end of rated power, those limitations can be more pronounced; at power ratings of 7.5 kW or less, turbine engines will be very difficult to operate at relatively high fuel efficiency, while IC engines will trend towards lower power densities.

The challenge for hydrocarbon-fueled engines (typically Otto, diesel or Brayton cycle) at very low rated power outputs has usually been maximizing the fuel efficiency and the power density in terms of specific power (i.e. power per unit mass). The combustion cycle requires tighter control of its stages to ensure optimization of combustion and power generation. Fuel delivery, injection and ignition subsystems must scale down and still operate with high effectiveness and durability. The means of controlling the excess heat flow and engine cycle acoustic emissions must also be considered in their impact on cycle performance and size/weight in addition to the overall engine and power system platform integration.

In addition to the goal to move small-scale propulsion systems into the UAS Trade Space (Figure 60), there has been a long standing DoD desire to move to a single kerosene-like fuel. As such, alternative fuels derived from biomass are one path towards a single-fuel concept as well as energy independence. Bio-derivative fuel sources are an important component to the United States gaining independence from foreign oil and developing cleaner, more secure methods of energy production. The United States Air Force has begun to partner with the civilian aviation industry in order to lessen the 15 to 16 billion gallon per year dependence on conventional petroleum based fuels. The Air Force has successfully tested domestically produced bio-fuels on three different platforms with the goal to have 50 percent of its aviation fuel coming from alternative sources by 2016.

The research detailed in this paper focuses on studying the performance effects and emissions profiles of using JP-8 and bio-diesel, in a small spark ignition internal combustion engine as compared to Avgas. Brake mean effective pressure (bmep) and brake specific fuel consumption (bsfc) are the key performance parameters to be studied in addition to emissions profiles. The test engine was a 33.5cc 4-Stroke Fuji-Imvac BF34-EI, and baseline data was gathered using 100 octane, low-lead Avgas. This engine was selected due to its current use in an RPA and in hopes of being able to apply achieved experimental results to larger scale engines.
II. Experimental Setup

The engine and dynamometer are the necessary elements of this experiment, all other devices and sensors are in support of these. A Magtrol ED-715 dynamometer was used to apply an engine load via a set speed and torque reading. The engine was started using a COTS on-board remote starter. A LabView control program was developed in-house to acquire sensor data and control the engine. Low speed data (1 Hz) was taken for 8 seconds once steady state conditions had been reached and high speed data snapshots (50 kHz) were also taken. The low speed data presented in this paper include torque, fuel flow, engine speed, air flow and temperatures. The high speed data presented in this paper include in-cylinder pressure.

To withstand the torsional loading intrinsic to single cylinder engines, two Lovejoy LF-2 torsional couplings were used along with a pillow block carrier bearing. Cooling air was supplied to reduce the significant heat buildup in the bearing after extended engine running (Figure 61). Shaft alignment was completed to within 2-3 thousandths of an inch.

![Figure 61: Dynamometer/engine coupler arrangement](image)

Accurate air flow measurement was needed to obtain equivalence ratios at different operating conditions. To eliminate unsteady air flow measurement, due to the oscillatory air flow characteristics of a single cylinder engine, a 55 gallon polyethylene drum was used as an air damper between the engine and mass air flow sensor. A TSI 4021 Mass Air Flow Meter was fitted to the drum and a 10 foot section of 3.5 inch air duct was attached to the opposite side of the drum (Figure 62). The large mass of air inside the drum, which is orders of magnitude larger than the mass of air being drawn through the carburetor, eliminated flow fluctuations. To remove any measurement delay issues, only averaged steady state data was to be taken for experimentation.

During experimentation it was difficult to run the engine on heavy fuels such as JP-8 that have a much lower vapor pressure (compared to Avgas). To aid in fuel evaporation and combustion, an intake air heater was developed to increase the final air/fuel mixture temperature. To aid in starting on heavy fuels, the engine was first started on Avgas, and then switched over to JP-8 once operating temperatures were reached. The fuel tanks were pressurized with 5 psi of Nitrogen to get the fuel to the engine. Intake heating temperature was set to simulate warm day operating conditions of 75 °F at the engine intake.
The stock crankcase vent routing caused problems in that under certain conditions the engine would pump significant fractions of its oil sump contents out of the engine. It is not known definitively what engine conditions caused this sudden loss of oil. Army research on this engine documented similar oil consumption issues. Following suite with the Army, the crankcase vent tubes were re-routed which eliminated most of the oil consumption (Figure 63). When operating with heavy fuels such as JP-8 more fluid gets pumped out the vent tubes. This is thought to happen because of unburned fuel seeping past the piston rings and into the sump, therefore increasing oil sump pressure.

An Enerac 700 emissions testing unit was used to collect emissions concentrations in the engine exhaust. An exhaust emissions sampling port was welded into the muffler away from the outlet of the muffler to avoid reading any fresh air. A sampling line of approximately 4 feet connected the Enerac unit and the muffler. Averaged data was taken for 10 seconds.

III. Experimental Data - Performance

All data collected were with respect to engine loadings that matched cruise conditions characteristic of USAF Group I UAS with two propellers; APC 17x10 and TopFlight Powerpoint 18x12. Six cruise-points/engine-loadings were used per propeller, five corresponding to cruise conditions at increasing airspeed and one for take-off at 35 knots. The points are extrapolated from required thrust at an airspeed, so it is not necessarily possible to hit some of the more highly loaded points. To hit a required load point, the dyno rpm was set at the specified speed and then throttle opening was varied until the desired torque was
achieved on the dyno readout. Once this point was reached data was averaged using the Labview control/data acquisition program for 8 seconds and an emissions file was captured.

For the heavy fuels, the procedure for gathering data involved starting the engine on Avgas and waiting for the engine temperature to reach 230 °F, then the fuel was switched to the testing fuel via a manual three-way valve, and the time was noted. To verify the engine was running on the test fuel (there was a short section of fuel tubing that needed to be cleared of start-up Avgas) one full minute was counted (about 150% of calculated time to clear the line) before taking data. To calculate the time to clear the line, the volume inside the tubing was divided by the observed volumetric fuel flow rate.

For all testing, the carburetor was adjusted to the manufacturers recommended position of 1.75 turns out for the high speed needle and 1.33 turns out for the low speed. This caused different equivalence ratios to be measured, but for the purposes of this experiment it was deemed the best way to keep things on the same basis. Due to the extreme sensitivity of the carburetor settings and the relatively low flowrates associated with small-scale propulsion and power systems, in future tests equivalence ratio will be standardized by electronic fuel injection. This will allow a much more “apples to apples” comparison of fuel performance.

Figure 64 shows the theoretical propeller load points with a solid line and the experimental loading for the different fuels. The plot demonstrates the ability to achieve the prescribed propeller load points. The 17 inch propeller has a smaller diameter and a less aggressive pitch than the 18 inch propeller. This results in the 18 inch propeller spinning at a lower speed for the same cruise velocity, therefore loading the engine to a higher torque for the same engine speed.

The higher the engine loading for the same, or in this case lower, engine speed, the more likely the engine will be to knock. Engine knocking is an undesirable auto-ignition condition in an Otto-cycle engine in which an auto-ignited flame front develops somewhere other than the spark plug at the same time or shortly after the spark event. High speed in-cylinder pressure will show a very unsteady pressure profile after the spark event if knock is present. Prolonged knocking can cause engine damage from increased engine temperatures and cylinder pressure peaks above the normal maximum.

A common goal of engine design is to maximize performance while avoiding loading situations that can induce knock. Take-off load points are not presented because for any fuel other than Avgas, knock was observed at the 18x12 take off point and the required torque could not be reached.

Figure 65 shows pressure spikes experienced at a cruise
point for a 22x10 prop. The pressure spikes happen after the spark event, so the observed situation is knock and not pre-ignition. Data from the unused 22x10 prop load points is shown because it exemplifies engine knock clearly.

![Graph showing BSFC and phi vs rpm for 17x10 and 18x12 props](image)

The results of Figure 66 show that JP-8 exhibits the lowest fuel consumption for the same bmep demand at all points except the third cruise load point for the 17x10 prop and the second cruise load point for the 18x12 prop. In both cases, the Camelina biofuel showed lower bsfc than JP-8 for this point. Also, for both props, the Camelina biofuel and JP-8 showed lower bsfc in all cases except one instance at the 5090 rpm load point in which the Camelina showed slightly higher bsfc than Avgas. With stock carburetor settings, at these load points, JP-8 and Camelina biofuel show better fuel consumption characteristics than Avgas. This will not necessarily hold true at more highly loaded points and the stock timing and carburetor setup, since the low equivalent Octane numbers of the Biofuel and JP-8 will result in knock, torque loss, and subsequently higher bsfc’s if required bmep’s are to be achieved.

The other set of data points on Figure 66 show the same correlations of equivalence ratio to the load points when compared to the bsfc numbers at those load points. It is valuable to note increased equivalence ratios correlate to increased fuel consumption; as phi rises, the air/fuel ratio decreases.

**IV. Experimental Data – Emissions**

Emissions data was collected for all three fuels. Figure 67 shows Carbon Monoxide (CO) and Carbon Dioxide (CO2) emissions for all three fuels. Avgas tends to show higher phi’s (richer mixtures) with higher CO emissions and lower CO2 emissions compared to the other fuels. JP-8 and Camelina biofuel tend to show slightly lower phi’s (leaner than Avgas, yet still richer than stoichiometric). They also exhibit lower CO emissions than Avgas and higher CO2 emissions than Avgas.
Figure 67: CO and CO2 Emissions vs bmep and phi

Figure 68 and Figure 69 show hydrocarbon emissions data collected for both propellers. For the lower loaded prop, Avgas showed higher hydrocarbon (HC) output than JP-8 and Camelina biofuel. Camelina showed lower HC output in almost all cases as well. In the high load prop, HC emissions became quite high for Avgas (an order of magnitude greater than the other two fuels), with Camelina still exhibiting the lowest HC emissions. Note the different scales on the y-axis of the 17x10 and 18x12 plots.

Figure 68: 17x10 Hydrocarbons vs bmep and phi

Figure 69: 18x12 Hydrocarbons vs bmep and phi
Figure 69: 18x12 Hydrocarbons vs bmep and phi

Figure 70 shows NOx emissions data for both propeller load point sets. The emissions sampling unit detected both NO and NO2 concentrations in the exhaust and added them together labeled as Oxides of Nitrogen. Avgas, showed lower NOx emissions than the two heavy fuels, while Camelina tended to show the highest NOx emissions. NOx levels decrease as phi increases.

Figure 70: NOx vs bmep and phi

V. Uncertainty

Uncertainty calculations for torque and bsfc were accomplished using the Root-Sum-Squared (RSS) method. The error for each variable used in the RSS calculation is shown in Table 4. Uncertainty calculations are shown in Table 5: Low Condition Uncertainty (Q=3.81cc/min, N=1500rpm) Table 5 and Table 6.

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Symbol</th>
<th>Possible Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Torque (ft-lbf)</td>
<td>( \tau )</td>
<td>( \pm 0.5% ) of full scale (4.58ft-lbf)</td>
</tr>
<tr>
<td>Fuel Flow (cc/hr)</td>
<td>Q</td>
<td>( \pm 0.2% ) of measurement</td>
</tr>
<tr>
<td>Engine Speed (rpm)</td>
<td>N</td>
<td>( \pm 0.0015% ) of measurement</td>
</tr>
</tbody>
</table>
VI. Fuel Injection

The Fuji engine is setup with a carburetor to meter the fuel to the engine cylinder. While this is a proven method and works well in the field, it is not adequate to conduct laboratory experiments that require precise metering of fuel to maintain a specific equivalence ratio. Most automotive fuel injectors displace between 200-500 cubic centimeters per minute (cc/min), which is too high for this application, considering the measured average fuel flow rate at ~7000 RPM is 14cc/min in the test engine.

Two types of fuel injection will be implemented on the engine test stand. The first, port fuel injection, is the most common type of fuel injection today. It involves injection of a fine mist, relative to droplet sizes typical of a carburetor, into the intake air stream after the air throttle body and before the intake valve. Most fuel injectors utilize a small nozzle size and relatively high fuel pressure to obtain fine droplet sizes. Droplet sizes less than 50 microns are common for port fuel injectors. We will be selecting and characterizing a fuel injector that is capable of injecting heavy fuels and will obtain the smallest droplet sizes for the low flow rates as possible. Other experimentation the Air Force Research Lab has seen the use of Orbital air-assisted fuel injectors. These injectors first meter fuel into a pre-chamber, and then utilize pressurized air (approx. 6 atm), forced through the pre-chamber with the fuel, to inject this air-fuel mixture. These injectors have been shown to be capable of injecting heavy fuels at a very small droplet size (less than 10 microns). These injectors are in use right now as direct fuel injectors.

### Table 5: Low Condition Uncertainty (Q=3.81cc/min, N=1500rpm)

<table>
<thead>
<tr>
<th>τ</th>
<th>δτ</th>
<th>δBSFC</th>
</tr>
</thead>
<tbody>
<tr>
<td>ft-lbf</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>0.4</td>
<td>5.73</td>
<td>49.69</td>
</tr>
<tr>
<td>0.5</td>
<td>4.58</td>
<td>25.44</td>
</tr>
<tr>
<td>0.6</td>
<td>3.82</td>
<td>14.72</td>
</tr>
<tr>
<td>0.7</td>
<td>3.27</td>
<td>9.27</td>
</tr>
<tr>
<td>0.8</td>
<td>2.86</td>
<td>6.21</td>
</tr>
<tr>
<td>0.9</td>
<td>2.54</td>
<td>4.36</td>
</tr>
<tr>
<td>1</td>
<td>2.29</td>
<td>3.18</td>
</tr>
<tr>
<td>1.1</td>
<td>2.08</td>
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<td>1.2</td>
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<td>1.45</td>
</tr>
<tr>
<td>1.4</td>
<td>1.64</td>
<td>1.16</td>
</tr>
</tbody>
</table>

### Table 6: High Condition Uncertainty (Q=15.2cc/min, N=7500rpm)

<table>
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<th>δτ</th>
<th>δBSFC</th>
</tr>
</thead>
<tbody>
<tr>
<td>ft-lbf</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>0.4</td>
<td>5.73</td>
<td>39.64</td>
</tr>
<tr>
<td>0.5</td>
<td>4.58</td>
<td>20.30</td>
</tr>
<tr>
<td>0.6</td>
<td>3.82</td>
<td>11.75</td>
</tr>
<tr>
<td>0.7</td>
<td>3.27</td>
<td>7.40</td>
</tr>
<tr>
<td>0.8</td>
<td>2.86</td>
<td>4.96</td>
</tr>
<tr>
<td>0.9</td>
<td>2.54</td>
<td>3.48</td>
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<tr>
<td>1</td>
<td>2.29</td>
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<td>1.2</td>
<td>1.91</td>
<td>1.47</td>
</tr>
<tr>
<td>1.3</td>
<td>1.76</td>
<td>1.15</td>
</tr>
<tr>
<td>1.4</td>
<td>1.64</td>
<td>0.92</td>
</tr>
</tbody>
</table>
Direct fuel injection involves having the fuel injector built directly into the head of the engine so that it can inject fuel directly into the cylinder. Air is taken into the engine through the intake valve and the direct fuel injector injects the fuel right into the cylinder. Some systems inject fuel while air is being taken in. Others inject later, once the air is being compressed. Much flexibility in injection timing and metering can be achieved through direct fuel injection.

In the immediate future, port fuel injection will utilized as a way to precisely meter the amount of fuel that is taken into the engine; meaning the equivalence ratio can be controlled more accurately than with a carburetor at different engine speeds and loads. Direct injection is more complicated and time intensive and will be implemented at a later time.

VII. Algae Based Biodiesel

Research is currently being conducted on the economical production of biodiesel using solar algae reactors at the University of Dayton Research Institute under the direction of Dr. Sukh Sidhu. Different types of algae are being studied, as well as methods for extracting oil from the algae and converting that oil into usable biodiesel fuel. This study will go on to compare the biodiesels produced to commercially available D2 diesel, other biofuels, and JP-8 when viable amounts of fuel for testing are produced. Performance and emissions data will be collected.

VIII. Conclusions

With the current experimental setup it can be concluded that it is possible to achieve cruise condition propeller load points for the two heavy fuels: JP-8 and Camelina Biofuel. For higher load points, such as in takeoff conditions, it may be difficult to achieve such loadings without inducing knock with the current carburetor setup and ignition timing not being optimized for heavier fuels. This could be remedied with electronic fuel injection maintaining a more ideal stoichiometric air/fuel ratio and spark ignition timing retardation in the future.

Disregarding takeoff points, for cruise conditions, the heavy fuels showed lower fuel consumption. This is promising from a strategic single fuel point of view and from an environmental and domestic fuel point of view. If the engine could be optimized to run the heavier fuels at startup, no Avgas would be required for starting and warming up the engine. Emissions data shows that the heavy fuels emitted less CO and more CO2 than Avgas. The heavy fuels showed lower HC emissions than Avgas across the board; this was largely exacerbated by putting the engine under heavy load. Under heavy load, Avgas HC emissions were extremely high (growing by factors as large as 5), while heavy fuel HC emissions tended to double. NOx emissions data showed Avgas emitting lower amounts of NOx than the heavy fuels, while Camelina showed the highest NOx emissions.

If the aforementioned issues with cold-starting, fuel metering, and knocking can be overcome, it would seem that heavy fuels can provide better fuel consumption for the same power requirements, lower CO emissions, and lower hydrocarbon emissions. At the same time, higher CO2 and NOx emissions will likely be seen.
The Performance and Emissions Effects of Utilizing Heavy Fuels and Algae Based Biodiesel in a Port-Fuel-Injected Small Spark Ignition Internal Combustion Engine

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Abstract

The Department of Defense has a sustained interest in the universal use of heavy, low volatility hydrocarbon fuels for safety and logistical reasons. Large amounts of resources go into moving quantities of high octane, high volatility fuel that is used primarily for Remotely Piloted Aircraft (RPA’s). This paper discusses the results of the continuation of experimentation in utilizing heavy fuels such as JP-8 and D2 diesel to satisfy the necessary power requirements of a 33.5 cc internal combustion, 4-stroke, spark ignition engine that is originally designed to run on gasoline. The engine was originally mechanically carbureted for simplicity and cost reasons. For this experiment, a port fuel injection system (PFI) was developed to more precisely meter fuel to the engine. Furthermore, increased environmental concern with fossil fuels has spurred research into the use of bio-derived liquid fuels as a replacement. Bio-derived fuels used for this paper were algae-based biodiesel, blended with D2 diesel in a 20% algae, 80% diesel ratio (by volume) and a plant based biodiesel derived from the Camelina plant. Brake specific fuel consumption (BSFC) was observed as a measure of performance and emissions data was gathered for the fuels. Propeller load point sets were used as a reference. Field data for different propellers matched to this engine was used to decide what torque points and engine speeds to compare the fuels at. Camelina showed some of the best BSFC’s before engine knock became an issue, but also demonstrated poorer emissions characteristics than diesel and the algae blend. The algae blend showed slightly worse (higher) BSFC’s than straight diesel, but showed similar CO, CO2, and NOx emissions, with higher unburned hydrocarbon (UHC) emissions than diesel. Knock is still present with the heavier fuels, which is expected with
fuels that have a lower equivalent Octane Number (ON), but this can be mitigated using spark timing control and carefully mapping of the fuels. Preliminary results indicate heavy fuels and heavy biofuels can meet power requirements while lowering fuel consumption.

I. Introduction

Unmanned aerial vehicles (UAV’s) and micro air vehicles (MAV’s) are seeing a sustained rise in service hours in the Air Force. The military services have a real need for vehicles powered by an internal combustion engine that can run efficiently on heavy hydrocarbon fuels, especially JP-8, for cost reduction and logistical reasons. Furthermore, environmental concerns about the consumption of fossil fuels have prompted much research into using biofuels in internal combustion engines.

The purpose of this paper was to study the effects of using heavy hydrocarbon fuels, including D2 diesel, JP-8, plant derived and algae-based biodiesel, in a small spark ignition internal combustion engine and study the performance and emissions characteristics compared to traditional gasoline or avgas. The equivalence ratio leading to best system performance and fuel economy was a key parameter to be studied. Emissions data was collected for the different fuels. Baseline data was gathered using avgas, since it most closely resembled the fuel type the spark ignition engine was designed for.

The engine used in this study was a 33.5cc Fuji-Imvac BF-34EI 4-Stroke spark ignition IC engine, factory rated for gasoline. The instrumentation included a TSI 4021 mass air flow sensor for air mass flow rate, a Magtrol WB-65 Dynamometer for power/torque, Max Machinery flow meter for fuel flow rate, Hall effect sensor for spark encoding, crank position sensing and valve position sensing, and CTS Industries 500 series throttle position sensor for throttle position data and servo feedback. Information was presented on experimental setup of an engine test stand, including information on measuring air flow. Furthermore, emissions data concerning CO, CO2, NOx, and unburned hydrocarbons was gathered using an Enerac 700 emissions analyzer.

With equivalence ratio as a controlled variable, optimization at various RPM levels was plotted and compared for each fuel. To show direct comparisons between different fuels, plots of specific fuel consumption versus engine speed were generated. Also, brake mean effective pressure versus engine speed plots were generated with propeller load lines from different available propellers that were currently in use on this engine platform.

II. Experimental Setup

Test Stand

The engine test stand consists of a 7 foot long t-slot table on an extruded aluminum frame with a rigid steel mounting structure that bolts to the engine (see Figure 72). The test stand has access to facility power and compressed air. The crankshaft was coupled to the dynamometer via two rubber couplers on either side of a jackshaft that rode on a pillowblock bearing. Two rubber couplers were used to isolate angular vibrations from the dynamometer. Many coupling methods were attempted before a durable solution was reached. The unsteady torque intrinsic to a single cylinder, 4-stroke engine was failing other smaller couplers after only a few minutes of run time. Figure 71 shows the engine to dynamometer coupler arrangement.

127
Cooling air was added to both of the rubber couplers as well as the pillowblock bearing, as considerable heat was building in all of these components. A steel shield was constructed that came down around the coupler arrangement to contain debris in case of a failure. The stock electronic spark ignition box was still being used at the time of this testing. V-blocks, a proximity sensor, and a LabView interface were used to align the dyno shaft with the engine crank shaft.

**Air Flow**

To measure air flow a hot wire TSI 4021 Mass Air Flow meter was used. To reduce air flow oscillations present in the reading a large air damper was constructed using a 55 gallon drum. This drum was sealed and ducted to the engine air inlet. To further reduce air flow oscillations, the inside of the barrel was wrapped with polyurethane foam and the air had to pass through two one inch thick layers of foam. This dramatically reduced oscillations and was considered satisfactory for air flow measurement (Figure 73).
Fuel Flow

Fuel flow was measured using a Max Machinery flow meter (Figure 74). The flow meter was in-line after two separate fuel filters between the fuel tank and itself. The fuel tank was pressurized using nitrogen. This pressure was required for the fuel injection system.

![Figure 74: Pressurized Fuel Tank and Fuel Flowmeter](image)

Fuel Injection

The Fuji engine was originally set up with a carburetor to meter the fuel to the engine. While this is a proven method and is robust in the field, it is not adequate to conduct precise laboratory experiments. To see how engine efficiency was affected, the air/fuel mixture needed to be adjusted. While a limited range of equivalence ratios could be obtained with a carburetor, the accuracy was not sufficient for this experiment.

To enable precise metering of fuel flow, port fuel injection was implemented on the engine. Port fuel injection is the injection of a fine mist, relative to droplet sizes typical of a carburetor, into the intake air stream after the throttle body and before the intake valve. Most fuel injectors utilize a small nozzle size and high fuel pressure to obtain fine droplet sizes. Droplet sizes less than 50 microns are common for port fuel injectors. Most automotive fuel injectors displace between 200-500 cubic centimeters per second (cc/s), which was too high for this application, considering the measured average fuel flow rate at ~7000 RPM was 14 cc per second for the Fuji engine. A fuel injector was selected and characterized that was capable of injecting heavy fuels and obtained smaller, more ideal, droplet sizes for these low flow rates.

Other experimentation in the Small Engine Research Lab had seen the use of Orbital air-assisted fuel injectors. These injectors first meter fuel into a pre-chamber, and then they utilize pressurized air (approx. 75 psi), forced through the pre-chamber with the fuel, to inject this air-fuel mixture. These injectors have been shown to be capable of injecting heavy fuels at a very small droplet size (less than 10 microns). These injectors are in use right now as direct fuel injectors on some scooters and boat engines.

First, a test stand was set up to characterize fuel flow through the air-assisted injector. This stand consisted of a compressed air tank for the air pressure to the air-assisted injector and a nitrogen tank to pressurize a fuel tank for the pressurized fuel portion of the injector. Furthermore, there was a Berkeley signal generation box and power supplies. The same fuel flow meter used on the engine test stand was used to track fuel flow on the injector rig and a pulse counter was used to read out pulses from the flow meter. The rig also utilized a Malvern droplet size characterization unit that used laser beam diffraction to calculate Sauder Mean Diameter for the spray coming out of the injector.
Different fuel pressures were tested and fuel flow rates were recorded. Furthermore, droplet sizes were recorded for the different pressures and fuel types (avgas and JP-8 were used as test fuels). This test stand was used to verify the ability to meter fuel at low enough flow rates and an improvement in droplet size. Figure 75 shows some examples of measured droplet sizes.

After characterizing the injector a manifold was designed and machined in-house to couple the injector to the engine (see Figure 76). The manifold was designed to use the original carburetor as a throttle body and incorporates a 90 degree bend for the air flow path. Although not ideal from a flow efficiency standpoint, this allowed the rather large air-assisted injector to be mounted horizontally, pointed straight toward the intake valve. The manifold also contained a pressure trace for reading manifold pressure via a pressure transducer and a mounting point for a bracket that contained the throttle servo and throttle position sensor.

Controls, Sensors, and Data Acquisition

LabView software, developed in-house, was used to acquire data and control the engine during operation. Fuel injection was controlled using a PIC18 chip (Figure 77) and in-house developed software. The software utilized a phase signal taken from a Hall Effect sensor that passed through the valve cover and picked up the intake valve. A 36 tooth encoder (see Figure 77) and crank position sensor were used to sense engine position. This information is used by the software to decide when to fire the air and fuel solenoids on the injector. Solid state relays were used to fire the solenoids, as they had a sufficiently fast response time (see Figure 78).

One issue experienced with the use of a hobby-industries grade throttle servo actuator was considerable hysteresis in throttle butterfly angle opening and closing. This issue was addressed by developing feedback using a throttle position sensor (Figure 78) and the hysteresis between opening and closing was eliminated.
III. Fuel Performance Characterization

Avgas (Aviation 100 Low Lead gasoline) was used as a baseline fuel for this paper. Other fuels that were used include JP-8, D2 diesel, algae-derived biodiesel 20% blend with D2 diesel (algae B20), and Camelina biodiesel (which is plant-based).

Engine knock is a phenomenon that takes place when unstable combustion happens after the spark event. This causes uneven cylinder pressure, power loss, and rapid in-cylinder temperature rise. The Fuji stock ignition system placed the spark event at the relatively advanced time of 40 degrees before top dead center (BTDC). This enabled fuels with a higher tendency to ignite (a lower equivalent octane number) more time to knock. Heavy fuels, like D2 diesel, JP-8 and the biodiesels have low equivalent octane numbers (in the 20-40 RON range), which makes them tend to knock at higher engine loads.

The performance characteristics of the fuels were gathered by running the fuels in the engine at different engine speeds while maintaining an equivalence ratio (phi) of 1. Then, throttle opening was varied from 10% to 100% and averaged data points were taken for 8 seconds at each throttle opening. The engine torque, power, engine speed, fuel flow, air flow, throttle position, brake specific fuel consumption, phi, volumetric efficiency, and intake manifold absolute pressure were all recorded in data files and analyzed. During testing propeller load lines for different fielded propellers where used to analyze real world torque and engine speed requirements. A load line consisted of 6 engine speeds and the required torques (backed out from engine thrust) at those speeds. These load lines changed with different propeller diameters and pitches.

A performance map was generated using the air-assisted port fuel injection system (PFI) to show change in performance versus the original carbureted setup while running on JP-8. Figure 79 shows the difference between carbureted performance and PFI performance. It is important to note that much of the improvement comes from the increased ability to control the flow rate of fuel at all engine speeds. When carbureted data was taken, the carb had to be set at specific needle settings and left for the entire run, this means that if different engine speeds caused the carburetor performance to change (higher or lower equivalence ratio), nothing could be done. It was

![Figure 79: JP-8 Carburetion vs. PFI System](image-url)
possible to maintain a phi of 1 while utilizing the PFI system. Other improvement from the PFI system could result from better atomization of the fuel and more complete combustion.

One of the most effective ways to look at fuel performance is to make a plot of torque versus engine speed and create BSFC contour lines. This shows how efficiently the engine makes power over the whole range of the engine for different fuels. It will also show where it is best to operate the engine with different fuels to maximize range. The following figures show the results of maintaining a phi of 1 and varying throttle position for avgas and JP-8 (Figure 80).

![Avgas and JP-8 Performance at Phi=1](image)

When comparing these two maps a 10-20% decrease in BSFC was seen when using JP-8. In the time frame of this paper, complete contour maps were only able to be created for these two fuels. For the other fuels, including D2 diesel, algae B20 blend, and the Camelina biofuel, propeller load lines were used. Different cruise point torque values were recorded and then maximum takeoff torque values were attempted for each fuel. At these highly loaded points, the lower equivalent ON’s of the heavier fuels became apparent, as knock became a barrier to achieving the maximum torque that is possible with Avgas.

Performance maps for all of the fuels, including the other test fuels: D2 diesel, algae B20 and Camelina can be seen in Figure 81.
These figures show that Avgas, which most closely resembles the fuel this engine was designed for, only shows mediocre performance at these load levels compared to the other fuels. At the lowest engine speeds, D2 diesel and the algae B20 blend with D2 diesel showed some of the worst (highest) BSFC’s, but these two fuels matched the other fuels more closely at higher engine speeds. Camelina consistently showed some of the best BSFC’s across the range of engine loads and speeds. Of other interest is the fact that JP-8 often outperformed avgas at low and medium loads.

IV. Emissions Characterization

To gather emissions data an Enerac 700 portable emissions unit was used. The unit analyzed concentrations of oxygen (O2), Carbon dioxide (CO2), Carbon Monoxide (CO), Oxides of Nitrogen (NO and NO2), and unburned hydrocarbons (UHC’s). An emissions pick up tube was welded into the muffler of the engine and plumbed into an oil trap (to catch any unburned liquid fuel that might make it through in a cold start scenario), into a thermoelectric dryer, and finally into the unit, where concentrations were measured.

Figure 82 shows concentrations of CO and CO2 versus brake mean effective pressure (BMEP) for both propeller load sets. Emissions data was taken for diesel D2, algae B20, and Camelina. Here it can be seen that all three fuels produced CO2 levels within 3-4% of each other across most BMEP’s. For the larger propeller, the diesel and algae blend showed slightly higher CO emissions, compared to Camelina. For the smaller propeller, though, the CO emissions were higher for the Camelina.
Figure 83 shows emissions data for unburned hydrocarbons (UHC’s) and Oxides of Nitrogen (NOx) which includes NO and NO2, for both of the propeller load lines. NOx emissions can increase with increased in-cylinder temperature, which resulted from engine knock. The figures show that NOx emission for D2 diesel and Camelina tended to be higher by 12-68%, compared to the algae blend.

D2 diesel showed very low unburned hydrocarbon emissions for both propellers compared to the algae blend and Camelina biofuel (approx. 70-85% lower). It is not shown in there figures, but it is valuable to note that more smoke was witnessed in the engine exhaust with diesel.

V. Uncertainty

Uncertainty for the experimentation for this paper can be found in the torque measured by the dynamometer, the air flow measurement, which includes the TSI mass air flow sensor and the air injector air flow rate, and the equivalence ratio, which includes fuel flow and air flow measurements.

The air injector air flow rate is calculated based on an experiment in which the injector was run at different speeds without fuel (but with air pressure) through the air damper and then through the mass air flow meter. These air flows were recorded and a linear regression equation was made (Equation 6) that included a correction based on conserved mass that accounted for the liquid fuel entering and leaving the injector. This equation was for volumetric air flow, which was then separately translated to mass air flow using a density equation, the ambient pressure, and the ambient temperature.

\[
\dot{V}_{tot} = \dot{V}_{maf} + (4.3197 \times rpm + 1631.7) - \dot{V}_{fuel} \left( \frac{\rho_{fuel}}{\rho_{air}} \right)
\]

The equivalence ratio, phi, was calculated using the stoichiometric air-fuel ratio, calculated by balancing the combustion equation, based on the fuels average number of carbon,
hydrogen and sometimes oxygen atoms, then by multiplying by the mass flow of fuel over the mass flow of air (Equation 7).

\[ \phi = \frac{\dot{m}_f}{\text{FAR}_s \dot{m}_a} \] (7)

Next, uncertainty for equivalence ratio was calculated by using the Root-Sum-Squared method (RSS). Equation 8 shows the equation used and Equation 9 and Equation 10 show the partial derivatives within Equation 8.

\[ u_\phi = \pm \sqrt{\left( \frac{\dot{m}_f}{\phi} \frac{\partial \phi}{\partial \dot{m}_f} u_{\dot{m}_f} \right)^2 + \left( \frac{\dot{m}_a}{\phi} \frac{\partial \phi}{\partial \dot{m}_a} u_{\dot{m}_a} \right)^2} \] (8)

\[ \frac{\partial \phi}{\partial \dot{m}_f} = \frac{1}{\text{FAR}_s} \frac{1}{\dot{m}_a} \] (9)

\[ \frac{\partial \phi}{\partial \dot{m}_a} = \frac{\dot{m}_f}{\text{FAR}_s} \left( \frac{-1}{\dot{m}_a^2} \right) \] (10)

The error associated with fuel flow, air flow, and torque measurements are shown in Table 7. Tabulated results for a low condition and high condition show uncertainty results for equivalence ratio in Table 8. An example of the uncertainty in torque measurement across the typical torque measurement range for this engine is shown in Table 9.

### Table 7: Instrumentation Measurement Error (2)

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Symbol</th>
<th>Possible Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass Fuel Flow (lb/hr)</td>
<td>(\dot{m}_{ot_f})</td>
<td>±0.2% of reading</td>
</tr>
<tr>
<td>Mass Air Flow (lb/hr)</td>
<td>(\dot{m}_{ot_a})</td>
<td>±2.0% of reading</td>
</tr>
<tr>
<td>Torque (ft-lbf)</td>
<td>(\tau)</td>
<td>±0.5% of full scale (4.58 ft-lbf)</td>
</tr>
</tbody>
</table>

### Table 8: Uncertainty for Equivalence Ratio (Low and High)

<table>
<thead>
<tr>
<th>(\dot{m}_{ot_f}) (lb/hr)</th>
<th>(\dot{m}_{ot_a}) (lb/hr)</th>
<th>(\phi) (A/F)</th>
<th>(u_\phi)</th>
<th>(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low Condition (2500 rpm)</td>
<td>0.2678</td>
<td>3.856</td>
<td>1.022</td>
<td>14.55</td>
</tr>
<tr>
<td>High Condition (7500 rpm)</td>
<td>0.9406</td>
<td>13.427</td>
<td>1.028</td>
<td>14.55</td>
</tr>
</tbody>
</table>
VI. Conclusions

Experimentation showed that it was much easier to control equivalence ratio through the use of an electronic fuel injection system compared to a mechanical carburetion system. BSFC was shown to be slightly decreased at lower loads for most of the heavy fuels, compared to avgas. Once knock became a factor, though, the advantages of the heavy fuels diminished as spark timing had to be retarded to mitigate knock, bringing the peak pressure further away from the ideal position and bringing BSFC back up. It would seem that through careful selection of cruise and takeoff points and correct propeller size and pitch selection, heavy fuels could be implemented to achieve the required torque values to operate this engine platform while reducing BSFC. Camelina biofuel shows particular promise in the reduction of BSFC.

Emissions characteristics of the fuels showed that Camelina emitted more CO and CO2 in all cases except for the 18x12 inch propeller, in which Camelina emitted slightly less CO than diesel and the algae blend with diesel. The algae blend showed promise in that it emitted slightly less NOx than Camelina and D2 diesel alone, while straight D2 diesel emitted the lowest traces of unburned hydrocarbons.

VII. Recommendations and Future Work

More thorough emissions maps will be created in the future for all of the test fuels that include 10% throttle position increments across the engine’s entire speed range. Since slightly lower volumetric efficiencies were witnessed with the new fuel injection system compared to the carbureted system, a new injector manifold will be designed. There is currently a 90 degree bend in the manifold air flow path that is most likely the cause of the decreased volumetric efficiency compared to the straight-through carburetor design. Also, a redesign of the coupler setup between the engine and dynamometer will be completed, as many couplers failed due to vibrations during the gathering of data for this paper.

Table 9: Uncertainty in Torque Measurement

<table>
<thead>
<tr>
<th>$\tau$ (ft-lbf)</th>
<th>$u_\tau$ (%)</th>
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</thead>
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<tr>
<td>0.3</td>
<td>7.63</td>
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<tr>
<td>0.4</td>
<td>5.73</td>
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<tr>
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<td>4.58</td>
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<td>0.6</td>
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