**EFFECTS OF FUEL NOZZLE CONDITION UPON GAS TURBINE
COMBUSTION CHAMBER EXIT TEMPERATURE PROFILES**

**EFFETS DE CONDITION D'AJUTAGE DU CARBURANT SUR LE
CHAMPS DE TEMPÉRATURE DE SORTIE DE CHAMBRE DE
COMBUSTION DE TURBINE A GAZ**

by

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B.Eng.**

A thesis submitted to
the Faculty of Graduate Studies and Research
in partial fulfillment of
the requirements for the degree of
Master of Applied Science
Ottawa-Carleton Institute for Mechanical and Aerospace Engineering

Department of Mechanical and Aerospace Engineering
Carleton University
Ottawa, Ontario
Canada

May 2009©

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ISBN: 978-0-494-51971-4
Our file Notre référence
ISBN: 978-0-494-51971-4

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To my support system,

Harley, Lita, and Rick

Acknowledgements

I would like to acknowledge the tremendous support received from within the Mechanical Engineering Department at the Royal Military College of Canada. I extend my gratitude to my Co-supervisor Dr. W.D.E Allan, as well as to Dr. M. LaViolette, and Mr. G. Pucher for all of their advice and guidance. Their knowledge and expertise were instrumental to the completion of this work.

The instrumentation and guidance provided by Nexum Research Corporation, more specifically, Dr. R. Underhill, was crucial during the examination of the fuel nozzle condition. His vast experience and abilities ensured that the numerous challenges associated with the installation of the test equipment and data analysis were overcome with minimal effort required by the author.

The support provided by M. Guimont, J. Fleming, B. McEwen, M. Gatien, D. Babcock, and D. Fisher is greatly appreciated. Without their technical assistance and expertise, the numerous technical challenges encountered with the combustion chamber sector rig would not have been overcome.

The funding and support of the Canadian Forces T56-A-15 Community, including Captain C. Prentis, Captain L. Holland, and Mr. P. Finch was greatly appreciated. Their assistance was essential in obtaining the specimens used in the experimentation process and their interest in this field of study was very motivational.

Much appreciation goes to Captain Clayton Kotzer, Mr. L. Bennet, and Mr. R. Andrews of the Royal Military College of Canada and to Mr. S. Guy and Mr. S. Fabbro of Standard Aero Ltd. for their technical advice and counselling provided during all phases of this research, particularly during the experimental set-up and data analysis portions.

I would like to extend my gratitude to Dr. Donald Gauthier, formally of Carleton University, who stimulated my interest in gas-turbine combustion and inspired me to complete my degree in this field of study. Finally, I would like to thank Dr. M.I. Yaras from Carleton University for accepting to be my Supervisor on such short notice and for providing me with the guidance and inspiration to challenge myself academically and produce something that I can truly be proud of.

Abstract

Bishop, Kristen Marie. M.A.Sc. Carleton University, May, 2009. *Effects of Fuel Nozzle Condition on Gas Turbine Combustion Chamber Exit Temperature Profiles*. Supervised by Dr. M.I. Yaras, Ph.D., and Dr. W.D.E. Allan, DPhil, the Royal Military College of Canada.

The turbine of any engine is responsible for power generation and any damages to this component can result in significant degradations in performance. Highly non-uniform temperature distributions at the exit of a combustion chamber can lead to thermal stressing of the nozzle guide vanes and first stage turbine blades causing erosion or cracks requiring extensive maintenance and increased operational down-time. To investigate the effects of fuel nozzle condition on exit temperature profiles, an optical patternator was used to examine spray patterns for differences in spray cone angle, symmetry, and fuel streaks. An ambient pressure and temperature rig constructed with Allison/Rolls Royce T56-A-15 gas turbine engine components was used to capture the exit temperature distributions. Rig tests simulated representative engine operating conditions by matching Mach Number, equivalence ratio, and droplet size (D_{32}).

The main conclusion of this research was that small deviations in the fuel nozzle spray pattern from a nominal distribution correlated to an increase in pattern factor. Reductions in the spray cone angle, increased variations in spray roundness and increased streak intensities resulted in a degradation of the mixing processes, which created large regions of very high temperature core flow and smaller regions of cooler temperatures within the combustion chamber exit plane, subjecting the nozzle guide vanes and turbine blades to hotter temperatures at their roots and tips. The spray cone angle had the most measureable influence, with a 10% decrease resulting in a 15% rise in pattern factor. The effects of spray roundness and streak intensity had slightly less influence on pattern factor. The results of this research were compared to past studies and the pattern factors were representative of the typical values expected for a T56-A-15 combustion chamber. Recommendations for future testing of fuel nozzle condition are made in this work.

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Nomenclature

Symbol	Description	Units
$\mathcal{A}$	Pre-exponential Factor	—
A	Area	m^2
B	Transfer Number	—
C	Coefficient	—
$\mathcal{D}$	Diffusion Coefficient	m^2/s
D	Diameter	m
E	Energy	J
G	Gibbs Free Energy	J/K
H	Enthalpy	$\text{J}/(\text{kg K})$
K	Nozzle Coefficient, in Equation 2.4	—
K	Equilibrium Constant, in Equations A.10 and A.11	—
K	Evaporation Constant, in Equations A.40 and A.41	m^2/s
L	Length	m
N	Quantity of a Substance	mol
P	Pressure	Pa
R	Ideal Gas Constant	$\text{J}/(\text{kg K})$
S	Entropy	$\text{J}/(\text{kg K})$
T	Temperature	K
U	Internal Energy	J/kg
V	Volume	m^3
$\mathcal{V}$	Voltage	V
Y	Mass Fraction	—
c	Specific Heat	$\text{J}/(\text{kg K})$
d	Diameter	m
f	Fuel to Air Ratio	—
h	Heat Transfer Coefficient	$\text{W}/(\text{m}^2 \text{K})$
j	Diffusion Mass Flux	$\text{mol}/(\text{m}^2 \text{s})$
k	Reaction Rate Constant, in Equations A.21 and A.22	—
k	Thermal Conductivity, in Equation A.34	$\text{W}/(\text{m K})$
m	Mass	kg
n	Quantity of a Substance	mol
q	Specific Heat	J/kg
r	Radius	m
t	Time	s
u	Uncertainty	—
α	Thermal Diffusivity	m^2/s
ε	Emissivity	—
ϕ	Equivalence Ratio	—
θ	Non-Dimensional Temperature, in Equation 4.2	—
θ	Spray Cone Angle, in Equation 2.11 and 4.1	Degree

Symbol	Description	Units
λ	Evaporation Constant	m^2/s
μ	Dynamic Viscosity	$\text{kg}/(\text{m s})$
ν	Kinematic Viscosity	m^2/s
ρ	Density	kg/m^3
σ	Surface Tension	kg/s^2

Subscript	Description
A	Activation
D	Discharge
L	Liner
P	Port
a	Air
atm	Atmospheric
c	Combustion
cr	Chemical Reaction
d	Diffusion
e	Evaporation
f	Fuel
fg	Formation
g	Gas
i	Inlet
liq	Liquid
max	Maximum
p	Pressure
r	Radius
ref	Reference
s	Surface
st	Stoichiometric
°	Initial
1	Final
3	Diffuser Exit
32	Sauter Mean Diameter
4	Combustion Chamber Exit

Superscript	Description	
-	Molar Basis	
·	Time Rate of Change	
°	Reference State	1/s
"	Per Unit Area	
'''	Per Unit Volume	$1/\text{m}^2$
^	Combined	$1/\text{m}^3$

Chapter 1

Introduction

1.1 Background

The development of technology is often fuelled by the demands of war to possess more advanced equipment with greater capabilities than those of one's enemies. It is no surprise then that the fundamental development of the jet engine occurred during World War II with parallel developments by an English Royal Air Force Officer, Sir Frank Whittle and a German engineer, Hans van Ohain. After the war, the jet engine saw many improvements and by the mid-twentieth century, the gas turbine was firmly established as the propulsion system of choice for aeronautical applications. However, early advances in gas turbines were limited by the development of a working combustion chamber capable of high combustion efficiency, smooth and reliable ignition, low pressure loss, flame stability and durability [1]. The process of combustion encompasses the diverse disciplines of fluid mechanics, heat and mass transport processes and chemical kinetics. Although the combustion chamber is a seemingly uncomplicated component, the complex nature of combustion makes the behaviour inside this component extremely difficult to understand and to predict. Today, there exist extensive studies on the effects of fuel spray, droplet evaporation, mixing and cooling and their influence on the internal aerodynamics of a combustion chamber and the resulting effects that will be experienced by downstream components.

Before discussing the complex inner workings of a combustion chamber, it is important to describe the gas turbine engine in which it is operating. Over the past few decades, the range of applications of the gas turbine engine has broadened substantially to include not only the propulsion system of small, mid-size and large aircraft, but it has also been widely used in marine and land transportation and in industrial applications like electrical power generation. The diverse configurations of gas turbines are all derived from its most basic form, consisting of a compressor, a combustion chamber and a turbine. The basic operation begins with air being drawn into the compressor, which is an array of fan-like airfoils set in rows and usually arranged in pairs, one rotating and one

stationary, which forms one compressor stage. The rotating airfoils, known as *blades* or *rotors*, are used to pressurize the fluid by redirecting and accelerating the flow. The stationary airfoils, known as *vanes* or *stators*, diffuse and redirect the fluid flow, preparing it for the rotor blades of the next stage. Typically, each stage gets progressively smaller moving away from the air inlet and generally speaking, the compressor increases the pressure of the air by reducing the volume that it passes through keeping the flow from slowing down as the density of the fluid increases. This high pressure air is then delivered to the combustion chamber where the fuel is added. This creates a fuel-air mixture that is highly turbulent and when ignited, thermal energy is released. The resulting hot efflux gases exit the combustion chamber through the nozzle guide vanes to the turbine blades, driving the turbine, from which some energy is used to drive the compressor. This is the simplest of configuration of a gas turbine engine and is depicted in Figure 1.1.

Figure 1.1: Basic Configuration of a Gas Turbine Engine [2]

Depending on the application requirements, multiple spools consisting of first and second stage compressors and turbines can be added along with intercoolers between the compressors to increase the power output and thermal efficiency of the gas turbine [1]. In aerospace applications, common configurations include turboprops and turbofans which use multi-spool arrangements consisting of high- and low-pressure compressors and turbines. In a turboprop gas turbine engine, part of the power generated by the turbine is used to drive the compressor and the remaining power drives a shaft, which in turn drives a propeller through a reduction gear box. Turbofans are very similar except that the remaining power generated by the turbine is used to drive a fan. Regardless of the configuration, the overall objective of a gas turbine is to take a low pressure gas, increase

its pressure and temperature such that energy or work can be extracted either in the form of compressed air, thrust, or shaft power, which can then be used to power aircraft, ships, trains, electrical generators and so on.

1.2 Gas-Turbine Combustion Section

Combustion chamber type and geometry are largely determined by the specifications of the engine, along with space availability. All chambers typically incorporate the same basic components consisting of a casing, diffuser, liner, fuel injector, igniter and discharge nozzle. Due to the diverse array of applications of gas turbine engines, a combustion chamber must be capable of satisfying a wide range of basic requirements, which have been outlined by Lefebvre [1] as:

- high combustion efficiency in all operating envelopes;
- reliable and smooth ignition at all operating conditions, including relight capabilities after flameouts;
- wide flame stability limits;
- low pressure loss across the whole system;
- a combustion chamber exit temperature distribution that is consistent with the turbine blades' and nozzle guide vanes' operating capabilities;
- minimum emission of smoke and other gaseous pollutants;
- minimum cost, size and weight;
- ease of manufacturing; and
- good reliability and maintainability.

Regardless of the type of engine, low fuel consumption, long operating life and low pollutant emissions are important factors. The operating life of a gas turbine engine is ultimately determined by the material structural limits of the components, and it is for this reason that this research project will be focused on examining a few key factors that affect the combustion chamber exit temperature profile and potentially cause excessive efflux gas temperatures that can cause serious damage to the hot section components.

1.2.1 Combustion Chamber Geometry

Many different combustion chamber configurations exist, including *tubular* (or *can*) and *annular* combustors each with their own set of advantages and limitations. A common type of combustor for use in aircraft engines is a combination of the two, known as the *can-annular* or *tubo-annular* combustor. The overall purpose of using this configuration is to combine the mechanical integrity of tubular combustors with the compactness of an annular combustor [1]. The can-annular configuration incorporates a group of tubular liners, typically six, uniformly spaced inside a single annular casing as seen in Figure 1.2.

The inner casing assembly acts as both a support and heat shield. The combustion chambers are completely surrounded by the airflow entering the liners through the various holes. Upon entering the combustion chamber, the air is mixed with the atomized fuel sprayed from the fuel nozzles and the mixture is then ignited by igniter plugs.

Figure 1.2: Can-annular Combustion Chamber [3]

Although the design of combustion chambers and the method in which fuel is added may vary significantly, most conventional combustor chambers consist of a diffuser and three distinct zones where different amounts of air are introduced to complete the combustion process. A conventional combustion chamber can be seen in Figure 1.3.

Figure 1.3: Primary Components of a Conventional Combustion Chamber [1]

The air exiting the compressor and entering the combustion chamber is often at speeds upwards of 150 m/s (500 ft/s), which is too high for combustion because it will cause flame extinction [4]. Therefore, the airflow must be decelerated using a diffuser, which also serves to recover dynamic pressure and provide stable and smooth airflow entry conditions to the combustion chamber. The gas turbine engine operates on a Brayton cycle, where air is compressed isentropically, combustion occurs under constant pressure and the efflux gases expand isentropically over the turbine back to the starting air pressure [2]. Combustion under constant pressure means the combustion chamber is open to flow in and out, and loss of pressure during the process of combustion must be minimized to prevent power loss and increases in fuel consumption. However, a certain amount of pressure loss is required to ensure adequate mixing of the air and fuel. A major design requirement for combustion chambers is minimum pressure loss while maximizing mixing [1]. The pressure drop across the combustor (ΔP_{comb}) is comprised of cold pressure losses, which arises due to frictional losses in the diffuser and liner

(ΔP_{cold}) and the pressure loss that occurs when heat is added to a flowing gas (ΔP_{hot}), according to Equation 1.1:

$$\Delta P_{comb} = \Delta P_{cold} + \Delta P_{hot} \quad (1.1)$$

Unlike the pressure drop across the liner, which is manifested as turbulence, promoting better mixing and more efficient combustion, the cold pressure losses in the diffuser do not provide any useful contributions, and therefore, it is important to design a combustion chamber with a diffuser that is as efficient as possible. An ideal combustion chamber would have zero pressure loss in the diffuser and ΔP_{cold} would be entirely due to the liner pressure differential [1]. This has not been achieved as of yet and typical values of cold pressure loss in modern combustion chambers are between 2.5 and 5 percent of the combustor inlet pressure [2], with the majority due to losses in the diffuser. Generally speaking, the overall purpose of the diffuser is to efficiently reduce the compressor outlet velocity to provide a uniform flow pattern at acceptable velocity levels to the liner.

The burning rate of aviation fuel at normal mixture ratios is only a few feet per second [2], therefore any ignited fuel will be blown away by the much faster air stream. As a result, a region of low airflow velocity has to be created within the combustion chamber for the flame to remain lit over the wide range of engine operating conditions. In most combustion chambers, the air is slowed down by the diffuser and is introduced through swirlers and other holes in the combustion liner, where it mixes with the fuel droplets. The overall mass fuel-to-air ratio (f) can be as low as 1:100 [4], while the stoichiometric fuel-to-air ratio (f_{st}) is typically around 1:15. The equivalence ratio (ϕ), which is defined as the actual fuel-to-air ratio divided by the stoichiometric fuel-to-air ratio, is given by Equation 1.2.

$$\phi = \frac{f}{f_{st}} \quad (1.2)$$

This value is typically above unity in the first section of the combustor, which is known as the *primary zone*, to ensure stable combustion [1]. According to Lefebvre [1], the major functions of the primary zone are to secure a stable flame and to provide

adequate time for fuel droplet evaporation and sufficient temperature and turbulence for complete combustion of the fuel-air mixture entering the chamber. Therefore, only about 20 percent of the overall airflow is introduced around the jet of fuel in the primary zone allowing the fuel and air mixture to be rapidly brought up to ignition temperature [2]. The air entering from the holes in the combustion liner and the air from the swirl vanes interact and create a region of low velocity recirculating air, which stabilizes and anchors the flame. This recirculation zone allows the fuel-air mixture to have adequate residence time for combustion and it also promotes quicker burning of newly injected fuel droplets by rapidly bringing them to ignition temperature [1]. The temperature of the gases released during the combustion process can be upwards of 2000 K, which is higher than the material limits for the hot section components, therefore the temperature must be reduced by gradually introducing more air into the combustion chamber [2].

The purpose of the *intermediate* or *secondary zone* is to continue and complete the combustion process. Due to the high combustion temperatures in the primary zone, significant concentrations of carbon monoxide and nitrogen oxides are present in the efflux gases as a result of the dissociation of the products of combustion. Therefore, cooling air is introduced at precise locations in the secondary zone to reduce the flame temperature and allow the completion of the combustion process, greatly reducing the concentration of any carbon monoxide and unburnt hydrocarbons [1].

The combustion process is completed before reaching the *dilution zone*, where the remaining air is introduced and reduces the temperature of the hot efflux gas, providing a uniform temperature distribution in the outlet exhaust stream, yielding a profile consistent with the turbine design limits. Typically, between 20 to 40 percent of the combustion section entry airflow is fed into the dilution zone to achieve the required distribution of temperatures [2]. The uniformity of the combustor exit temperature profile is dependent on the degree of mixing between the dilution air and the exhaust gases, which is dependent on the location of the holes in the liner and the length of dilution zone. According to Lefebvre [1], the size and shape of the holes are selected to optimize the penetration of the air jets and mixing with the efflux gas stream. The length available for mixing, i.e. the length of the dilution zone, greatly improves the amount of mixing. However, if the diameter is too large, the combustion rates will be slower. Therefore, it is

desirable to have long and relatively slender combustion liners and typical length-to-diameter ratios are in the range of 1.5 to 1.8 [1]. Ideally, the temperature distribution or *pattern factor* produced at the exit of the dilution zone would be such that the temperatures experienced by the most critical points of the nozzle guide vanes and the turbine blades are minimized, which are typically located at the blade roots and tips [2]. This will reduce overheating of the blades and prevent damage. The primary means of maintaining a uniform temperature distribution is through the precise placement and sizing of the dilution holes and the geometry of the dilution zone, ensuring that the turbine blades will not experience *hot streaks*. Hot streaks are the result of localized fuel rich zones usually generated by a faulty fuel nozzle, which causes the fuel to burn beyond the intended primary and secondary zones, creating streaks of hot combustion gases that can cause excessive damage to downstream components.

1.2.2 Test Article Combustion Section

Components of a combustion system of the Allison/Rolls Royce T56-A-15 turboprop engine were available for experimentation in this research. The combustion section of this engine is comprised of six combustion chambers, which are circumferentially positioned around the main drive shaft. Each chamber is fixed in place at the inlet by a fuel nozzle centered within a flared fitting in the dome and at the exhaust exit by the discharge nozzle, commonly called the *fishmouth*, engaging the turbine inlet guide vanes. The axial position of the chambers is maintained by the placement of igniter plugs in two of the chambers and dummy igniters in the remaining four chambers. The combustion chambers are interconnected by six cross-over tubes, which provide flame transfer during engine start-up. A schematic of a T56-A-15 Series III combustion chamber complete with an estimate of the air flow proportions [5] can be seen in Figure 1.4.

To ensure the combustion chambers are protected from the convective and radiative heat transfer loads produced during the combustion process, cooling air is introduced at various stages along its length. Rizk *et al.* [5] predicted that the airflow distribution was divided such that approximately 24% was used for combustion, 20% for dilution and 56% for cooling. The portion of the airflow used for combustion enters the dome through eight radial rows of equally spaced holes and creates a recirculating airflow in the

primary zone. Film cooling is provided to the combustor wall through baffles along the combustor barrel and corrugation strips. The airflow used for dilution enters the dilution zone through holes of varying size that are specifically placed to promote mixing, completion of the combustion process and a uniform temperature distribution.

Figure 1.4: T56-A-15 Series III Combustion Chamber [5]

1.2.3 Fuel Injection

When examining the performance of gas turbine combustors, emphasis must be placed on the processes of fuel atomization and evaporation. The volatility of current aviation fuel is relatively low and cannot produce sufficient vapours for ignition and combustion unless it first undergoes atomization to produce a large number of smaller-sized droplets,

which evaporate more quickly than larger droplets [1]. Since the fuel is injected from isolated sources, spray characteristics like droplet spatial distribution, droplet size and spray cone angle, dictate how well the spray will mix with combustor airflows, which is a very important factor to combustion efficiency. If the fuel injector operation is hindered or restricted by any means, non-ideal fuel-to-air ratio variations can result, which may in turn lead to accumulation of soot, pollutant formation, reduced liner durability and life and hot streaks that could cause damage to the turbine. It is therefore critical that fuel injectors achieve optimal atomization and concentric sprays to avoid these adverse effects on combustor performance and emissions.

The primary method of achieving fuel atomization is by applying a pressure to the liquid to set it in motion and pass it through a small orifice. The liquid emerges as a thin film, which is broken up into smaller filaments due to disruptive aerodynamic forces [1] and eventually assumes droplet form, due to the surface tension of the liquid itself. In the majority of cases, liquid fuels cannot be properly burned without atomization due to an insufficient amount of liquid surface area exposed to the surrounding air. There are numerous different types of devices, variously known as *atomizers*, *injectors*, or *nozzles*, which perform this essential task. In addition to atomizing the fuel, injectors also disperse the spray in a specific pattern and direction to achieve good combustor performance over a wide range of engine operating conditions.

There are three main types of nozzles used in aeronautical gas turbine engines: airblast, simplex and dual-orifice pressure swirl atomizers, which are shown in Figure 1.5. An airblast atomizer flows low pressure fuel over an edge located in a stream of high velocity air. The fuel is atomized by the jet of air once it flows over the edge and then the small droplets are carried into the combustion zone by the airstream. The main advantage of this type of atomizer is that it provides a high volume of airflow, which produces a uniform fuel distribution and a consistent temperature profile, ultimately resulting in more complete combustion, less exhaust smoke and cooler liner walls. However, there are several disadvantages associated with this type of nozzle. At conditions with low air velocity, the atomization quality is very poor and at conditions with high air velocity, the flame stability limits are quite narrow [1].

A pressure-swirl nozzle takes a fuel under pressure and passes it through a specially designed orifice. It is the energy of the fuel itself that is used to achieve atomization. Generally speaking, the high pressure of the fuel is converted to kinetic energy by passing it through a small orifice. Therefore, the fuel exiting the orifice will have a higher velocity relative to that of the surrounding air. As the relative motion between the surface of the fuel spray and the surrounding air increases, the aerodynamic interactions or shear forces cause irregularities in the smooth surface of the fuel jet. These irregularities become amplified, eventually detaching from the surface and form ligaments that are shaped into droplets [1]. There are many different types of pressure-swirl atomizers. The most commonly used in the aviation industry are the simplex and dual-orifice nozzles.

Figure 1.5: Atomizer Schematics: (a) Airblast, (b) Simplex Pressure Swirl and (c) Dual-orifice Pressure Swirl [1]

The simplex nozzle contains a swirl chamber, which emits fuel through tangential slots, often referred to as a *swirlplate*, causing a swirling motion. The centrifugal forces created by the swirling motion of the fuel cause the spray to fan out over a wider angle, forming a hollow conical spray [6]. The dual-orifice nozzle contains two simplex nozzles, arranged in two different circuits for fuel flow and discharge, which are labelled *primary* or *pilot* and *secondary* or *main*. The nozzle contains swirlplates for both the primary and secondary circuits and an internal switching valve which opens only the primary orifice at low fuel flow rates. For high fuel flow rates, the valve opens the secondary orifice in addition to the primary. The advantages of these types of nozzles are their simple design, good mechanical reliability and good flame stability over a wide range of operating

conditions [1]. The main drawback is the susceptibility of the small passages or orifices to become blocked by contaminants in the fuel.

1.2.4 T56-A-15 Turboprop Fuel Nozzle

The Allison/Rolls Royce T56-A-15 turboprop engine utilizes an air-assist dual-orifice pressure-swirl atomizer as seen in Figure 1.6. This nozzle contains an air shroud designed to direct the airflow across the face of the nozzle to help prevent carbon build-up and nozzle blockage. A main drawback of this nozzle is that poor atomization occurs around the opening of the secondary orifice. This is due to the fact that the primary orifice remains at the delivery pressure, but the pressure through the secondary circuit is dependant on the pressure required to open the internal switching valve, which is identified in detail B of Figure 1.6 as part number 8.

According to Canadian Forces Inspection Manual for the T56-A-15 engine [7], the internal switching valve opens at a fuel delivery pressure of 448 kPa (65 psi) with a tolerance of ± 35 kPa (5 psi). Therefore, the atomization quality of the secondary orifice will be poor, producing large droplets that take longer to evaporate, thus adversely affecting the combustion efficiency until the transition to an adequate pressure for atomization is completed.

1.3 Thesis Objective

All Canadian Forces T56-A-15 turboprop fuel nozzles are currently being tested for serviceability by the Aerospace and Telecommunications Engineering Support Squadron (ATESS) located in Trenton Ontario. Their testing involves a visual inspection of the spray cone angle at fuel delivery pressures ranging from 345 kPa (50 psi) to 1724 kPa (250 psi) [7]. Serviceability is determined based on the concentricity of the spray pattern, as well as the cone angle. The current inspection interval for this nozzle is every 950 flying hours; however, ATESS has been finding nozzles that are failing at less than 350 flying hours. The cause of these failures is unknown, but it is suspected that it is due to the internal condition of the fuel nozzles and potential blockage due to accumulation of coke on the swirlplates.

Number	Part or Assembly
1	Air shroud
2	Body assembly
3	Lockscrew
4	Filter
5	Orifice
6	Valve guide assembly
7	Valve guide
8	Valve
9	Locknut
10	Seat
11	Spring
12	Stop
13	Cage
14	Spray tip assembly
15	Spacer
16	Spring
17	Director
18	Inner tip
19	Outer tip
20	Body
21	Filter
22	Spring
23	Holder

Figure 1.6: T56-A-15 Fuel Nozzle [8]

Studies sponsored by the *Allison Engine Company* [9] have been conducted on the T56-A-427 turboprop engine, which uses a similar nozzle as the T56-A-15 turboprop engine. These studies have indicated that at least one swirlplate will become blocked in 30% of the engines operated after 200 hours, 45% of the engines after 400 hours and 88% of the engines after 800 hours. Recent studies by *NAVAIR* [10, 11] have concluded that most nozzles require inspection or maintenance after only 10% of the expected life of 3000 flying hours set by the US Navy. It has been concluded that this blockage could potentially cause uneven spray distribution and hot streaks, which could lead to thermal stressing of the downstream components, exceeding of design specifications and to erosion of the nozzle guide vanes and turbine blades due to the break off of carbon.

The current Canadian Forces fleet of Allison/Rolls Royce T56-A-15 turboprop engines have experienced an increase in the number of degraded nozzle guide vanes and turbine blades since the fleet switched from JP-4 to JP-8 aviation fuel, which is known to contribute to significant coking on the fuel nozzles [12] in most engines. Currently, the Canadian Air Force are finding and replacing on average 200 degraded nozzle guide vanes annually during scheduled and unscheduled maintenance, leading to additional maintenance costs in excess of \$400 000. The fleet of T56-A-15 engines are aging and can only expect to see increased maintenance costs, putting a strain on already limited human and financial resources.

The main objective of this research was to investigate the relationship between the physical condition of a substantial number of T56-A-15 fuel nozzles with varying service lives and combustion chamber exit temperature distributions. The fuel spray cone angle and droplet spray distribution are important characteristics of these fuel nozzles and have been shown to influence flame stability, combustion efficiency, liner wall temperature and emissions. Experimentation was performed to examine the spray distributions of several nozzles each with varying service life using an optical patternator at a fuel mass flow rate for a matched T56-A-15 engine operating condition. Further experimentation of these nozzles was conducted using a combustion chamber sector rig to obtain a comparison between the consistency of the temperature distributions and the fuel nozzle spray characteristics. These tests were repeated after the nozzles had been cycled through the fuel nozzle condition testing and cleaning procedures conducted at ATESS to

compare the spray distributions and downstream effects after they had been deemed serviceable and would be cycled back into the Canadian T56-A-15 fleet. Peripheral objectives of this research were to carry out upgrades to the combustion chamber sector rig to increase the air mass flow available, to prolong thermocouple life and to automate the experimentation process.

1.4 Thesis Layout

This thesis has been organized to provide a progression from the most general topics relating to combustion to a specific study correlating the physical spray characteristics of a widely used fuel nozzle to the overall combustor exit temperature distribution. Chapter 2 provides a brief narrative of the background information relating to fuel nozzle spray characteristics and the combustion chamber exit temperature distribution. The description of the experimental setup for both the optical patternator and the combustion chamber sector rig, as well as, the calibration of the associated instruments is presented in Chapter 3. Chapter 4 describes the experimental procedure and presents the results. The correlation and interpretation of the data are discussed in Chapter 5 and the conclusions and recommendations are provided in Chapter 6.

Chapter 2

Gas Turbine Combustion, Fuel Spray Characteristics and Pattern Factor

2.1 Introduction

Combustion is essentially the study of chemically-reacting flows and the means by which they transform the energy stored in chemical bonds to heat that can be used in various ways [13]. A fundamental interest in combustion and its control has developed over the past century due to its extensive use in industrial and aerospace applications like electrical power generation and aerospace propulsion. An efficient operation of combustion chambers depends on the knowledge of the oxidation reactions and heat transfer between the fuel and the oxidizer. It is also dependent on the heat transfer between the combustion products and the chamber walls, with the material design temperature being a critical factor. In gas turbine engines, the fuel is introduced into the combustion chamber through the fuel nozzle, which delivers a fuel spray with varying droplet sizes depending on the delivery pressure, fuel properties, nozzle type and nozzle geometry. These fuel droplets evaporate to form a highly combustible vapour, which mixes with the surrounding air and then is ignited by the introduction of a spark. Therefore, having a thorough understanding of fuel atomization is extremely important to understanding the combustion process. The combustion in a gas turbine engine is affected by numerous physical processes, the most fundamental being chemical kinetics, thermochemistry, heat and mass transfer and fluid mechanics [13]. Appendix A provides a review of several of these physical processes including chemical reactions and associated reaction rates, droplet evaporation and combustion, along with heat and mass transfer occurring during combustion.

In this chapter, the process of fuel atomization and the mixing processes within the combustion chamber are examined. In addition, the resulting temperature distribution leaving the combustion chamber, also referred to as the *pattern factor*, is discussed. The review of fuel atomization will provide a summary of droplet sizing and distribution, nozzle characteristics and coking, along with the methods of spray analysis.

2.2 Fuel Spray and Atomization

Most gas turbine combustors utilize hydrocarbon fuels with relatively low volatility, which will not burn in bulk since there is insufficient liquid surface exposed to the surrounding air to allow for the production of enough vapour to facilitate easy ignition and efficient combustion. Therefore, the bulk liquid must be broken up into smaller droplets, greatly increasing the surface area of the liquid exposed to the air, allowing for better mixing and reduced evaporation times. This process is known as *atomization* and is accomplished by forcing fuel under pressure through an orifice, normally housed in a fuel nozzle. As the relative motion between the surface of the fuel spray and the surrounding air increases, the aerodynamic interactions or shear forces cause irregularities in the smooth surface of the fuel jet. These irregularities become amplified, eventually detaching from the surface and form ligaments that are shaped into droplets [1]. The characteristics of the spray formed are dependent on both the type of nozzle used and the physical properties of the fuel, the most important being its density, viscosity and surface tension.

2.2.1 Performance Characteristics of Pressure-Swirl Atomizers

As described in Section 1.2.3, a dual-orifice pressure-swirl atomizer passes fuel, depending on the delivery pressure, through either the primary or both the primary and secondary tangentially inclined slots or swirl chambers, causing the fuel to spin around, and produces an air-cored vortex, as described by Ortman and Lefebvre [14]. At the exit of the swirl chambers, there is a discharge orifice, through which the rotating fluid is forced forward, creating a spray with both tangential and axial velocity components. Due to the fuel density and the circumferential and axial momentum of the fuel, a hollow conical sheet (with an internal air core) is produced, as shown in Figure 2.1. This sheet eventually breaks up into ligaments due the viscous interaction of the surrounding air with the fuel jet surface and the fuel surface tension causes the ligaments to form into droplets of various sizes. The process of disintegration and droplet breakup is comprehensively described in Figure 2.2.

Figure 2.1: Spray Produced by a Pressure-Swirl Atomizer [15]

Figure 2.2: Breakup of a Fuel Jet [1]

The internal flow characteristics of a pressure-swirl atomizer are of primary importance when examining the spray distribution produced and the resulting spray characteristics; cone angle, droplet spatial distribution and mean droplet size.

2.2.1.1 Flow Number

The *Flow Number* (FN) is described by Lefebvre [1], as the ratio of the nozzle throughput to the square root of the fuel delivery pressure differential. The most commonly used equations for the FN of pressure atomizers are:

$$FN_{USA} = \frac{\dot{m}_f}{\Delta P^{0.5}} \quad \left[\frac{lb/hr}{psi^{0.5}} \right] \quad (2.1)$$

$$FN_{UK} = \frac{\dot{V}_f}{\Delta P^{0.5}} \quad \left[\frac{UK \text{ gals/hr}}{psi^{0.5}} \right] \quad (2.2)$$

where $\dot{m}_f$ and $\dot{V}_f$ are fuel mass and volumetric flow rates, respectively. From these general equations, Lefebvre [1] derived an expression for the FN in SI Units that included the fuel density and is given as:

$$FN = \frac{\dot{m}_f}{\Delta P_f^{0.5} \rho_f^{0.5}} \quad \left[\frac{kg/s}{Pa^{0.5} (kg/m^3)^{0.5}} \equiv m^2 \right] \quad (2.3)$$

The FN of a nozzle is an important internal characteristic because it provides an indication of the effective flow area of the nozzle, which affects the size of the air core, fuel film thickness and the discharge axial and tangential velocities of the film. The liquid film thickness is essentially dependant on the air core formed in the swirl chamber. The liquid film thickness is of importance to combustion because it influences the fuel spray angle, length of time required for the jet to breakup into droplets and spray droplet-size distribution. Kim *et al.* [15] examined the effects of nozzle internal geometry on the film thickness and air core variations. They found that as the orifice length was increased, the film thickness increased along with the diameter of the air core. They concluded that the lengths and diameters of the orifice and swirl chambers have a significant impact on the film thickness and mass flow rate. Rizk and Lefebvre [16] had similar findings and noted

that by increasing the discharge orifice diameter and inlet port area, the fuel flow rate would increase and increase the film thickness. Additionally, they found that increasing the swirl chamber diameter actually reduced the fuel flow rate due to the increased swirling motion of the fuel causing a larger air core. They also noted increasing the pressure drop produced a thinner film and better atomization. This agrees with the observations made by Joyce [17], in that the air core diameter tends to increase as the fuel delivery pressure increases, which has a direct effect on the film thickness and the discharge coefficient of this type of nozzle.

2.2.1.2 Discharge Coefficient

The *discharge coefficient*, C_D , for any nozzle is directly related to geometrical properties of the nozzle through the dimensionless nozzle constant, K , which is defined as:

$$K = \frac{A_p}{d_s d_o} \quad (2.4)$$

where A_p represents the swirl chamber port area and d_s and d_o are the swirl chamber and discharge orifice diameters, respectively. The primary internal characteristic of the fuel nozzle that affects the coefficient of discharge is the size of the air core because it determines the effective flow area at the discharge orifice. According to Lefebvre [1], early research led to an understanding that a relationship existed between the atomizer dimensions and the resulting air core area and was given by Equation 2.5:

$$C_D = 1.17 \left[\frac{(1-X)^3}{(1+X)} \right]^{0.5} \quad (2.5)$$

where

$$X = \frac{\text{air core area}}{\text{discharge orifice area}} = \frac{A_a}{A_o} = \frac{(d_o - 2t)^2}{d_o^2} \quad (2.6)$$

and t is the fuel film thickness in the discharge orifice. The following relationship was derived from large quantities of data obtained during the research of simplex nozzles by Rizk and Lefebvre [16]:

$$C_D = 0.35K^{0.5} \left(\frac{d_s}{d_o} \right)^{0.25} \quad (2.7)$$

They noted that C_D is relatively low for pressure-swirl atomizers because of the presence of the air core in the swirl chamber, which tends to increase in diameter as the delivery pressure increases to a certain limit. This limit on the growth of the air core diameter is due to viscous or frictional losses, thus limiting the radial velocity of the fluid and preventing further growth. More recent studies by Datta and Som [18] have found similar results in that increasing the flow rate at the lower range does have the same effect of decreasing the discharge coefficient, but this parameter becomes independent of flow rate at higher pressures.

2.2.2 Droplet Sizing (Sauter Mean Diameter)

As mentioned above, the properties of the fuel that have the most significant impact on the droplet size and distribution of the spray and, thus the combustion efficiency of the engine are density, surface tension and viscosity. According to Wang and Lefebvre [19], research has shown that for the range of densities of aviation fuel, the influence of this property on drop size is quite small and can be considered to have little to no effect on atomization. The surface tension of the fuel is the force that causes the surface of one portion of the fluid to be attracted to another portion of surface within the fluid. This is an important parameter for atomization because it resists the formation of new surface area. Therefore, the higher the surface tension, the more difficult it is to break-up the bulk fluid into smaller droplets. The most important fuel property in the atomization process is viscosity. Viscosity is a measure of the resistance of the fuel to deformation due to imposed stresses, such as shear stress. Thus, fluids with a higher viscosity will be more resistant to mixing with air, thus delaying the onset of atomization and leading to larger droplet sizes [1].

In addition to the effects of the fuel properties on atomization, the drop size and distribution are also affected by the temperature of the fuel, the delivery pressure, mass flow rate and temperature and pressure of the surrounding air. These properties were studied extensively by Wang and Lefebvre [19], with the following observations:

- increasing the fuel temperature resulted in better atomization (reduction of the mean droplet size) and increased combustion efficiency. This was also observed by Jasuja [20] and Heitor and Whitelaw [21];
- increasing the injection pressure differential resulted in reduced droplet sizes;
- increasing the flow rate by either changing the injection pressure or the flow number (geometry) of the nozzle reduced the mean droplet size;
- increasing the air temperature produced smaller droplets and better combustion efficiency. This was also observed by Heitor and Whitelaw [21], Dodge and Biaglow [22] and Chen *et al.* [23]; and
- increasing the ambient pressure above atmospheric resulted in an increase in droplet sizes. This was equally noted by Tabata *et al.* [24].

Based on the number of studies that exist on the effects of droplet size on gas turbine combustion, it is an important parameter that must be examined when researching the influence of nozzle condition on combustion efficiency. The diameter of the droplets produced by an atomizer range in size and a common practice is to determine an average droplet size, which can be used when determining the time required for evaporation and combustion. To obtain an average droplet size, the poly-sized spray must be converted to an averaged mono-sized spray. The average droplet size is determined by conserving several properties between the two sprays, including the droplet diameter, number of droplets, droplet volume and surface area [25]. For nozzle spray and combustion applications, the overall spray volume, which produces the heat addition and the overall surface area, which effects the evaporation time, are typically conserved. Spherical droplets are assumed and the overall spray volume and surface area are used to determine to the mass-averaged diameter of the droplets using the generalized equation shown in Equation 2.8. This average diameter is typically referred to as the *Sauter mean diameter*

or D_{32} [1]. The D_{32} is defined as the ratio of the sum total of volume to that of surface area of the measured droplets:

$$D_{32} = \frac{\int_{D_{min}}^{D_{max}} aD^{(\alpha+3)} \exp(-bD^\beta) dD}{\int_{D_{min}}^{D_{max}} aD^{(\alpha+2)} \exp(-bD^\beta) dD} \quad (2.8)$$

where a , b , α and β are constants determined experimentally [25]. The most popular empirical correlation is based on the basic relationships between the mean droplet diameter and the fuel properties, delivery pressure and mass flow rate and is given by:

$$D_{32} \propto \text{constant } \sigma^a \nu^b \dot{m}_f^c \Delta P_f^d \quad (2.9)$$

where σ and ν are the surface tension and kinematic viscosity of the fluid, respectively, $\dot{m}_f$ is the fuel mass flow rate and ΔP_f is the fuel injection pressure. The subscript f in all of the equations within this Section represents fuel. Numerous researchers over the years have provided different values for a , b , c and d based on various derivations of experimental results for different nozzle types. A summary of the development of these equations and the experimentally determined equation for several researchers is adequately presented by Wang and Lefebvre [19]. Extensive research has concluded that one of the most commonly used correlations [16, 23, 26, 27] for the determination of the mean droplet size from a pressure-swirl atomizer was developed by Lefebvre [25], which is:

$$D_{32} = 2.25 \sigma^{0.25} \mu_f^{0.25} \dot{m}_f^{0.25} \Delta P_f^{-0.5} \rho_{air}^{-0.25} \quad (2.10)$$

The effect of mean droplet size on the combustion efficiency of gas turbine engines has been widely studied with varying observations. It was found during numerous studies [6, 14, 16, 19, 28] conducted by Lefebvre and several other co-authors, that inadequate atomization would greatly influence the evaporation rates of the fuel, thus affecting ignition, flame stability, combustion efficiency and emissions. However, more recently, it has been suggested by Odgers *et al.* [29] and Krestchmer and Odgers [30] that the calculated evaporation and burning time for a typical fuel is actually estimated to be longer than the time available for mixing within an actual combustion chamber and thus,

the droplet size would have a minimal effect on combustion efficiency. Krestchmer and Odgers [30] determined that most recent studies over-estimate the evaporation time and they explain this situation by stating that in a real combustor, the droplets are actually broken up into finer droplets, which become entrained in the recirculating flow, allowing them enough time for evaporation. To prove their theory, they conducted a series of experiments involving a variety of fuels in several different combustion chambers over a range of inlet pressures and temperatures. Their results indicated that there was no measureable effect of fuel droplet size on combustion efficiency for droplets ranging from 14 to 150 μm [29]. The majority of droplets produced by pressure swirl atomizers will remain well below 150 μm . The overall observations of Odgers *et al.* [29] and Krestchmer and Odgers [30] are of great significance to this research indicating that the actual diameter of the droplets and their evaporation rate in a real gas turbine combustor are rarely deciding factors in the overall combustion efficiency.

2.2.3 Droplet Distribution

In addition to ensuring adequate atomization to achieve desirable droplet sizes, the actual distribution or spray produced by a nozzle should be symmetrical and uniform to ensure a homogeneous temperature distribution of the efflux gases, high combustion efficiency and low pollutant emissions [25].

2.2.3.1 Spray Cone Angle

One means of examining the spray pattern produced by a nozzle is to study the spray cone angle, which when increased, can improve atomization by increasing the exposure of the droplets to the surrounding air [1]. Studies have shown that the nozzle spray cone angle can greatly affect combustion efficiency, flame length and liner wall temperatures [31, 32]. The spray typically produced by a nozzle is in the form of a hollow cone with all the fuel concentrated at the periphery. Since the spray has curved boundaries, as shown in Figure 2.3, the spray cone angle is often measured by drawing two straight lines starting from the discharge orifice and extending to the edge of the spray.

Figure 2.3: Spray Cone Angle from a Pressure-Swirl Atomizer [32]

Through experimental research, Rizk and Lefebvre [16] developed a correlation for the spray cone angle from a pressure-swirl atomizer:

$$2\theta = 6K^{-0.15} \left(\frac{\Delta P_f d_o^2 \rho_f}{\mu_f^2} \right)^{0.11} \quad (2.11)$$

where K is the dimensionless nozzle constant defined in Equation 2.4, d_o is the orifice diameter and μ_f is the dynamic viscosity of the fuel. For a dual-orifice pressure swirl atomizer, the atomization from the secondary nozzle is poor at lower pressures, because the bulk of the fluid is still passing through the primary port and the majority of the differential pressure is being used to open the valve. Therefore, there is a range of fuel flow rates, beginning at the point where the valve opens, that the droplet sizes are sufficiently large. To address this situation, the primary spray cone angle is often designed to be slightly wider than the secondary spray cone angle, allowing the two sprays to combine almost immediately after leaving the nozzle [1]. The spray cone angle can be affected by a number of factors, which can be seen when examining Equation 2.11. Based on the research contained in [6, 14-20, 32], Table 2.1 provides a summary of

the various effects of fuel properties, ambient air pressure, fuel injection pressure and nozzle dimensions on the spray cone angle.

Table 2.1: Summary of Parameter Effects on Spray Cone Angle

Parameter	Description	Change in Property	Effect on Spray Cone Angle
ρ_f	Fuel Pressure	↑	↑
μ_f	Fuel Viscosity	↑	↓
ΔP_f	Fuel Injection Pressure	↑	↑↓
$P_{ambient}$	Ambient Pressure	↑	↓

The fuel nozzle differential pressure (ΔP_f) has a conflicting effect on the spray cone angle. Several studies [6, 16] have found that by increasing the fuel injection pressure at atmospheric conditions, the spray cone angle also increased. Other studies [14, 32] suggest that the cone angle initially increases with increases in ΔP_f at atmospheric pressure, but then starts to contract past a certain point. Joyce [17] predicted that this phenomenon was the result of the viscous losses imposing an upper limit to tangential velocity, causing the hollow air core to decrease and thus the spray angle to decrease as more fuel is forced through the discharge orifice. Observations have also been made regarding the effect of the ambient pressure on the spray cone angle. Studies by Wang and Lefebvre [19] and Jasuja [20] have shown that increases in ambient pressure cause a reduction in the spray cone angle. This contraction of the cone angle has been related to the presence of a low pressure region within the spray volume contained by the hollow cone as compared to the surrounding ambient pressure on outside of the spray. De Corso and Kemeny [32] concluded that this pressure difference acts on the spray and causes the cone angle to collapse inward. Ortman and Lefebvre [14] found that above a certain ambient pressure, the cone angle would no longer contract, but would remain constant. It was reported by Chen and Lefebvre [6] that the spray cone angle has a significant effect on gas turbine performance because it greatly impacts the combustion efficiency, influencing the ignition stability, lean blowout limits, temperature distributions of the efflux gases and soot and smoke concentrations.

2.2.3.2 Radial and Circumferential Fuel Distribution

The spray distribution produced by a nozzle can be influenced by numerous parameters like imperfections in the orifices, plugged or blocked flow passages, scratched surface finishes and nozzle damage. Any non-uniformity in the spray can lead to regions of the primary zone that are either too lean or too rich, which can result in slow burning rates leading to unburnt hydrocarbons or extreme flame radiation, increases in liner wall temperatures, excessive soot concentrations and disproportionate temperature distribution of the efflux gases [19]. The visible spray cone angle does provide a crude indication of spray symmetry, but it cannot show if there are radial and circumferential increases or decreases in the fuel distribution within the total spray. *Patternation* is the term most commonly used to describe the process of measuring the spray distribution from a fuel nozzle and *patternators* are the instruments used in these processes. Many different methods exist and they will be discussed in subsequent sections. The radial fuel distribution is often characterized by the measurement of the volume of fuel at specific angular locations around the spray axis. A spray distribution curve can be obtained by plotting these values against one another. This curve can be reduced to one value, denoted the *effective* or *equivalent spray angle* [1]. Figure 2.4 shows the typical measurement setup for the radial distribution of a spray with the corresponding variables.

The derivation of the *equivalent spray angle* is provided in several papers [33] and is given as:

$$\theta_{eff} = \theta_L + \theta_R \quad (2.12)$$

$$\theta_L \text{ (or } \theta_R) = \frac{\sum y \theta \sin \theta}{\sum y \sin \theta} \quad (2.13)$$

The subscripts *L* and *R* represent the left and right lobes of the spray distribution, θ is the angle that corresponds to the location of the measuring device for either the left or right lobe. The effects of the fuel and air properties shown in Table 2.1 also apply to θ_{eff} .

nozzle face, or by the circulation of the fuel through other systems to act as a coolant, the lighter components in the fuel are released and eventually small particles of carbon are left behind [13]. Coke can form on liner walls or within the passages, slots and on the face of the fuel nozzle and cause blockages leading to non-uniformities in the fuel spray distribution. This can result in streaks or distortions in the spray symmetry within the combustion chamber, which can subsequently lead to deterioration of the combustor liner walls, turbine inlet thermocouples, inlet guide vanes and turbine blades. Since the fuel is used as a coolant elsewhere in the T56-A-15 engine [34], it is often heated prior to reaching the nozzle, with fuel temperatures as high as 440 K [9]. Therefore, nozzle fouling by the deposition of carbon is quite common. Due to increases in the damage detected in the turbine section of the T56-A-15 engine, the National Research Council of Canada conducted a series of tests on fouled fuel nozzles to determine their effect on the downstream components [35]. They noted that fouled fuel nozzles with a streaking flow pattern produced visible hot spots in thermal exhaust patterns. They also noted deviations in the fuel flow rates of the fouled nozzles when compared with serviceable nozzles. Studies by Williamson [9] and *NAVAIR* [10] found similar results. They observed that the majority of fuel nozzles removed from in-service engines had significant coke deposition, resulting in blocked slots in the swirlplates, leading to streaks in the fuel spray pattern. Some of the key design features to reduce gum and coke formation within the nozzle are summarized by Lefebvre [1] and are:

- reduce the surface roughness within the fuel nozzle passages to prevent deposition;
- reduce the flow passage area to increase the fuel velocity, reducing the fuel residence time;
- add insulated air gaps to act as thermal barriers; and
- replace metal components with ceramic components to minimize wetted-wall temperature.

Studies are still on-going to rectify this problem for pressure-swirl atomizers and as indicated in [9, 10], inspection cycles of these nozzles need to be re-examined to prevent possible damage to the combustor and downstream components.

2.2.5 Methods of Spray Analysis

The injection of fuel into a combustor is a very rapid process with atomization, evaporation and mixing occurring in only a few milliseconds. Therefore, the resultant atomized fuel spray is not easily characterized. The most important parameters that can be used for gas turbine fuel nozzle spray measurements are the mean droplet size, spray cone angle and fuel mass distribution. Different methods have been developed in recent years to examine each of these key spray characteristics. One of the easiest devices used to measure the fuel mass distribution and the spray cone angle is the *mechanical patternator*. However, this device is not capable of determining the mean droplet size. Due to the limitations of mechanical patternation, more advanced methods that are capable of capturing additional characteristics of the spray were developed. *Optical patternation*, which is much less labour intensive, is capable of capturing the spray distribution, droplet-size distribution and cone angle without disturbing the spray or air flow. *Phase Doppler Particle Analyzers* (PDPA) can also be used to determine the individual droplet sizes, along with the droplet velocities. When used in conjunction with one of the above techniques, a full characterization of the spray is possible.

2.2.5.1 Mechanical Patternation

One of the oldest methods of obtaining a spray pattern is by mechanical patternation. This technique involves the collection of the sprayed fuel into containers or tubes that have been divided into sections. The nozzle is typically centered above the bins or tubes at a predetermined height and the fuel is sprayed into the collection bins at a preselected fuel and air flow rate. The fuel accumulates in the sampling tubes until one is nearly full, at which point the fuel is turned off and the volume of each tube is measured. When determining the radial distribution of the spray, the tubes are typically orientated radially equidistant from the origin of the spray as can be seen in Figure 2.4 [1]. Many researchers like Ortman and Lefebvre [17], Rizk *et al.* [5], and Cohen and Rosjord [31] used radial patternators to study fuel nozzle sprays under various conditions and configurations.

Their work led to a better understanding of the effects of air and fuel properties, injection pressures and nozzle geometry on the characteristics of a spray.

The circumferential patterning of the spray involves a similar set-up, but instead of tubes being spaced radially, a circular vessel that has been divided into a number of pie-shaped sectors is placed below the nozzle. Each sectorized compartment collects the fuel and typically drains into a sampling tube for measurement. The sector measurements are then averaged and normalized against the mean. Circumferential irregularities can be found by determining the standard deviation of the normalized values. Figure 2.5 provides a schematic of a typical circumferential patternator.

The circumferential uniformity of the spray is often characterized by a parameter termed the *Sector Uniformity Index* (SUI), which is found by measuring the fuel mass in each sector and then comparing that value to the theoretical value that would be found from a perfect nozzle [36]. The equation for the *SUI* of a particular nozzle is given as:

$$SUI = n \sqrt{\sum_{s=1}^n \left[\frac{\left(X_s - \frac{100}{n} \right)^2}{n-1} \right]} \quad (2.14)$$

where n is the number of sectors and X_s is the percentage of fuel in each sector.

Figure 2.5: Circumferential Patternator (adapted from [23])

Underhill *et al.* [36] indicated that acceptable values of *SUI* up to 30 have been found to exist in typical nozzles. Other studies, like Chen *et al.* [23] have used similar correlations to quantify the extent of spray uniformity.

Although mechanical patterning does provide a relatively good average representation of the spray cone angle and spray distribution, there are several disadvantages to this method. This is an obtrusive method that interferes with both the spray and its associated airflow, which can cause inaccuracies when attempting to truly characterize the spray. Hung *et al.* [37] recommended that mechanical patterning not be used to characterize spray due to the evaporation of very small drops that will never be collected and measured in the patternator. Additionally, this method is only capable of collecting time-averaged data and cannot examine instantaneous variations in the spray. These disadvantages led researchers to begin looking for alternative methods of analyzing the characteristics of a spray, which resulted in the development of optical patterning.

2.2.5.2 Optical Patterning

With the advent of the laser, numerous different optical techniques have been created to measure and visualize flows in a non-obtrusive manner. Some common methods include *Particle Image Velocimetry* (PIV), *planar laser-induced fluorescence* (PLIF) and *planar Mie-scattering*. PIV is a method that involves the use of multiple pulses of planar or 2-D laser light sheet to illuminate particles in the flow field and the images are recorded using multiple-exposure photographic methods. This method is most often used to obtain velocity measurements within the flow; it is not widely used for the study of fuel nozzle sprays.

PLIF involves the addition of a chemical to the fluid flow, which when mixed with the fluid and illuminated with a light source will fluoresce. Most PLIF setups involve the use of a pulsed laser as the light source, which is diffracted into a sheet by an arrangement of lenses. The light then hits the particles causing them to fluoresce, which is captured and intensified by *Charged-Coupled Device* (CCD) cameras, which are also used in PIV, that record the images of the fluorescence intensity field. CCD cameras are designed to convert optical brightness into electrical amplitude signals and then reproduce the image using the electric signals [36]. Recent studies [38-40] have shown the effectiveness of this method in characterizing fuel sprays.

An optical patternation method that has seen numerous advancements in the past decade is the *Planar Mie-scattering* technique, also referred to as *planar laser scattering* (PLS) and it is based on the *Lorenz-Mie theory of scattering*. Gouesbet *et al.* [41] provides an excellent generalization of this theory, placing emphasis on one of the most important results; that the light scattered by a spherical particle is unpredictably dispersed and is dependent on refractive index, droplet diameter and the laser light characteristic. Using this generalization, it can be shown that the scattered light intensity is an oscillating function and these oscillations, due to interference between diffracted, refracted and reflected rays emanating from the particles, are the basis of PLS. PLS systems are commonly referred to as *Optical Spray Pattern Analyzers* (OPSA) and consist of a laser with cylindrical optics to produce a fan shaped light sheet geometry, an array of lenses to focus the light sheet onto a set of photodiodes and a CCD camera to capture images of the scattered light. An optical spray pattern analyzer developed by Nexum Research Corporation, RMC and Queen's University, is shown in Figure 2.6.

Figure 2.6: Schematic Representation of an OPSA [36]

Although this system does have a great advantage over mechanical patternators, in that it is non-obtrusive and has much higher temporal resolution, it requires image corrections for light intensity variations and geometry before any images can be captured and analyzed. These corrections, along with a more detailed description, will be provided in subsequent sections. Recent studies by Underhill *et al.* [36, 42-44] have shown the

effectiveness of this system in determining spray cone angle, droplet diameters, *SUI* and radial and circumferential distributions under varying operating conditions.

2.2.5.3 Phase-Doppler Particle Analysis

Phase-Doppler Particle Analyzer (PDPA) is another type of instrument that can be used to measure the droplet-size distribution of fuel sprays. Phase-Doppler systems perform measurements at a specified point within a spray. The most common set-up consists of PDPA transmitting optic (laser), a beam splitter and a PDPA receiving optic (detector). A beam splitter separates the laser beam into two beams of equal intensity and focuses them at an intersection point, which acts as the control volume for the measurements. When a droplet passes through this volume, light is scattered and the refracted light signal is collected by a detector, which converts the phase-shift information from the scattered light into droplet diameters. Before converting the information directly into an average droplet diameter (D_{32}), the detector interprets and classifies the data by running each point through a series of logic tests, and if it passes, then it is converted to a drop diameter and placed into a collective database. At the completion of testing, the database produces a drop-size distribution for that specific location in the spray [45]. Typically, the control volume is positioned at several sequential locations in the spray, starting from the spray centre and moving radially outwards until the volume is no longer within the spray. This results in a spatial variation in spray droplet-size distributions. Figures 2.7 and 2.8 show a standard PDPA setup and droplet-size distribution. This method of examining droplet-size distributions in sprays has been widely used over the past few decades. Ghaffarpour and Chehroudi [46] used PDPA to examine droplet-size distributions and velocities being produced by a pressure-swirl atomizer in a gas turbine combustor with and without combustion. Ohkubo *et al.* [47] conducted a study of the effect of inlet air temperature, air pressure, air velocity and swirl number on droplet evaporation using a PDPA, while Kim and Lee [48] utilized this method to examine the transient atomization of a spray. It is clear that PDPA is a widely used and trusted method within the combustion research community.

Figure 2.7: Phase-Doppler Particle Analyzer Set-up (adapted from [45])

Figure 2.8: Droplet-Size Histogram [1]

2.3 Combustion Chamber Mixing

The aerodynamic processes involved in ensuring efficient combustion and prolonged component lives are extremely complex, but vital to the design and performance of gas turbine engines. The main objective when designing a gas turbine combustor is to ensure adequate mixing throughout the liner and a stable flow pattern that minimizes pressure loss across the combustor [1].

Research in the aerodynamics of gas turbine combustors began in the late 1950's by using cylindrical ducts designed to emulate tubular combustion chambers. A thorough account of the progress in this field of research is given in [1] and only the main observations will be presented here. Wheatley *et al.* [34] noted that the size and shape of the holes, the angle of air injection and the air jet-to-mainstream momentum flux ratio all greatly influenced the mixing process. More recently, studies have been conducted by using *computational-fluid dynamics* (CFD) to predict mixing. Talpallikar *et al.* [49] utilized CFD and experimental analysis to examine the effects of altering the air jet-to-mainstream momentum flux ratio and slot aspect ratio on jet mixing and emissions of a Rich-burn/Quick-mix/Lean-burn (RQL) combustor. One of the main issues when trying to predict the mixing process within combustion chambers is the presence of highly turbulent flows. Numerous CFD models, like the one developed by Celik *et al.* [50] have been created to attempt to predict turbulent flows in gas turbine combustors, but it was not until recently that research was conducted in this field, involving the mixing of highly turbulent air flows with liquid reacting sprays. Kojima and Nguyen [51] observed turbulent mixing in a direct-injection combustor using single-shot spontaneous Raman spectroscopy, which relies on light scattering techniques and concluded that the proper design of the air admission holes is crucial to ensuring efficient combustion and prolonged component life.

The mixing within a combustion chamber is directly related to the airflow distribution through the liner and air admission holes. Typically, one third of the air enters the primary zone, through both the cooling corrugations and air admission holes. To ensure complete combustion and thus high combustion efficiency, there must be sufficient residence time for the chemical reactions to complete and this is accomplished by limiting the flow velocity entering the primary zone to about 25 m/s [52]. The flow

entering from the combustion chamber dome is introduced in a swirling motion, which creates a recirculation zone designed to promote flame stability. Additional air is then introduced, typically one third of the main airflow, into the secondary zone, which lowers the gas temperatures, allowing the chemical reactions of incomplete combustion from the primary zone to occur. The remainder of the air enters the dilution zone through both the cooling corrugation and holes, diluting the hot combustion gases and creating an acceptable exit temperature profile. Admitting too little air will result in poor mixing, higher temperatures, excessive flame radiation and increased emission and smoke, while admitting too much air could lead to quenching of the chemical reactions, resulting in incomplete combustion and increased CO emissions [52]. Therefore, the design of gas turbine combustors focuses largely on ensuring the proper fuel-air mixture ratio, flame temperature, and residence time in both the primary and secondary zones, and the proper dilution of the efflux gases to ensure a uniform combustion chamber exit temperature distribution that will not cause damage to the turbine blades due to localized high-temperature gases.

With the growing concerns about the impact of emissions on the environment, extensive research has been conducted on the T56-A-15 combustion liner to gain a better understanding of why this particular engine produces such excessive smoke. The location of the air admission holes and cooling corrugations, along with the airflow distribution for both the T56-A-15 Series II and III combustion liner are shown in Figures 2.9 and 2.10.

The airflow distributions for these liners were obtained from measured areas using the coefficients of discharge for this combustion system. The main differences between these liners are the variations in the air hole geometry and exhaust gas outlets. Testing completed by Skidmore and Schofield [53] concluded that the Series II liner emitted less smoke and was more efficient over a range of operating conditions than the Series III liner. In separate studies, Skidmore [54, 55] suggested that the reason for the difference in the smoke emissions between the two liners was due to the design of the primary zone and the amount of air that was introduced into the liner through the first set of circumferential holes. He noted that there was a decrease from 6.4% to 3.4% of the total airflow from the Series II to the Series III liner, respectively. This decrease in airflow

would have the effect of decreasing the mixing in the primary zone and produce fuel rich regions within the flame leading to soot formation.

Figure 2.9: T56 Series II Combustion Liner [53]

Figure 2.10: T56 Series III Combustion Liner [53]

Skidmore and Schofield [53] also noted that the manufacturing of the Series III liner allowed for a maximum possible variation in the flow area from approximately 4820 mm² to 5140 mm². By conducting extensive airflow tests on liners with different flow areas, they showed that these variations resulted in a combustion chamber exit temperature variation in excess of 90 K.

2.4 Combustion Liner Temperature

In the past two decades, there has been an increase in typical operating temperatures for most systems using combustion to generate shaft power or thrust, thus resulting in the need for better component durability and longer life [56]. It was the desire to increase the overall operating life of combustion liners that led to study of the heat transfer processes that govern liner-wall temperatures and thus determine the life of the component. The need to increase the useful life also sparked numerous studies that involved methods of reducing the heat flux to the liner wall to achieve a longer liner life.

Generally speaking, the liner is heated by convection and radiation from the hot gases inside the container and it is cooled by radiation to the outer casing and by convection to the surrounding air. In equilibrium conditions, the liner temperature is such that the internal and external heat fluxes are equal at any point and the loss of heat by conduction along the liner is comparatively small and often neglected [1]. Figure 2.11 provides a summary of the heat transfer processes occurring at the liner wall.

In steady-state conditions, the rate of heat transfer or heat flux into a wall element must be balanced by the rate of heat transfer out [1]. Liner walls are usually very thin, therefore it is assumed that the wall elements within the surface area of each wall are considered equal [1], and the rate of heat transfer through the wall can be summarized as:

$$q_{rad,g}'' + q_{conv,g}'' = q_{rad,c}'' + q_{conv,a}'' = q_{cond,w}'' \quad (2.15)$$

where the subscripts *g*, *c*, *a* and *w* represent the efflux gas, casing, annulus air, and liner wall. In most combustion chambers, the majority of the heat transferred from the hot efflux gases contained by the liner to the liner itself is through internal radiation.

Figure 2.11: Basic Heat Transfer Process (adapted from [1])

By knowing the size and shape of the gas volume and the overall pressure, temperature and chemical composition of the fuel, the internal radiation can be estimated. The external radiation or radiation from the liner wall to the outer casing is usually quite small when compared with the external convective heat transfer and it can often be neglected [1]. Internal convection is the most difficult heat transfer process to estimate accurately due to the rapid physical and chemical changes of the gas in the primary zone of the combustor. There are numerous uncertainties in this region including airflow pattern, the state of boundary-layer development, and the gas temperature. These uncertainties make it extremely difficult to derive an accurate model to determine the heat transfer in this region. According to Lefebvre [1], the classical heat-transfer relation for a straight pipe can be used by making three assumptions for calculations in this region,

- (1) The primary zone contains a reversal of flow and only the area directly next to the wall has flow similar to the straight pipe such that mass flow rate of the gas across the whole section can be found by summing the upstream and downstream flow components.
- (2) The local gas velocity at the wall will be greater than the downstream component.
- (3) The primary zone radial gas temperature profile is specifically designed to give lower-than-average gas temperatures near the wall.

The final mode of heat transfer is the conductive heat transfer through the liner wall due to a temperature gradient within the wall.

The liner wall temperature that could be expected in a combustion chamber with no film cooling, is fundamental to the geometric design and airflow distribution of the combustion chamber. Current liner materials are limited by their maximum allowable temperature (approximately 1123 K for gas turbine combustion liners [56]) and cooling is required to permit higher operating temperatures and therefore, more efficient combustion.

Film cooling is the most traditional method of removing heat from the liner. It involves passing air along the inner or hot surface of the liner wall, which provides a protective film of cooling air between the wall and the hot combustion gases. The air is usually passed through small slots in the liner that are evenly spaced to provide uniform cooling. The most widely used film-cooling devices are wiggletrips, stacked rings, splash-cooling rings and machined rings. These devices typically use the total pressure drop across the liner as the driving force for the injection of the film-cooling air [1]. The basic configuration of these devices can be seen in Figure 2.12 and due to their simplistic design and ease of manufacturing, they are widely used in gas turbine combustors.

Figure 2.12: Film-cooling Devices (a) Wiggletrip, (b) Stacked Ring, and (c) Splash-cooling Ring (adapted from [1])

Impingement cooling is a method of augmenting the convention film-cooling process by using a casing with small holes which permits cool air to enter the casing and impinging jets of cool air onto the outside of the liner wall. This method has the ability to reduce the

liner wall temperature by approximately 85 K without increasing the amount of cooling air [13].

Another and very effective cooling method is *transpiration cooling*. In this method, the liner wall is constructed from a porous material that provides a large internal area for heat transfer to the air passing through it. Since the pores are uniformly dispersed, the air jets from each pore form a protective layer of cool air over the entire inner surface of the liner and the convective heat transfer from the hot gas to the liner walls can be greatly reduced. One disadvantage of this method is that the pores can easily be clogged by particles or oxides. A variation of this cooling method is known as *effusion cooling*, where instead of using a porous liner, the liner wall is perforated by a large number of small holes (typically with a diameter large enough to prevent clogging) thereby producing a relatively uniform film of cooling air [1].

A unique method of increasing convective heat transfer on the cool side of the liner is through the use of fins, pedestals, and ribs on the backside of the liner to increase the effective area for heat exchange.

Most recently, developments in ceramics have led to the use of *thermal barrier coatings* (TBCs) consisting of refractory material on the liner walls. TBCs with low emissivity and low thermal conductivity reduce the wall temperatures by reflecting a large part of the gas radiation and by providing a layer of thermal insulation between the hot gas and the liner so that metal wall temperature is reduced. The resulting reduction in temperature is typically in the order of 150 K [1], which not only allows the operating temperatures of the combustor to increase, but it can greatly increase the life of the combustion chamber.

All of these methods are effective in their own way at reducing the liner wall temperature and numerous studies have been carried out to determine which method is most effective. The T56-A-15 Series III combustion liner currently uses film cooling and has four wiggle strip cooling corrugations, which can be seen in Figure 2.10. Rizk *et al.* [5] conducted a series of tests on the effects of burning different types of fuels on the liner wall temperatures of a T56-A-15 Series III combustor liner. The first testing involved using two types of thermal paint to determine the overall liner temperature distribution at a takeoff condition. This gave them an overall representation of the wall

temperatures in each zone of the combustor and then placed 30 C/A type thermocouples distributed on the dome, liner, and fishmouth or transition section. They determined that the transition section had the highest liner wall temperatures due to limited cooling in this area. The combustor durability is directly dependant on the liner wall temperatures that it is continually exposed to during day-to-day operations. Any hot streaks resulting from reduced air flow or nozzle blockages can cause excess temperature gradients, damaging the liner and reducing its overall life. Therefore, understanding the regions where the majority of the heat is transferred to the liner walls during combustion is important when designing the liner cooling methods and selecting the liner materials.

2.5 Combustion Chamber Exit Temperature Distribution

The mixing within a combustion chamber is an extremely important process that greatly affects the temperature distribution of the efflux gases discharging into the turbine. Not only does this delivery temperature affect the life and durability of the hot section components, but it also influences the maximum average turbine inlet temperature, which is directly linked to the overall engine output power and engine thermal efficiency. The delivery of the air in the dilution zone is designed to provide a gas temperature profile that is tailored to increase radially from the inner radius of the combustion chamber exit plane to a peak in the middle and then decrease again at the outer radius. This provides the turbine blade with a cooler temperature at the blade root where the blade physical stresses are greatest, and tip, which is thinner and more difficult to cool, thus prolonging blade life and allowing for a higher overall maximum average turbine inlet temperature. It is very important to try to avoid hot streaks and uneven temperature distributions for the turbine blades, but non-uniformities in the fuel spray characteristics often occur. Therefore, the turbine inlet guide vanes, which are fixed relative to the combustor and are required to operate in the hottest part of the flow, are designed to handle these non-uniformities and much higher mean temperatures. Studies have been conducted by Taylor [57] and Povey *et al.* [58], which attempted to understand the effects of hot streaks on downstream components and a means of reducing them, but met with minimal success. In these studies, measuring non-uniformities or hot streaks in the temperature

distribution of the efflux gases is a difficult task. Therefore, temperature profiles are often measured in terms of a uniform distribution of high and low temperature regions.

The index of the uniformity of the overall temperature distribution at the exit of a gas turbine combustor is defined as the *pattern factor* (PF) [1]. It involves a comparison between the differences of the maximum and mean exit temperature, and the mean and compressor exit temperature, and is normally given as:

$$PF = \frac{T_{max} - T_4}{T_4 - T_3} \quad (2.16)$$

The combustor exit temperature, T_4 , is normally calculated using the overall mass-flow-weighted or area-weighted mean temperature of all the recorded exit temperatures. Equation 2.16 is designed to recognize deviations in the temperature distribution that exceeds the mean value (*i.e.* peaks in the distribution) and will be closer to zero for instances where there is a perfectly uniform exit-temperature distribution [1] and fewer dips in the temperature distribution. Typical values of a pattern factor for a gas turbine combustors range from 0.17 to 0.4 [5, 21, 59] and most nozzle guide vanes are designed to withstand the maximum temperature found in any distribution [1].

Studies by Bicen *et al.* [59] and Friedman and Carlson [60] used mass-flow-weighted mean temperatures to determine the effects of altering various parameters like air-fuel ratio, inlet temperature, and pressure on the pattern factor. Using this method involved measurements of both velocity and temperature, thus creating a velocity contour map to evaluate the efflux gases mass-flow-weighted mean temperature. This method increases the accuracy of the measurements, but also increases the complexity of experimentation requiring exhaust gas sampling, and it is therefore not commonly used. A much more simplistic technique that is more commonly employed for gas turbine combustor studies involves taking temperature measurements at equal area increments to obtain area-weighted mean temperatures.

The pattern factor is the non-dimensional parameter that is most significant to the design of the nozzle guide vanes. As for the turbine blades, since they are not fixed relative to the combustor, the circumferentially averaged radial temperature profile is of more interest. The parameter used to describe the radial temperature distribution factor is

known as the *profile factor*. The maximum circumferential mean temperature, T_{mr} , is used instead of the maximum exit temperature that is used for the determining the pattern factor. The profile factor is given as:

$$\text{Profile Factor} = \frac{T_{mr} - T_4}{T_4 - T_3} \quad (2.17)$$

A parameter that is also a function of radius and is used to compare the design temperature profile to the actual temperature profile is known as the *turbine profile factor* and is defined as:

$$\text{Turbine Profile Factor} = \frac{(T_{4,r} - T_{4,des})_{max}}{T_4 - T_3} \quad (2.18)$$

This parameter is used to show the maximum temperature difference between the design temperature at any given radius around the circumference and the average measured temperature at that same radial location [1]. Figure 2.13 provides an example of these differences.

Figure 2.13: Example of Exit Temperature Profile Parameters [1]

According to Lefebvre [1], the turbine profile factor is most commonly used when examining air cooling of the nozzle guide vanes and turbine blades. Experimentation is often limited when using the turbine profile factor due to the availability of the design temperature profiles from manufacturers. Therefore, most researchers use the profile factor and pattern factor to examine the combustor exit temperature distribution. After completing an extensive survey of both the past and current research being conducted on gas turbine combustors, it is evident that the pattern factor is the most commonly used parameter to describe the temperature distribution seen by the hot section components [5, 21, 29, 57-60].

There are many different factors that determine the temperature profile of the efflux gases leaving a combustion chamber. Barringer *et al.* [61] suggest that the most significant components of the combustor that contribute to the non-uniformities in the temperature distributions are the dome swirlers, film cooling corrugations, and the size, shape, and discharge coefficients of the liner holes. All of these components influence the distribution of the air throughout the liner, and thus the mixing of the products of combustion with the cooling air. The products of combustion and their associated temperature distribution, which is eventually altered by the dilution air and seen by the turbine, are strongly affected by fuel spray characteristics like droplet size and spatial distribution of droplets. There is limited research into the effects of each of these parameters on the temperature distribution of the efflux gases. Kotzer [62] conducted a study on variations in the physical geometry of a group of T56-A-15 Series III combustion liners, and concluded that even small deviations from nominal dimensions in the dilution zone correlated to an increase in pattern factor. Sharma and Som [63] investigated the influence of fuel volatility and spray characteristics on the combustor exit temperature distribution, combustion efficiency, and emissions. They noted that for highly volatile fuels, the pattern factor was reduced with a decrease in droplet size and an increase in spray cone angle. The effect of altering the size and location of the primary zone holes was examined by Bicen *et al.* [59], and the dilution holes by Cox [64] at various operating conditions. Both studies involved the use of an annular combustor and examined the effects of changing hole geometry and distribution of cooling and dilution air. Bicen *et al.* [59] concluded that moving the holes by only $\frac{1}{2}$ pitch in the primary

zone resulted in poorer mixing and an increase in pattern factor. Cox's [64] analysis suggested these results in that he concluded that the pattern factor was greatly affected by the mixing process in the primary zone and the resulting temperature profile in that region.

Lefebvre [1, 65] concluded that there must be sufficient penetration of the dilution air jets to form adequate mixing regions and ensure a satisfactory temperature profile. The degree of mixing and the level of penetration of the dilution jets are directly related to the pressure drop across the liner. He also noted that there must be enough time and distance for adequate mixing of the combustion products and dilution jets, therefore the combustor length is also a parameter that strongly influences the combustion chamber exit temperature profile. Equation 2.19 provides a generalized relationship between the pressure drop across the liner, the liner length, and the pattern factor.

$$PF = f\left(\frac{L_L}{D_L}\right)\left(\frac{\Delta P_L}{q_{ref}}\right) \quad (2.19)$$

where the pattern factor is a function the liner length, L_L , liner diameter, D_L , and the liner pressure loss factor, $\Delta P_L/q_{ref}$ and q_{ref} is a function of the mean velocity across the plane of maximum cross-sectional area of the liner casing. Lefebvre [65] provides refined correlations for tubular and tubo-annular combustion chambers, as seen in Equations 2.20 and 2.21, respectively:

$$PF = 1 - \exp\left[-0.070\left(\frac{L_L}{D_L}\right)\left(\frac{\Delta P_L}{q_{ref}}\right)\right]^{-1} \quad (2.20)$$

$$PF = 1 - \exp\left[-0.070\left(\frac{L_L - L_e}{D_L}\right)\left(\frac{\Delta P_L}{q_{ref}}\right)\right]^{-1} \quad (2.21)$$

where

$$L_e = \left(\frac{0.33 \cdot 10^6 \dot{m}_a}{\rho_f A_L}\right)\left(\frac{d_o^2}{\lambda}\right) \quad (2.22)$$

where L_e is the liner length employed in fuel evaporation, A_L is the liner area, d_o is the orifice diameter, $\dot{m}_a$ is the air mass flow rate, ρ_f is the fuel density and λ is a measure of the fuel volatility. Equation 2.20 represents the case for high combustion pressures (1 MPa [65]) and assumes that the entire liner length is available for mixing. This assumption is based on the premise that at high combustion pressures, the evaporations rates are so fast that the time and space required for the fuel to evaporate are negligible [1]. For the low combustion pressure (100 kPa [65]) correlation represented in Equation 2.21, there is an associated liner length that is required for evaporation, as given in Equation 2.22, which is dependent on the fuel properties, and cannot be ignored. Utilizing the above correlations, the pattern factor over a range of operating conditions can be predicted.

2.6 Summary of Literary Review

Combustion is a complex process that is influenced not only by the chemical interactions occurring, but also by the physical processes like thermodynamics, heat transfer, mass transfer, and fluid dynamics. The combustion process within a gas turbine combustor can be dominated by the chemical kinetics, evaporation rates, or the degree of mixing, which are all dependant on the operating condition and surrounding environment. The performance of most combustors relies heavily on adequate atomization and mixing, rather than the rate of chemical reactions. The degree of mixing is greatly influenced by the droplet size and spatial distribution produced by the fuel nozzle and the location and size of the air admission holes in the combustor liner. Numerous studies involving different analytical methods have been undertaken to more thoroughly understand the influence of inadequate atomization and mixing on combustion efficiency. The primary means of quantifying the performance of the combustor is the examination of the uniformity of the efflux gas temperature profile, which is a critical design requirement for gas turbine engines to ensure minimal damage and durability of the hot section components. The temperature distribution is often described by what is known as the pattern factor. Many correlations for the pattern factor based on the temperature profile produced in the primary zone and the mixing in the dilution zone have been developed and used extensively in gas turbine combustion experimentation. There are still many

unknowns when trying to quantify the performance of a combustor and experimentation is still required to gain a better understanding of all the processes and their subsequent effects on the exit temperature distribution of combustion chambers.

Chapter 3

Experimental Setup

3.1 Experimental Focal Point

The Canadian Air Force currently operates Hercules CC-130E and CC-130H transport aircraft and the Aurora CP-140 maritime surveillance aircraft, which are each powered by four Allison/Rolls Royce T56-A-15 turboprop engines. This engine consists of a power section, which contains the compressor, turbine and combustion assemblies, the torquemeter and anti-icing shroud assembly, and the reduction gear assembly. In the power section, air is delivered to a combustion section containing six turbo-annular combustion chambers through a 14-stage axial flow compressor. After combustion occurs, the liners direct the hot efflux gases into a four-stage axial flow turbine, which drives the compressor and reduction gear box through a single shaft. Figure 3.1 provides a breakdown of the assemblies found in the T56-A-15 engine.

To determine the influence of nozzle condition on the pattern factor, the combustion section of the Allison/Rolls Royce T56-A-15 engine has been selected as the focal point of this research. The combustion chambers are positioned radially around an inner pressure casing, which houses the primary drive shaft, and enclosed by an outer pressure casing that guides the airflow. A dual-orifice pressure swirl fuel nozzle is centred within the inlet flared fitting of each combustion chamber and the combustor exhaust transition slots into the turbine inlet guide vane assemblies, thus setting each liner's radial position. The axial positioning of the liners is accomplished by placing igniter plugs in two liners, and dummy igniter plugs in the four remaining combustion chambers [5]. The liners are interconnected with cross-over tubes to allow for flame transfer during start-up. The efflux gas temperatures are monitored by three air-cooled thermocouples located in the aft of each combustion chamber.

The Canadian Air Force has been experiencing problems with premature damage to the hot section components of the T56-A-15, more specifically the turbine inlet guide

vaness and first stage turbine blades. The potential cause of this damage can be related to the condition of the fuel nozzle and the spray pattern being produced. The fuel flow rate is controlled using measurements taken from single point thermocouple probes at exhaust of each combustion chamber. Any probe could be subjected to a hot or cold streak due to a non-uniform spray from a faulty or blocked nozzle. This streak or non-representative temperature value could in turn trigger the delivery of more or less fuel to achieve the desired turbine inlet temperature distribution and ultimately result in a further degradation of the turbine components, leading to reduced component lives and increased maintenance costs.

<u>Number</u>	<u>Engine Section</u>
1	Power Section Unit Assembly
2	Compressor Assembly
3	Turbine Section
4	Fuel System
5	Electrical System
6	Control System
7	Power Transmission Unit Assembly
8	Torquemeter, Anti-icing Shroud and Safety Coupling
9	Reduction Gear Assembly
10	Housing Assembly

Figure 3.1: T56-A-15 Turboprop Assemblies [66]

In addition to these hot streaks, faulty fuel nozzles can often cause the formation of hard carbon within the combustion chamber, which can break off and cause further damage to the downstream components.

Studies have been conducted by the National Research Council of Canada in conjunction with *Standard Aero Limited* (SAL) [35] to examine known faulty nozzles to determine their effect on downstream components. To expand upon this study, this researcher has selected the T56-A-15 engine for the experimentation of a variety of nozzles taken from operational CC-130 aircraft over a vast range of operating hours. The study focuses on determining the nozzle condition and the effect of its spray characteristics on the combustion chamber exit temperature distribution. This study will also be used to evaluate the effectiveness of the current T56-A-15 nozzle condition testing and cleaning being conducted at ATESS. Several different test apparatus were used during this experimentation including an optical patternator, a Phase Doppler Particle Analyzer, and a Combustion Chamber Sector Rig (CCSR). These experimental installations were inherited from previous researchers and were modified to meet the requirements of this work.

3.2 Optical Patternation Setup

3.2.1 Overview

Over the past few years, the Royal Military College of Canada in conjunction with the Department of National Defence and *Nexum* Research Corporation, have been conducting tests of an optical spray pattern analyzer (OSPA) to develop a more efficient and detailed means of analyzing fuel nozzle spray images. The OSPA method is based upon the principles of Mie scattering and provides a non-intrusive means of examining spray characteristics. The T56-A-15 nozzle spray test setup consists of fuel delivery system, collection tank, fuel nozzle test stand, and a *Nexum* Instrumentation OSPA 200 optical spray pattern analyzer, which can be seen in Figure 3.2. MIL-C-7024D Type III calibration test fluid [67] was selected to perform the fuel nozzle spray tests. This fluid has similar properties to JP-8 (Jet-A1) [68], which are summarized in Table 3.1.

Table 3.1: Chemical Properties Comparison

Property	MIL-C-7024D Type III	Jet-A1
Flash Point, °C	40	42
Density at 15°C, kg/m ³	780	804
Viscosity at -20°C, mm ² /s	2.6	3.5

Figure 3.2: T56-A-15 Fuel Nozzle Optical Patternation Test Setup

Since this is a comparative study between the spray characteristics of the fuel nozzles, any differences between the two fluids are considered to have negligible impact on the results. The test fluid was transferred from a 50-US gallon drum and pressurized using nitrogen. A solenoid on/off switch was used to control the flow and the mass flow rate was controlled using a pressure regulator valve. Once the valve was opened and the spray reached the desired pressure, the OSPA system was used to capture the image of the spray. Images for each nozzle were captured at fuel delivery pressures of 345 kPa (50

psi), which was prior to the secondary port opening, 517 kPa (75 psi), which was just after the secondary port opens, and 620 kPa (90 psi).

The OSPA consisted of a solid state *Lasiris*TM SNF laser, which was operated at its maximum power of 100 mW and with an 690 nm output. This laser produced visible light, which provided the advantages of easier alignment and calibration, but required increased safety precautions to avoid injury. The laser would normally produce a straight beam, so it was fitted with a cylindrical lens to transform the beam into a fan shaped light sheet. The light sheet was uniform when it was properly aligned with the lens and the light intensity varied smoothly across the sheet.

The light sheet fanned outward with an angular spread of 30 degrees. To compensate for any variations in light intensity, a series of lenses were placed opposite the laser that focused the light onto a set of photodiodes. The variation in the light sheet intensities could then be monitored and measured over one degree increments. The monitored intensities served two purposes. They provided an indication if the light sheet was stable and within normal operating parameters; and they could be used to correct the image for any inhomogeneities in the light sheet [43]. The light sheet remained evenly distributed throughout the testing and did not require re-calibration or adjustments. As the spray passed through the light sheet, the light is dispersed by the droplets through Mie scattering. A CCD camera was used to collect the scattered light, which had been placed so that the horizontal centre of the image coincided with the location where the light sheet intersected the axis of the spray. The signal from the camera was then fed back to the computer where the image analysis software was used to average the image over a number of frames. In this research, 30 frames were averaged, which corresponded to approximately one second of spraying.

It was known that with additional airflow and no combustion of the fuel and air mixture, the spray would tend to collapse into a thin plume as observed by Reddy and Mishra [69], which would make it difficult to see clear streaks or non-uniformities in the flow. Therefore, the patterning was conducted using open air tests, where the spray patterns were observed under ambient temperature and pressure with no additional airflow.

3.2.2 Corrections and Confirmations

The distance between the fuel nozzle discharge orifice and the light sheet required some corrections to obtain an accurate representation of the spray at the three test pressures. Most researchers have noted an increase in droplet size as the axial distance between the discharge orifice and patternator increased. Dodge and Bigalow [22] noted that the D_{32} (droplet Sauter mean diameter) decreased along the centreline of the spray as the axial distance increased, whereas Rizk and Lefebvre [16] found that the D_{32} initially decreased with axial distance and then continually increased. Reddy and Mishra [69] observed that D_{32} increases with radial distance from the spray centre, which indicated the presence of smaller droplets at the centre of the spray, with larger ones around the peripheral. For optical patterning, researchers agree with the results achieved by mechanical patterning [16, 22, 70] in that as the spray moves away from the nozzle, it spreads out, with larger more dispersed droplets at the outer edges. Underhill *et al.* [43] suggest that the nozzle should be placed as close to the light sheet as possible in the region where the spray has the best spatial definition. Sample tests were conducted at 13, 19, 25.4, and 38 mm from the nozzle and a typical test is depicted in Figure 3.3.

Figure 3.3: Intersection of Spray and Light Sheet

It was found that the clearest images were obtained at 25.4 mm from the nozzle and images at greater nozzle distances lost most of their definition. This coincided with the experimental setup and results found in the research conducted by Underhill *et al.* [36, 43, 70].

The quality of the images was examined over a range of fuel delivery pressures. It was found that as the pressure was increased above 517 kPa, some of the spray recirculated back into the light sheet, causing the light to scatter outside of the pattern and a halo was produced that was captured by the camera and distorted the image [43]. Additionally, small droplets were sucked back into the hollow area inside the cone of the spray resulting in images with filled-in interiors. To remedy this, a fine-mesh honeycomb material was placed below the spray and out of the line of sight of the camera, which helped reduce the amount of spray recirculation. An air extraction system was also added to the collection bucket, which served to suck the spray down and prevented any small droplets from bouncing back into the field of view.

A final and important correction that was made in all optical patterning tests was the positioning of the camera to reduce the geometric distortion of the image. Figure 3.4 shows the geometry of a typical OSPA system. In this figure the camera is positioned to collect forward scattering light, whereas in the actual experimentation setup, the camera was positioned to collect backward scattering light.

Figure 3.4: OSPA geometry [43]

As the angle of the camera increased, the scattering efficiency dropped off exponentially, and as the angle was reduced, the perspective distortion effects increased, as first noted by Underhill *et al.* [43]. Extensive testing has been conducted by *Nexum*, determining that an angle of 24.5 degrees above the plane of the light sheet was the best compromise between the distortion due to the camera angle and the decrease in light intensity that occurs when the angle increases.

The images collected by the camera must be corrected for both the scattering efficiency and the viewing angle distortion. This process was thoroughly described by Underhill *et al.* [36, 43, 70]. The geometric or perspective corrections are accounted for with a simplistic geometric transformation. A square calibration grid is placed in the laser sheet plane and the image of this grid can be transformed into a uniform square grid of the image, which serves as a verification of the accuracy of the geometric transformation. The image analysis software that was used to obtain the images found in the Results Section of this report was developed by *Nexum* Research Corporation [71] and incorporated these corrections.

3.3 Phase Doppler Particle Analyzer Setup

To accurately measure the droplet sizes being produced by the T56-A-15 fuel nozzles at the scaled operating conditions, an *Artium*TM PDPA was used. The PDPA method is based upon principles of light scattering interferometry. The T56-A-15 nozzle spray test setup consists of the same fuel delivery system, collection tank, and fuel nozzle test stand used in the optical patternation spray analysis setup, except it uses an *Artium*TM PDPA instead of a *Nexum* OSPA. In the PDPA, two lasers intersect 25.4 mm below the nozzle discharge orifice to form a non-intrusive probe or measuring volume. Droplets within the measuring volume cause both beams to scatter, which are both Doppler shifted and then collected by a collecting lens and superimposed on the multi-detector receiving probes. These receiving probes were strategically located at an off-axis collection angle. The Doppler frequency difference between the two recovered beams from the different detectors was determined, which was proportional to the size of the droplets. The number of droplets or particles crossing the measuring volume at any given time was known as the *burst density*. The resulting Doppler signal was comprised of the superposition of

many bursts and was continuous over the whole observation period. This type of signal was used to calculate the average Doppler frequency shift and therefore, the average droplet diameter within the measuring volume.

Figure 3.5: *Artium*TM PDPA Experimental Setup

Before arriving at the signal processor, the Doppler signal leaving the multi-detector probe was band-passed filtered to remove any high-frequency noise, allowing for easier determination of the Doppler frequency. Figure 3.5 is a photo of the PDPA system used during testing. The laser beam intersection is clearly visible in the T56-A-15 nozzle spray setup. Measurements of the distribution of droplet sizes in terms of D_{32} were taken radially at 3 mm intervals from the spray centre out to a radius of 38 mm, at an axial distance of 25.4 mm from the nozzle discharge orifice.

During the course of this experimentation, it was found that using this particle test apparatus posed several health risks to the operator. The test fluid used was extremely toxic and caused severe illness when inhaled. Therefore, it is recommended that future tests be conducted using a closed test chamber with adequate fume extraction.

3.4 Combustor Chamber Sector Rig

3.4.1 Overview

From 2005 to 2007, the Royal Military College of Canada in conjunction with *Standard Aero Limited*, designed and built a *combustion chamber sector rig* (CCSR), which will be referred to as the test rig, for research studies on Allison/Rolls Royce T56-A-15 combustion components. The test rig operates under ambient pressure and temperature inlet air conditions and was designed as closely as possible to simulate the airflow that would be experienced by a T56-A-15 engine under normal operating conditions. The test rig was constructed using a combination of manufactured and actual engine components for the air, fuel, and exhaust systems, and T56-A-15 engine components from the diffuser to the turbine inlet. It was designed to investigate the efflux gas temperature distribution from a single combustion chamber. It was concluded that due to the symmetrical design of the T56-A-15 turbo-annular combustion section and assuming the fuel spray characteristics and airflow conditions are constant, the results from one combustion chamber would be analogous to any of the remaining five combustion chambers.

The test rig was divided into several different rooms for safety considerations. The fuel system, air system, and main electrical panel were all located separately from the test rig and its electrical control panel. The main electrical panel provides power to the air system, fuel system, and the test rig electrical control panel and also houses the main fuses, disconnects, transformers, relays, and electrical control strips. A diagram depicting the location of all the major components of the test rig can be seen in Figure 3.6, detailed electrical schematics and a components list are included in Appendix B.

The test cell itself, contains a diffuser cone, diffuser, annular casing that houses one combustion chamber test specimen and two dummy chambers, driveshaft casing and turbine inlet. Figure 3.7 provides a schematic of the combustion test section showing all of the internal components. The diffuser cone served to force the air into a 120° sector, which was centred on the test specimen. To maximize the airflow in to the test section, a 240° blanking plate was welded in the diffuser entrance, as seen in Figure 3.8.

Figure 3.6: Test Rig Schematic

The two dummy combustion chambers were placed on either side of the test specimen to model the side flow around the test chamber. The air admission holes and cooling corrugations of the outside halves of the dummy cans were blocked using blanking plates and fins were welded axially on the top and bottom of the chambers; they extended axially to the pressure casing and driveshaft casing respectively. These fins served to maintain the axial flow in the test cell. Fuel nozzles were installed in each dummy combustion chamber along with igniters to support alignment, but no fuel was supplied to these nozzles.

Figure 3.7: Combustion Test Cell Cross-section [62]

Figure 3.8: Diffuser Front View [62]

Adjustable dampers installed in the exhaust exit of the dummy chambers were used to balance the airflow between the test and dummy combustion chambers. The dampers were attached to one main input rod; therefore all adjustments would be symmetrical. A thermocouple rake for measuring the exhaust gas temperatures was attached to a sweeper arm on the rear turbine casing and was connected to a screw jack assembly. The movement of the screw jack was controlled by a stepper motor and moved the sweeper arm along the circumference of the 120° sector. Figure 3.9 provides a rear view of these components and the combustion test section looking upstream.

Figure 3.9: Combustion Test Section Looking Upstream

3.4.2 Fuel System

The fuel system of the test rig consisted of a 210 L fuel drum, gear pump, and an explosion-proof motor. The gear pump was powered by the 746 W (1 hp) motor and

extracted fuel from the storage drum, which was fitted with a direct-acting pressure relief valve to prevent over-pressurization. The pump was capable of providing the test rig with a fuel flow rate of 9.5 L/min at a pressure of 1.38 MPa (200 psi). Figure 3.10 shows the motor, pump, and fuel drum on its solid stand within the fuel system room. A 3 μm filter was fitted within the fuel line to ensure no particulate from the drum or pump could pass into the test rig.

Figure 3.10: Test Rig Fuel System

The test rig fuel delivery system was fitted with back pressure and throttling valves to control the fuel flow rate and maintain a steady delivery pressure. Fuel was supplied only to the centre test combustion chamber through a T56-A-15 dual-orifice pressure swirl nozzle. Excess fuel was returned to the fuel drum through a heat exchanger. Several detectors were mounted inside the test cell to monitor for excess fumes and the presence

of fuel. If a fuel leak was detected, power from the main electrical panel was designed to be immediately disconnected.

3.4.3 Air System

The test rig air flow was supplied by a two-stage centrifugal blower. Power was supplied to the blower by a 40 hp electric motor and the motor speed was controlled by a variable-frequency drive. Figure 3.11 shows the blower, electric motor and variable speed drive, along with the main electrical panel for the test rig.

Figure 3.11: Test Rig Air System and Main Electrical Panel

The air was fed to the test rig through galvanized ducting, which also housed a Type-K thermocouple, and several pressure sensors. The temperature and pressure readings were used to determine the air mass flow rate and pressure loss across the combustor, which will be discussed in subsequent sections.

The exhaust system immediately downstream of the test rig was comprised of stainless steel ducting with a water jacket placed after the test section. The water jacket acted as a thermal break from the high conductive heat transfer that was occurring in this portion of the ducting. Stainless steel ducting was selected due to the high exit temperatures of the combustion rig and the corrosiveness of the efflux gases.

3.4.4 Modifications

The experimental setup described in Sections 3.4.1 to 3.4.3 was commissioned and used in research prior to this study. Based on the findings and recommendations of the study conducted by Kotzer [62] on this test vehicle, several modifications were carried out. Kotzer [62] conducted his experimentation by modeling an engine operating condition of 463 km/hr (250 knots) at 7620 m (25 000 ft), which equated to a test rig air mass flow rate of 0.93 kg/s and a fuel mass flow rate of 8.3 g/s, giving an equivalence ratio of 0.26. This fuel mass flow rate occurred just after the opening of the secondary port, meaning that the delivery pressure to the secondary port would be low and could possibly result in large droplets being produced. To remedy this situation, Kotzer [62] blocked the secondary port of the test rig fuel nozzle. With all the fuel being forced through the primary port, there was excellent atomization with very small droplets being formed. However, the resulting fuel spray characteristics in this arrangement are not representative of what is actually being produced by a fully functional nozzle. Since the current experimentation examines the effects of the fuel nozzle condition on the exit temperature profiles, it was not appropriate to block the secondary port of the fuel nozzle. To achieve a more representative fuel flow, a higher fuel flow rate was required to ensure the secondary port was fully open. The experimentation operating conditions needed to be matched as closely as possible with an engine operating condition; therefore, the fuel and air mass flow rate were increased. A typical military cruise operating condition of 463 km/hr (250 knots) at 7650 m (25 000 ft) was selected for this research, which

equated to a test rig air mass flow rate of 0.98 kg/s and a fuel mass flow rate of 9.5 g/s, giving an equivalence ratio of 0.28. It was recognized that by increasing the fuel flow rate, the flame temperature would also increase. In past experimentation conducted using the test rig, the Type-K thermocouples were operating at their upper limit and Kotzer [62] recommended they be replaced with Type-R thermocouples, which have a higher maximum operating temperature. As a result of the increase in fuel mass flow rate, several modifications were made to the test rig.

3.4.4.1 Foam Insert

The first modification was to redirect and force the airflow into the 120° test section by reducing the pressure loss in the inlet duct, thus increasing the maximum possible experimentation air mass flow rate. In addition to the wall-boundary-layer losses, the inlet duct takes a sharp 90° bend immediately before the diffuser cone, which causes a further pressure drop from the exhaust of the blower to the inlet of test section. To achieve a smoother transition of airflow from the inlet duct to the test section, a foam insert was manufactured that reduced the area of the inlet duct by 60%. To reproduce the interior shape of the inlet duct and diffuser cone duct, *EvercoatTM* two-part expanding foam, which expands to 20 times its volume, was poured into each duct and allowed to expand and harden. The two pieces of foam were then removed and shaped to form a tongue-shaped insert. Since the experimentation inlet air temperature can be in excess of 40 K, the inserts were coated with fibreglass using *WEST SYSTEMTM* epoxy resin and hardener and a fine mesh E-glass cloth to prevent damage and potential breakaway into the test section. The two inserts were adhered in place in the ducts using standard silicone caulking. Any gaps were filled in with *BondoTM* body filler. Figure 3.12 shows the foam inserts in both the 90° duct and the diffuser cone duct before they were combined. CAD models of the combined insert outside of the rig and then in situ are shown in Figure 3.13.

Extensive testing was conducted to determine the pressure drop between the inlet duct and the test section before and after the installation of the foam insert. Tests were conducted by connecting a vertical water manometer to static pressure ports located in the inlet duct and in the test combustion chamber. Manometer readings were recorded

for blower speeds ranging from 400 to 2700 rpm. Three tests were conducted without the foam insert in place with only a 0.5% variation between the tests. Another three tests were conducted with the foam insert in the inlet duct and there was no deviation between the results of these tests.

Figure 3.12: Inlet Duct Foam Inserts

Figure 3.13: CAD Models of Foam Insert

Air mass flow rates were calculated at each blower speed using the inlet air temperature and pressure, the dynamic static pressure obtained, and the dimensions of the inlet duct. Figure 3.14 provides a comparison of the results obtained for the two scenarios and

shows that an 11% increase in the static pressure drop with the foam insert resulted in a 9% increase in the air mass flow rate available for experimentation. Sample calculations for the air mass flow rate can be found in Appendix C.

To ensure that the foam insert had minimal impact on the airflow distribution to the test section, several tests were conducted using the same blocked nozzle as Kotzer [62] and a new combustion chamber at similar test conditions. Figure 3.15 provides a comparison between the temperature distributions obtained by Kotzer [62] and by this researcher with the foam insert in the inlet ducting.

Figure 3.14: Air Mass Flow Rate at Inlet to Test Section at Varying Blower Speeds

After analysing the data, the resulting temperature distribution was very similar to that obtained during his research with characteristic reductions in temperature found at the 12°, 30° and 48° locations due to the presence of the engine thermocouples and a hot zone at the 10° position. Kotzer [62] achieved pattern factors ranging from 0.4 to 0.6 for

a wide range of serviceable combustion chambers. Tests conducted by this researcher resulted in pattern factors ranging from 0.42 to 0.56, thus in close agreement with his research. From this comparison, it can be concluded that the foam insert had no apparent impact on the airflow distribution within the test section.

Figure 3.15: Non-Dimensional Exit Temperature Distributions (a) Kotzer, and (b) With Foam

3.4.4.2 Stepper Motor and Process Automation

During the experiments conducted by Kotzer [62] and Pucher *et al.* [72], the thermocouple rake was traversed through the test section by the operator manually

turning a hand dial that was connected to the screw jack assembly until the desired positioning was achieved. This was a cumbersome and time consuming process that required the researcher to gain proficiency in manoeuvring the thermocouple rake in very small increments as described by Kotzer [62]. This also led to problems with repeatability in the measurement position. To reduce the experimentation time and amount of error associated with the positioning of the thermocouple rake, an *Applied Motion Products* HT23-401 stepper motor, ST5-Si driver, and *Si Programmer*TM software were installed. A high-torque stepper motor, with a maximum torque of 1.4 N·m, was selected due to the amount of torque required to turn the screw jack and the holding torque required to maintain the thermocouple rake's position in the combustion chamber exhaust flow and to prevent backlash.

The stepper motor was hard-mounted to the back wall of the electrical control panel and a high torque 1.0 m steel cable was used to attach the motor to the screw jack assembly inside the test cell. Commands were sent to the motor by the ST5-Si driver, which delivered 24 VDC with an output current of 5.0 Amps to the motor. The driver was provided inputs by the *Hewlett Packard* data acquisition unit (model 3852A), which interfaced with the *Si Programmer*TM software installed on a standard desktop computer through a *general purpose interface board* (GPIB). A motion program was designed using the software, which controlled the location and movement speed of the thermocouple rake using *LabVIEW*TM software. It is known that as the speed increases, the torque available decreases, therefore the rotation speed was kept at the lower limit of the motor's capabilities, which was 1 rev/sec.

3.4.4.3 Thermocouple Rake

The thermocouple rake that was designed and used in the research by Kotzer [62] and Pucher *et al.* [72] was also used in this research. The rake was designed and manufactured based on the recommendations of Saravanamuttoo [4] for combustion study applications. It consisted of a thermocouple mount constructed from 304 stainless steel with open circuit internal water cooling, since it was exposed to temperatures close to the material's melting point, and four Type-K exposed-bead thermocouples held in place at the rear of the mount with set screws. The cooling water was cycled through the

mount during testing and sprayed into the exhaust gases downstream providing further cooling for the exhaust duct. The cooling water mass flow rate was approximately 0.1 kg/s at an average temperature of 285 K and experienced an average temperature rise of 30 K during experimentation. Kotzer [62] conducted a heat transfer analysis on the rake and determined that to maintain sufficient cooling within the mount, the water temperature needed to remain under 373 K. All the experiments conducted during this research remained well below this value.

In accordance with the recommendations provided by Saravanamuttoo [4], it was imperative that the thermocouple rake not disturb either the upstream or downstream flow patterns. Since the experimentation involved examining the temperature profile that the hot section components are exposed to in the engine, the flow around the thermocouples was not to be altered from the naturally occurring patterns. Any disturbances in the flow could affect the temperature measurements. It was for this reason that Kotzer [62] minimized the width of the mount and conducted a flow analysis to determine if there were any upstream disturbances introduced by the thermocouple mount. He concluded the flow reached 99% of the free stream velocity at 60 mm from the rake leading edge and there would be negligible disturbances generated by the rake.

The positioning of the thermocouple rake was also an important factor in accurately representing the temperature distribution experienced by the nozzle guide vanes and first stage turbine blades. Kotzer [62] designed and positioned the rake, with the thermocouples extending 17.78 mm from the edge of the mount, so that the rake swept circumferentially behind the struts and three engine thermocouples in a measurement plane that was at approximately 50% of the nozzle guide vane axial chord. He equally spaced the four 3.175 mm diameter thermocouples in the mount, which was centred in the turbine inlet, thus providing a radial measurement resolution of 6.82 mm per measurement. This configuration can be seen in Figure 3.16 and 3.17 and was maintained for the current research.

The four thermocouples used by Kotzer [62] were *Omega* Type-K exposed-bead thermocouples with *CHROMEALOMEA*® thermocouple wire protected by 3.175 mm diameter Super *OMEGACLAD*® XL sheathing filled MgO insulation and had a maximum operating temperature of 1645 K. During his experiments he noted that rig exhaust

temperatures were close to this value and witnessed accelerated oxidation of the exposed thermocouple beads, requiring replacement every five tests. The current experimentation involved operating the test rig at increased fuel mass flow rates and thus higher equivalence ratios, which was expected to result in elevated combustion chamber exit temperatures.

To broaden the testing range and capabilities of the test rig, the Type-K thermocouples needed to be replaced with thermocouples that had a higher maximum operating temperature and a greater resistance to physical degradation at high temperatures. Table 3.2 provides a summary of the operating ranges of the thermocouples that were commercially available at the time of this research.

Figure 3.16: Installed Thermocouple Rake Looking Downstream [62]

Figure 3.17: Thermocouple Rake with (a) Type-K [62] and (b) Type-R Thermocouples Installed

Table 3.2: Thermocouple Operating Ranges [73]

Thermocouple	Maximum Service Temperature
J – Iron/Constantan	482°C (900°F)
K – Nickel-Chromium/Nickel-Aluminum	982°C (1800°F)
T – Copper/Constantan	260°C (500°F)
E – Nickel-Chromium/Constantan	593°C (1100°F)
N – Nicrosel/Nisil	982°C (1800°F)
R,S – Platinum-Rhodium/Platinum	1480°C (2696°F)
B – Platinum-Rhodium/Platinum-Rhodium	1700°C (3092°F)

Preliminary tests conducted at the selected operating condition produced temperatures in excess of the maximum operating temperature of Type-K thermocouples and resulted in complete degradation of the exposed beads. Therefore, only the Type-R, -S and -B thermocouples would meet the current experimental requirements. Saravanamuttoo [4]

recommended the use of Type-R Platinum-Rhodium (Pt-13% Rh) thermocouples for any testing involving gas path temperature measurements in gas turbine engines and components. Based on this recommendation and the results of a cost comparison between the various types of thermocouples, four Type-R thermocouples were custom manufactured by *Omega*, and used a 24 AWG Pt-13% Rh thermocouple wire. The thermocouple wire was protected by 3.175 mm diameter Pt-13% Rh sheathing filled with MgO insulation. As shown in Table 3.2, the maximum operating temperature for the thermocouples was 1753 K. Figure 3.17 provides a comparison between the thermocouple rake with Type-K thermocouples installed and Type-R thermocouples installed. It is important to note that due to the high cost of the Type-R thermocouples, their length had to be limited to 200 mm where they were connected to an *Omega* high temperature ceramic connector, which was connected to *Omega* Type-R glass braided extension wire that fed into the HP3852A data acquisition unit. The conditioning required for the thermocouple signal was built into the data acquisition card. The Type-R extension wire was rated to a maximum temperature of 473 K [73]; therefore ambient air temperature tests were completed to determine the air temperature surrounding the thermocouple rake. This testing was conducted by placing Type-K thermocouple probes at distances of 100 mm and 150 mm below the bottom edge of the thermocouple mount to determine the gas temperature at these locations. It was found that the gas temperature was 523 K at the 100 mm location and 453 K at the 150 mm location. Therefore, a probe length of 200 mm was selected for this research to ensure the extension wire was within its operating limits.

Another consideration when replacing the thermocouples was the fact that the Type-R thermocouples were ungrounded from the sheathing, which meant that the response time would be much slower (thermocouples having a higher time constant) than the exposed junction thermocouples used by Kotzer [62] and Pucher *et al.* [72]. A *time constant* (τ) is defined as the amount of time required by a sensor, in this case a thermocouple, to reach 63.2% of a step change in temperature under a specific set of test conditions.

From the response time curves provided by *Omega* [73] and shown in Figure 3.18, it can be seen that the 3.175 mm (0.125 in)-diameter Type-R thermocouple has a time

constant of 12.4 s in air at room temperature and atmospheric pressure moving with a velocity of 19.8 m/s (65 ft/s).

Figure 3.18: Comparison of Time Constant Versus Overall Outside Diameter of Thermocouple in Air [73]

To estimate the time constant for these thermocouples under conditions closer to those expected during this experimentation, a comparison was conducted using the Nusselt numbers at both conditions, since it is known that the heat transfer occurring is related to the Nusselt number. It was found that the heat transfer between the air flow and the thermocouple will occur 1.21 times faster, resulting in a time constant of 9.8 s. Although this is an improvement on the time constant measured by *Omega*, it would still equate to a total experimentation time of over 35 minutes if the thermocouples were allowed to settle to 95.3% of the step change value. To determine the actual response time of the thermocouples, tests were conducted by subjecting the thermocouples to step changes by traversing the rake from a cooler spot in the exhaust temperature distribution at the 56° location in the combustion chamber exit plane to a hotter location at 54° at the

selected operating condition. The temperature was allowed to settle and the time constant was calculated to be approximately 5.93s.

Figure 3.19: Time Constant Predictions

Figure 3.19 provides a comparison between the actual exit temperature values obtained through experimentation and those predicted using the time constant, along with the uncertainty in the temperature reading at each time constant. From the graph it can be seen that there was only a 60 K difference between the temperature readings from the onset of stabilization at the 54° location and the thermocouples reached 100% of the total step change after 30 s. Further testing and calibration of the thermocouples will be discussed in subsequent sections.

3.4.5 Instrumentation and Calibrations

There are numerous different pressure transducers and thermocouples placed throughout the rig to monitor the various experiment parameters. The inlet air pressure was monitored through a static port in the inlet ducting wall, which was connected to a pressure transducer. Similarly, a venturi flow meter was connected to a differential

pressure transducer, which was used in calculating the experiment air mass flow rate. A turbine flow meter situated in the fuel piping was used to obtain the fuel flow rate and the fuel pressure was measured using a pressure transducer mounted in the inlet of the fuel lining entering the test cell. The air and fuel temperature measurements for the test rig operating condition were obtained using Type-K thermocouples. The inlet air temperature was monitored by a Type-K thermocouple placed just downstream of the venturi flow meter, but before the test cell, and the exhaust gas temperature was measured by a Type-K thermocouple placed after the test cell enclosure in the exhaust ducting. The fuel temperature was also obtained by the same type of thermocouple, which was placed in the fuel supply line just prior to entering the test cell. An angular potentiometer was used to motor the position of the sweeper arm during the tests.

All of these instruments provided voltage outputs that were fed into a HP3852A *data acquisition* (DAQ) unit. A *general-purpose interface board* (GPIB) was used to create an interface between the DAQ and a standard desktop computer. The computer contained a *LabVIEWTM* program, which was used to convert all the instrumentation voltages to the measured parameters. Equations were created within the *LabVIEWTM* program to take the inlet temperature and pressure, and the differential pressure across the venturi flow meter, and continuously calculate the air mass flow rate at a sampling frequency of 1 Hz.

To convert all of the measured voltages into the desired experimentation parameters, calibration curves were created to relate the output voltages of the various instruments to the required parameters for display in *LabVIEWTM*. Each of these calibration curves and the equations used in the *LabVIEWTM* for the inlet air static pressure sensor, venturi differential pressure sensor, fuel pressure sensor, fuel turbine flow meter, and sweeper arm potentiometer were verified prior to testing, midway through testing and at the completion of testing to confirm the accuracy of the measured parameters.

3.4.5.1 Pressure Transducers

The inlet air static pressure was measured using a static port placed in the inlet duct, which was attached to an *Omega* PX4202 pressure transducer with a range of 0 to 5 psig, an output of 0 to 5 VDC and a reading accuracy of $\pm 0.25\%$ of the full scale value. This

sensor was made up of a strain gauge whose resistance varies with force and the sensor uses a mechanical-to-electrical converter to translate increases in pressure into an electrical resistance, which is then measured. This transducer was calibrated to measure the pressure relative to the atmospheric pressure; therefore it provides a gage pressure output. Therefore, prior to each experiment, the atmospheric pressure was measured using a barometer. This value was entered into the *LabVIEWTM* for the inlet air pressure to be calculated and used for the determination of the air mass flow rate. The calibration curve provided by the manufacturer for this sensor can be found in Appendix C and was used to convert the measured voltage outputs to pressure readings. To verify that the calculated values were accurate, tests were conducted prior to experimentation, midway through testing and upon completion of all testing. These calibration tests involved measuring the inlet air pressure at blower speeds ranging from 400 to 2700 rpm using a vertical water manometer and comparing these values to those measured by the pressure sensor at the same blower speeds. The comparison of the outputs can be found in Figure 3.20 with less than 1% error between the tests as represented by the marker size. The linear fit provides an indication of the accuracy of the transducer.

The air mass flow rate was continuously determined during experimentation using the inlet air pressure and temperature along with the readings obtained from the venturi differential pressure sensor, which was an *Omega* PX653 pressure transducer with a range of 0 to 25 inH₂O, an output of 1 to 5 VDC and a reading accuracy of $\pm 0.5\%$ of the full scale value. This pressure transducer consisted of two separate ports for measuring the pressure difference between two segments of a venturi that have a different aperture. The difference between the two pressures was directly proportional to the flow rate through the venturi tube. This was a capacitance pressure transducer, which typically consists of a diaphragm element that is exposed to the experimental pressure on one side and to a reference pressure on the other side. As the diaphragm moves according to the changes in experimental pressure, it causes the sensing elements to detect these changes, which correlates to a change in capacitance that is detected with a bridge circuit. These changes in capacitance are converted to an output voltage. The calibration curve provided by the manufacturer for this sensor can be found in Appendix C and was used to convert the measured voltage outputs to pressure readings.

Figure 3.20: Static Pressure Transducer Calibration – Measured Values and Instrumentation Outputs

Calibration tests similar to those conducted on the static air pressure sensor were carried out on this instrument involving a vertical water manometer attached to the high and low pressure ports to obtain the differential pressure readings at blower speeds ranging from 400 to 2700 rpm. The comparison of the measured values and instrumentation outputs can be found in Figure 3.21, and by conducting a linear fit, the accuracy of the pressure transducers has been confirmed. Again, the marker size is representative of the uncertainty and variation between the three tests. Uncertainty calculations for the air mass flow rate can be found in Appendix C.

The fuel pressure was monitored during this experimentation using a *GE Sensing* PTX1240 pressure transducer with a range of 0 to 300 psig, an output of 4 to 20 mA DC, and a reading accuracy of $\pm 0.25\%$ of the full scale value. Calibration of this instrument was conducted by the manufacturer prior to testing and was not confirmed during this experimentation since this parameter was used simply for monitoring purposes and was not employed in any experimental calculations. The manufacturer's calibration curve can be found in Appendix C.

Figure 3.21: Differential Pressure Transducer Calibration – Measured Values and Instrumentation Outputs

3.4.5.2 Fuel Flow Rate

The fuel mass flow rate was measured using a *FTI Flow Technology* FT4-8AEXS-LEA-5 turbine flow meter with a range of 0 to 230 mL/s, an output of 0 to 10 Volts, and an accuracy of $\pm 0.25\%$ of the full scale value. This sensor measured the fuel flow rate through the fuel line located just prior to the cell via a rotor that spins as the fuel passes through its blades. The rotational speed, a direct function of the fuel flow rate, was sensed by magnetic pick-ups, which was translated into a voltage output. The turbine flow meter voltage output was then converted to a mass flow rate using the flow meter calibration curve and the fuel density. The manufacturer's calibration curve is presented in Appendix C and, assuming a fuel density of 804 kg/m^3 as taken from [68], the voltage output can be converted to a fuel mass flow rate. Uncertainty calculations for the fuel mass flow rate can be found in Appendix C.

To determine if the instrumentation output was accurate, calibration tests were conducted prior to, during, and upon completion of all fuel nozzle tests. These tests were

conducted by attaching a test cell fuel line to a serviceable fuel nozzle, which was then inserted into a container that was seated on a digital scale. The fuel was sprayed for 120 s during which the instrumentation mass flow rate was recorded. The measured mass flow rate was then determined and compared to the recorded instrumentation output. This was conducted over a range of pressures from 483 kPa (70 psi) to 827 kPa (120 psi). The results of these tests can be seen in Figure 3.22.

Figure 3.22: Fuel Mass Flow Rate Calibration – Measured Values and Instrumentation Outputs

By conducting a linear fit using *ExcelTM* for Figure 3.22, the performance of the turbine flow meter has been established. The marker size is representative of the uncertainty between the measured values and instrumentation outputs, which was less than 1%.

3.4.5.3 Potentiometer

To monitor the position of the thermocouple rake during testing, a *potentiometer* (often referred to as *pot*) was mounted to the shaft of the sweeper arm. A potentiometer is a type of bridge circuit used for measuring voltages by comparing a small fraction of a known input voltage that can be precisely measured, and then the two circuits are balanced until

there is no current flow, and this voltage is measured. The input voltage to the potentiometer was altered depending on the location of the sweeper arm. To calibrate the position of the sweeper to the output voltage of the potentiometer, Kotzer [62] attached a digital protractor to the sweeper arm and circumferentially rotated the arm through its full range of motion.

Figure 3.23: Sweeper Arm Position – Measured Values and Instrumentation Outputs

The instrument output voltages and sweeper arm angles were recorded from -20° to 70° and his calibration results can be seen in Appendix C. He determined a relationship between the arm location and the potentiometer's output voltage and used this equation in *LabVIEWTM* to record the sweeper arm position. To verify the accuracy of this equation, a similar test was conducted by attaching a digital protractor to the sweeper arm and the arm was traversed through its entire range of motion. Measurements were taken at various locations and compared to the instrumentation output, which is represented by Figure 3.23. A linear fit was conducted, thus verifying the accuracy of the relationship developed by Kotzer [62].

3.4.5.4 Thermocouples

The four Type-R thermocouples used in the measurement of the combustion chamber exit temperatures were calibrated prior to testing and after the completion of every five test runs. The calibration was carried out by conducting ice bath tests. During the tests, distilled water and ice were stirred in a beaker and the temperature was allowed to stabilize. The four thermocouples were taken from room temperature and inserted into the beaker. Measurements were continuously taken for 60 s using the *LabVIEW™* software. The tests were completed twice and gave identical results. A representative test is presented in Figure 3.24. Examining this figure, the temperature measurement for the fourth thermocouple was slightly above 273 K, which is due to the fact that it was the last thermocouple to be calibrated and the ice was beginning to melt causing a temperature rise in the ice bath. Repeat tests were completed; alternating the order in which the thermocouples were tested and the last thermocouple placed in the ice bath during each test consistently measured a slightly higher temperature.

These thermocouples were manufactured to Special Limits of Error (SLE) of 0.1%, meaning they are the same as the standard thermocouple wire, but with the added feature of having a slightly better accuracy specification [73]. The time constant for these thermocouples were determined experimentally to be 5.93 s and, as seen in Figure 3.24, the calibration of these thermocouples was confirmed with all readings providing a stable representation of the reference temperature.

To determine the temperature fluctuation at a fixed position in the combustion chamber exit plane, the thermocouple rake was placed in the flow at 55° position and allowed to stabilize. Temperature measurements were then recorded for 75 s. This test was repeated three times and an example of the temperature fluctuation tests can be seen in Figure 3.25.

To determine if the temperature fluctuations followed a normal distribution, normal probability plots were created for all four thermocouples. Figure 3.26 provides an example of the distributions. Their linear nature indicates the temperature fluctuations followed a normal distribution. Using these plots, it was determined that there was a standard temperature measurement fluctuation of ± 6 K in a static location.

Figure 3.24: Thermocouple Ice Bath Test Results

Figure 3.25: Extended Exhaust Gas Temperature Measurements – Thermocouple Rake Position of 55°

Figure 3.26: Normal Probability Plots for Exhaust Gas Temperature Fluctuations – Thermocouple Rake Position of 55°

To investigate further the response time of the thermocouples, two separate tests were conducted. The first involved the effects of the addition of cooling water to the thermocouple mount on the temperature readings of the thermocouples under normal test conditions. The test involved positioning the rake at the 55° position and ramping up the air mass flow until it reached the operating parameter at 0.98 kg/s. The air flow was allowed to stabilize and the thermocouple temperature readings were monitored for 60 s at which point water was added to the rake at a mass flow of 0.1 kg/s and a temperature of 290 K. The thermocouple temperatures stabilized within 30 s and experienced a temperature reduction of only 0.2 K, which can be seen in Figure 3.27. Therefore, it was confirmed that the conduction between the water and the thermocouple sheath had little impact on the temperature reading and would be considered negligible for these tests.

Figure 3.27: Effects of Thermocouple Mount Cooling Water on Temperature Readings

The second test involved subjecting the thermocouple rake to a step position change and monitoring the change in temperature and thermocouple response time. This test was carried out by first bringing the rig up to its operating point with an air mass flow rate of 0.98 kg/s, a fuel mass flow rate of 9.5 g/s for an equivalence ratio of 0.28. The rake was moved to the 56° position and the temperatures were allowed to settle for 60 s, after which the rake was repositioned to the 54° location. Trials of step position changes were also conducted from 34° to 32° and 2° to 0°. The temperature of the thermocouples did not settle instantaneously and it took approximately 15 s for them to reach 95.3% of the step change in temperature, as demonstrated in Figure 3.28. This was taken into account during the experimentation process and the thermocouples were allowed to settle for 20 s prior to taking any temperature readings.

Figure 3.28: Thermocouple Response to a Thermocouple Rake Position Change from 56° to 54°

3.5 Scaling Parameters and Operating Conditions

The majority of experiments involving the study of gas turbine engines involve the use of scaled apparatus [1]. This allows the researcher to conduct representative tests at a fraction of the cost of running full-scale tests. Numerous tests have been conducted on the T56 engine involving scaled models of the combustion section, which has typically been geometrically scaled to a 60° sector to permit the testing of only one combustion chamber. It is imperative that the experimental setup is as realistic as possible to reproduce what is actually occurring in the engine. This not only involves the physical orientation and size of the test rig, but also the parameters at which the rig is operated. Skidmore [54, 55] and McCabe [74] conducted scaled water model tests to examine the flow within a T56-A-15 Series III combustion liner by using only one combustion chamber, thus greatly reducing the amount of air and fuel required for their testing. By using this configuration, they were able to obtain results more quickly and effectively than if they had run full-scale tests. A similar scaling practice was used in this research.

To closely model the most common aircraft operating regime, the typical military cruise condition of 463 km/hr (250 knots) at 7650 m (25,000 ft) was selected as operational control for this research. Correctly modeling the test rig conditions is imperative to this research since it allows the experiment to yield scaled results of an actual operating condition for the comparison of the combustion chamber exit temperature profiles produced by fuel nozzles of varying physical condition. Since the test rig was operating at ambient inlet air temperature and pressure with a maximum possible airflow of 0.51 kg/s for a single can, the combustion chamber inlet temperature and pressure of the actual engine during the military cruise condition is much higher and the exact temperature and airflow distribution is not achievable in an atmospheric test rig.

When using an ambient inlet air temperature and pressure experimentation rig, only the fuel and air mass flow rates can be controlled. Jermy *et al.* [75] provided a detailed analysis of the various parameters that can be used for scaling rig operating conditions with reduced air and fuel flow rates below typical engine values, but with conserving certain flow quantities. Their strategy focused on ensuring that the mixing of the air and fuel were identical in the rig and the engine, meaning that the air flow and fuel patternation should have followed or deviated along the same streamlines that they would have in the actual combustor. Following their strategy, three parameters were used to scale the operating conditions: *Mach number*, equivalence ratio, and fuel droplet size D_{32} .

Mach number, Ma , is the ratio of the air velocity to the local speed of sound, and provides a measure of the thermodynamic changes of high-velocity flow. To match the engine compressor exit conditions to the rig, a *Mass Flow Parameter* ($\dot{m}\sqrt{T}/P$) is often used. This is essentially equivalent to conserving the Mach number [75] and provides similar airflows in the test rig as those that are present in the actual engine, but at a reduced pressure. Guy and Allan [76] concluded that the air mass flow for a T56-A-15 single combustion chamber rig at a target condition of 7620 m (25,000 feet) and 463 km/hr (250 knots) airspeed was 1.22 kg/s. At the same operating conditions, Rizk *et al.* [5] determined the inlet pressure and temperature test conditions for a single can rig to be 464 kPa (67 psi) and 563 K with an air mass flow rate of 1.17 kg/s. To conserve this value for an ambient T56-A-15 combustion section rig, Guy and Allan [76] determined that a blower would have to deliver an air mass flow rate of 0.465 kg/s. A more realistic

calculation was conducted by Pucher *et al.* [70], which factored in a compressor efficiency of 85% and determined that under the test conditions, the matched air mass flow rate was 0.486 kg/s. Utilizing the air mass flow rate obtained by Pucher *et al.* [70] for a single combustion test rig that conserved the physical dimensions of the engine and factoring in that the test rig has the area of two T56-A-15 Series III combustion chambers, an air mass flow rate of 0.98 kg/s was selected.

The next parameter that was to be conserved was the equivalence ratio. Conserving this value between the engine and the rig will allow the temperature rise in the combustor to be matched. It is recognized however, that the test rig exit temperatures will be substantially less than the engine due to the reduced inlet air pressure and temperature in the test rig. The actual fuel flow delivered to a T56-A-15 engine will vary depending on the mission requirements. As the payload increases, more fuel flow will be required per engine to maintain the desired airspeed. The United States Air Force (USAF) C130 flight manual [77] provides the fuel flow rate supplied to the engines at any operating condition. At a standard cruise condition of 7620 m (25,000 ft) and 463 km/hr (250 knots) airspeed, Taylor [78] determined that the air mass flow rate in a T56-A-15 engine is 7.66 kg/s equating to 1.28 kg/s per combustion chamber. Using this value and the fuel flow rates from USAF flight manual [77], it was determined that at this cruise condition the equivalence ratio can vary from 0.24 to 0.38 depending on payload. An equivalence ratio of 0.28 was selected for this research, which correlates to a fuel mass flow rate of 9.5 g/s. This value is somewhat higher than equivalence ratio selected by Kotzer [62] and Rizk *et al.* [5] due to the fact that a higher fuel delivery pressure was required to examine the effects of the spray characteristics being produced by both the primary and secondary ports of the T56-A-15 fuel nozzles. After conducting extensive fuel flow rate tests at various delivery pressures for which the results are presented in the next chapter, it was found that this fuel flow rate occurred at a fuel delivery pressure of approximately 517 kPa (75 psi). This value varied slightly between the nozzles. Equivalence ratio and fuel flow rate calculations can be found in Appendix C.

The final parameter that was to be conserved was the fuel droplet Sauter mean diameter or D_{32} . Matching the D_{32} and fuel distribution between the test rig and the engine ensures that the droplet evaporation and residence times are the same. By

matching the Mach number between the rig and the engine, the patterning of the fuel is essentially preserved. Conserving the droplet evaporation rates is more difficult due to the fact that the test rig will have a much lower inlet air temperature and pressure than the air exiting the compressor of a real engine, therefore it is important to ensure the droplet sizes are small enough to achieve adequate evaporation and mixing, thus minimizing any effects downstream in the combustor exit temperature distribution. Using Equation 2.10, the engine D_{32} and the test rig D_{32} can be predicted. For the selected operating condition, the fuel mass flow rate was found to be 27 g/s as determined from the charts found in [77], which equated to a fuel pressure of 965 kPa (140 psi), making the primary and secondary port pressures 965 kPa (140 psi) and 517 kPa (75 psi), respectively. The combined engine D_{32} was determined to be 42 μm , which coincides with the values obtained by Rizk *et al.* [5] and Kotzer [62]. Calculating combined D_{32} values involves amalgamating the primary and secondary D_{32} values, taking into consideration the difference in fuel flow rates between the primary and secondary ports. Appendix C provides a detailed calculation of the test rig combined D_{32} . At a fuel delivery pressure of 517 kPa (75 psi), making the primary and secondary port pressures 517 kPa (75 psi) and 138 kPa (10 psi), respectively, the combined rig D_{32} was determined to be 110 μm . Guy [79] predicted a D_{32} of 104 μm for an ambient pressure test rig with operating parameters set to model a cruise condition. The differences between the predicted values for the engine and test rig are discussed in Chapter 5 of this report and can be related to the assumptions associated with Equation 2.10. Referring back to the study conducted by Odgers *et al.* [29], it was concluded that there was no measureable effect of fuel droplet size on combustion efficiency for droplets below 150 μm . Since the D_{32} values predicted for the engine and the test rig were well below the upper threshold determined experimentally by Odgers *et al.* [29], any minor differences between these values were deemed to have negligible effect on the combustion chamber exit temperature distributions.

Chapter 4

Experimental Test Procedures and Results

4.1 Fuel Nozzle Test Specimens

The fuel nozzle specimens selected for this research were used in the T56-A-15 Allison/Rolls Royce engine, which is the primary powerplant for the Canadian Forces CC-130 Hercules and CP-140 Aurora aircraft. Twenty-four fuel nozzles were obtained for this experimentation. Seventeen of the fuel nozzles were in-service components that were removed from several CC-130 aircraft undergoing preventive maintenance. They were acquired from the Canadian Forces Aerospace and Telecommunications Engineering Support Squadron (ATESS) prior to any inspection, cleaning, or repairs. Twelve of these nozzles had matched life cycles and were removed from the same engine to allow for comparison between nozzles that were operated in the same environment for an identical amount of time. Six additional fuel nozzles were also selected for this research; they were used in recent experiments by Guy [79] to examine the fuel spray pattern produced by serviceable and unserviceable fuel nozzles under both combusting and non-combusting conditions. One newly refurbished fuel nozzle was obtained from the fuel nozzle overhaul and repair facility operated by Standard Aero Ltd. as a basis of comparison for the rest of the nozzles. The fuel nozzles were divided into three categories to describe them during this research: *serviceable*, *unserviceable* and *unknown*.

All the fuel nozzles obtained from ATESS were initially in the unknown category and had completed at least one overhaul cycle, and shown to conform to the specifications listed in the engine maintenance manual [8]. The term unknown is used to describe these nozzles because they had not undergone any inspections or testing prior to being received for this study; therefore their physical condition was unknown to this author. These components had operating times ranging from 136 to 1000 hours since their last inspection. The nozzles used in the research conducted by Guy [79] were also classified as unknown. Once these nozzles were assessed for their physical condition, spray

characteristics, and run through the combustion chamber sector rig to obtain a combustion chamber exit temperature profile, they were returned to ATESS to undergo inspection and cleaning.

At the time of this research, ATESS conducted all initial inspections and testing of T56-A-15 fuel nozzles using an Allied Signal test assembly, part #284021-1-1 and MIL-C-7024D Type III calibration fluid. A nozzle was first cleaned using a soft nylon cleaning brush, part # 8448462 and Aerotech 2000 (Brulin 512M) carbon remover and then it was placed in the test assembly. The nozzle was tested to determine if the spray pattern was conical and uniform by rotating it by 180 degrees to observe the pattern shape and spray cone angle, which should be between 75 and 95 degrees [8]. All tests were conducted by trained Canadian Forces personnel at fuel delivery pressures of 345 kPa (50 psi), 690 kPa (100 psi), and 1624 kPa (250 psi) to ensure a uniform spray with only the primary orifice open and with both the primary and secondary orifices operative. Tests were also conducted to determine if the secondary valve opened at the designed operating point of 448 kPa (65 psi). Once ATESS completed the testing of these nozzles, and they were classified either serviceable or unserviceable, they were returned to this researcher for further testing. The newly refurbished nozzle that was obtained directly from Standard Aero Ltd was deemed to be in the serviceable category. Only one new nozzle could be obtained as a basis of comparison for this research. The fuel nozzle test specimens with their serial numbers, *Time Since Inspection* (TSI), and categories are listed in Table 4.1.

Prior to commencing any testing, the physical condition of each nozzle was examined, in particular they were inspected for any carbon build-up on the nozzle orifices and air shroud. Since the nozzles were in-service components for the Canadian fleet of CC-130 aircraft, it was not possible to remove the lock wire and examine the internal components for damage or blockages because this would require the item to be sent for overhaul. Therefore, only images of the nozzle face and orifices could be taken prior to testing and after they had been cycled through ATESS for inspection and cleaning. Figure 4.1 provides an example of two nozzles received directly from in-service engines. Appendix D contains images of all the nozzles prior to testing and post-cleaning.

Table 4.1: List of Fuel Nozzle Test Specimens

Serial Number	Part Number	TSI (hours)	Category	Reason/History
VN1AMT1839	23074274	136.6	Unknown	Removed during maintenance
VN1AMM1301	23074274	136.6	Unknown	Removed during maintenance
AGM291	5232105	341.1	Unknown	Removed during maintenance
2192	6809618	380.7	Unknown	Removed during maintenance
123BQ	6794320	410.4	Unknown	Removed during maintenance
94EX	6794320	410.4	Unknown	Removed during maintenance
AGK944	5232105	510.3	Unknown	Removed during maintenance
4298	5232105	510.3	Unknown	Removed during maintenance
31AG	6809618	800.2	Unknown	Removed during maintenance
AM8FM	5232105	824.4	Unknown	Removed during maintenance
48BM	5232105	824.4	Unknown	Removed during maintenance
VN1AMM0335	23074274	932.2	Unknown	Removed during maintenance
AG347G	5232105	942.8	Unknown	Removed during maintenance
14-FN	5231836	943.8	Unknown	Removed during maintenance
AG3599	5232105	943.8	Unknown	Removed during maintenance
AGH3055	6809618	1000.3	Unknown	Removed during maintenance
AGH4084	6809618	1000.3	Unknown	Removed during maintenance
AGL824	5232105	N/A	Unknown	Nozzle used in Guy's [79] research
83-BJ	52321975	N/A	Unknown	Nozzle used in Guy's [79] research
AG7553	6794320	N/A	Unknown	Nozzle used in Guy's [79] research
AG4306	6809618	N/A	Unknown	Nozzle used in Guy's [79] research
AG4309	6809618	N/A	Unknown	Nozzle used in Guy's [79] research
AG56522	5232105	N/A	Unknown	Nozzle used in Guy's [79] research
47M	6809618	0.0	Serviceable	Cleaned and Tested at ATESS
VN1AMT1839	23074274	0.0	Serviceable	Cleaned and Tested at ATESS
VN1AMM1301	23074274	0.0	Serviceable	Cleaned and Tested at ATESS
AGM291	5232105	0.0	Serviceable	Cleaned and Tested at ATESS
2192	6809618	0.0	Serviceable	Cleaned and Tested at ATESS
AGK944	5232105	0.0	Serviceable	Cleaned and Tested at ATESS
31AG	6809618	0.0	Serviceable	Cleaned and Tested at ATESS
VN1AMM0335	23074274	0.0	Serviceable	Cleaned and Tested at ATESS
14-FN	5231836	0.0	Serviceable	Cleaned and Tested at ATESS
AG3599	5232105	0.0	Serviceable	Cleaned and Tested at ATESS
AGH4084	6809618	0.0	Serviceable	Cleaned and Tested at ATESS
123BQ	6794320	0.0	Unserviceable	Failed at 50 psi for streaks
94EX	6794320	0.0	Unserviceable	Failed at 50 psi for streaks/voids
4298	5232105	0.0	Unserviceable	Failed at 50 psi for streaks /voids
AM8FM	5232105	0.0	Unserviceable	Failed test at 50 psi for streaks
48BM	5232105	0.0	Unserviceable	Failed test at 50 psi for streaks
AG347G	5232105	0.0	Unserviceable	Failed test at 50 psi for streaks
AGH3055	6809618	0.0	Unserviceable	Failed at 50 psi for streaks Failed at 250 psi for voids

Based on the examination of the pre-testing images, it can be seen that there was a large amount of carbon deposited on both the primary and secondary orifices, as well as, the face of the air shroud. The obstruction of nozzle passages like the swirlplates and the primary and secondary orifices by gum and carbon deposits can be detrimental to nozzle performance because it will cause significant distortions in the spray patterns. The formation of hard carbon is directly related to the chemical composition of the fuel. Most aviation fuels are multi-component fuels consisting of hydrocarbon chains. When the fuel becomes heated, either by the flame radiation to the nozzle face, or by the circulation of the fuel through other systems to act as a coolant, the lighter components in the fuel are released and eventually small particles of carbon are left behind [13]. These particles can clog the internal passages of the fuel nozzle and deposit on the nozzle face. In addition to resulting in a non-uniform spray pattern, which could result in thermal stressing of the downstream components, the coke can eventually break off and led to erosion of the nozzle guide vanes and turbine blades [1]. Therefore, the amount of coke deposition on the nozzle face of the test specimens was qualitatively assessed based on the fuel nozzle service life, otherwise termed as the Time Since Inspection.

During the survey of the specimens' physical condition, it was found that fuel nozzles with service lives below 300 hours had significantly less coke build-up on the nozzle face and in both orifices as shown in Figure 4.1. The majority of the coke build-up for the nozzles with TSI's above 300 hours was located in the opening of the secondary orifice and did not vary greatly as the TSI increased. These nozzles were subjected to longer operating times and experienced thermal stressing for prolonged periods.

Upon examining the post-cleaning images of the fuel nozzle test specimens, no evidence of physical damage to the primary or secondary orifices was found except for minor abrasions in the surface finish on the secondary opening of several nozzles. A secondary finding of the physical survey of the fuel nozzles was a visible difference between the size of the primary and secondary ports for nozzles with different part numbers. It was found that the nozzles with the part number 23074274, like fuel nozzle VN1AMM1301, had slightly larger primary orifices and significantly smaller secondary orifices, as shown in Figure 4.2. Additionally, these nozzles had the least amount of visible carbon build-up in the openings of the primary and secondary ports.

Figure 4.1: Fuel Nozzle (a) VN1AMM1301 at 136.6 TSI hours and (b) AG3599 at 943.8 TSI hours

Figure 4.2: Fuel Nozzle (a) VN1AMM1301 and (b) AG3599 Post-Cleaning

4.2 Fuel Mass Flow Rate

All the fuel nozzles examined during this study were flow-tested by this researcher to determine their fuel mass flow rate prior to testing and post-cleaning. This testing was carried out as a confirmation of the fuel delivery pressure required to achieve the desired testing parameter of a fuel mass flow rate of approximately 9.5 g/s and to determine the

opening point of the secondary valve for each nozzle. Tests were conducted using a Swagelok® 1.1 MPa (160 psi) analogue pressure gauge, a digital stop watch, a digital mass balance, a fuel collection container and the fuel delivery system described in Section 3.2 of this report. The pressure gauge was calibrated just prior to testing by Guy [79] and results were used for this experimentation.

Fuel was supplied to the nozzle at pressures ranging from 138 kPa (20 psi) to 796 kPa (130 psi). The pressure was allowed to stabilize and then the nozzle was suspended in the fuel collection container, which was positioned on the mass balance. Fuel was allowed to accumulate for 120 s and then the accumulated mass was recorded. The uncertainty assigned to the digital scale was ± 5 g and the uncertainty of the time measurement was ± 0.25 s to account for any error associated with human response time, which resulted in a total fuel mass flow rate uncertainty of ± 0.1 g/s. An uncertainty calculation for the fuel mass flow rate can be found in Appendix C. Figure 4.3 provides a comparison between the mass flow rates of nozzle AGM291 prior to testing and post-cleaning.

Figure 4.3: Flow Characteristics of T56 Nozzle AGM291

An examination of Figure 4.3 reveals that the cleaned nozzle had a slightly lower fuel mass flow rate over the range of delivery pressures than the dirty nozzle. This was consistent with the comparisons completed for the rest of the nozzles with a maximum difference between the flow rates of a dirty and cleaned nozzle being only 6%. The largest variation of the flow rates between the different nozzles was observed for fuel delivery pressures ranging from 448 to 517 kPa (65 to 75 psi). This was primarily due to the point at which the secondary valve is designed to open, which is typically between 448 and 483 kPa (65 and 70 psi) [7]. Figure 4.4 provides a comparison between several different fuel nozzles from the three different categories operating at these fuel delivery pressures.

Figure 4.4: Flow Characteristics of Serviceable, Unserviceable and Unknown Nozzles at Opening of Secondary Valves

The largest deviation in fuel mass flow rate occurred between the serviceable and unserviceable nozzles at a fuel delivery pressure of 483 kPa (70 psi) with a standard deviation of 0.6 g/s. This operating point is typically only experienced for less than a few seconds during engine start-up at sea level conditions and is within the ± 3.06 g/s

tolerance allowed by Allison/Rolls Royce for the T56-A-15 atomizers [53]. Since all fuel flow rates measured were within this tolerance, the curves shown in Figure 4.3 can be considered equal and despite these negligible differences in fuel flow rates, all the nozzles' secondary valves opened and were fully functioning prior to reaching a fuel delivery pressure of 483 kPa (70 psi). Additionally, all of the serviceable, unserviceable, and unknown nozzles demonstrated the desired testing fuel flow rate of 9.5 g/s at approximately 517 kPa (75 psi).

4.3 Fuel Nozzle Spray Characteristics

4.3.1 Open Air Patternation

To determine the spray characteristics of each nozzle, open air patternation was conducted using a *Nexum* optical patternator (OSPA) and the fuel delivery system described in Section 3.2 of this report. The validity of open air patternation, as being representative of engine conditions was established by Guy [79]. In that work, he concluded that qualitative features are maintained despite the presence of shroud air and other flow characteristics of the combustion chamber. All nozzle spray patterns were obtained by placing the nozzle face 25.4 mm (1.0 inch) above the laser light sheet and the fuel delivery pressure was allowed to stabilize for 15 s prior to capturing the images. The images were obtained at three different fuel delivery pressures: 345 kPa (50 psi), 517 kPa (75 psi), and 620 kPa (90 psi). These delivery pressures were selected to observe the spray pattern produced by only the primary orifice, at the testing operating pressure, and well after the opening of the secondary nozzle. Once the images were captured, they were manipulated using *Nexum* Research Corporation software to correct for the camera angle (geometry) and light intensity (scattering and attenuation). The software used a scaling factor, or *zoom*, to obtain images of approximately the same size for the three different pressures. A scaling factor of 1.0 was used for a fuel delivery pressure of 620 kPa and 1.1 for delivery pressures of 345 and 517 kPa, since the sprays produced were notably smaller at these pressures.

Once all the spray images for the unknown, serviceable, and unserviceable fuel nozzles had been captured, they were analyzed for the spray diameter, cone angle, uniformity and variations in intensity using *Nexum* Research Corporation image analysis

software and results are shown in Appendix D. The units for the spray diameter were originally provided in pixels, but were converted to SI Units using the geometry scale factor in the software.

Using the spray diameter and the known nozzle height, as depicted in Figure 4.5, the spray cone angle can be calculated using the following relationship:

$$2\theta = 2 \arctan\left(\frac{d_{\text{spray}}}{2l_{\text{nozzle height}}}\right) \quad (4.1)$$

Figure 4.5: Spray Geometric Parameters

In addition to using the spray diameter to determine the spray cone angle, it can also be used as a measure of the uniformity or symmetry of the spray. The *Nexum* software is used to generate both a perfectly round perimeter around the spray based on the average spray diameter and a perimeter of the actual spray, which can be seen in Figure 4.6. This *actual* perimeter is determined by locating the maximum intensity in the spray pattern along a radius and then repeating this measurement 180 times at equal angular intervals. A total of 180 points are produced at two-degree intervals which define the perimeter of the spray. From this analysis, four parameters are then extracted to describe the quality of the spray. The first two describe the variation of the spray from perfectly round and the last two describe the variation in intensity around the perimeter of the spray. The first parameter given is the Root Mean Square (RMS) Out of Round variation between the actual perimeter and the perfect circle. The second is the maximum variation between the measured radius and the perfect circle (Maximum Out of Round). The next two parameters involve determining the intensity profile along the spray perimeter, which is

first averaged to eliminate slow variations due to imperfect image corrections (attenuation and scattering). This is highlighted in Figures 4.7, 4.8, and 4.9, which clearly show the smoothed intensity profiles for representative fuel nozzles in each category. The RMS variation between the measured intensity and the moving average is reported as the RMS Streakiness. In addition, the maximum variation in the intensity from the moving average is reported as the Maximum Streakiness. In the case of both the radial and intensity measurements the Maximum value is a more precise measure for detecting sharp variations indicative of streakiness. These four parameters would have ideal values of 0.0 indicating no radial or intensity variations along the perimeter of the spray.

Figure 4.6: (a) Serviceable, (b) Unknown, and (c) Unserviceable Measured and Perfectly Round Spray Perimeters at a Fuel Pressure of 517 kPa

Figure 4.7: Serviceable Nozzle Variation in Spray Intensity at a Fuel Pressure of 517 kPa

Figure 4.8: Unknown Nozzle Variation in Spray Intensity at a Fuel Pressure of 517 kPa

Figure 4.9: Unserviceable Nozzle Variation in Spray Intensity at a Fuel Pressure of 517 kPa

The results of these spray characteristics for the three categories of nozzles are summarized in Appendix D and Figures 4.10 to 4.15 provide a comparison of the typical spray patterns from several nozzles within each category at different fuel delivery pressures.

From examining Figures 4.10 to 4.15, which are representative of the majority of the results obtained for each nozzle category, it was evident that the fuel nozzles that came directly from in- service engines (unknown) and several of the unserviceable nozzles displayed the most non-uniformities in their spray patterns resulting in streaks and voids at the fuel delivery pressures examined. Fuel nozzles from all three categories had similar fuel spray cone angles at all three delivery pressures with larger angles at 345 and 620 kPa and a dip in the spray cone angle at 517 kPa just after the opening of the secondary valve, as shown in Figure 4.10. The serviceable and unserviceable fuel nozzles had larger spray cone angles at all fuel pressures, with the exception of a few of the unknown nozzles at the highest fuel pressure.

Figure 4.10: T56 Fuel Nozzle Spray Cone Angle Comparison

Figure 4.11: T56 Fuel Nozzle Spray Diameter Comparison

The unknown fuel nozzles had the largest variation in spray diameters at the highest fuel delivery pressure, as shown in Figure 4.11. The majority of the spray angles observed were within the acceptable limits of 75 to 95 degrees, as dictated in the engine inspection manual [7] with the exception of a few of the unknown and unserviceable nozzles which had spray angles in excess of the prescribed limits.

To assess the uniformity of the spray, the Maximum Out of Round parameter was selected for comparison between the nozzles, since it provided an indication of the maximum deviation from a perfectly symmetrical and circular spray. Figure 4.12 provides a representative comparison between the three categories of nozzles, clearly showing that the unknown and unserviceable nozzles consistently produced sprays that had visible deviations from a circular pattern at all fuel delivery pressures.

Figure 4.12: T56 Fuel Nozzle Max Out of Round Comparison

The serviceable nozzles typically remained between 5% and 10% deviation at the fuel delivery pressures examined, indicating a nearly perfectly round spray. The unknown nozzles' deviation from round was most evident at fuel delivery pressure of 345 and 620 kPa with deviations in excess of 24%, indicating that both the primary and secondary

orifices were producing sprays with streaks and voids causing them to be skewed from a perfectly round spray. The majority of the unserviceable nozzles that had been cycled through the cleaning process at ATESS exhibited nearly round spray patterns at the lower fuel delivery pressures, but became increasingly out of round as the spray from the secondary orifice progressed to being fully developed, as indicated by an average deviation from round of over 16% and as high as 30%.

Figure 4.13: Max Out of Round for Representative Fuel Nozzle Examples at 517 kPa Fuel Pressure

At the experimentation fuel pressure of 517 kPa, Figure 4.13 clearly showed the highest deviations from round occurring in the unknown fuel nozzle category with a standard deviation from the mean value of 0.04 as compared to 0.01 and 0.02 for the serviceable and unserviceable fuel nozzles, respectively.

The presence of localized fuel concentrations in the spray patterns were assessed by using the Maximum Streakiness parameter, which gave an indication of the largest localized concentration of fuel in the spray as compared to the overall average. Figure 4.14 provides a representative comparison between the three different categories of nozzles at different fuel delivery pressures. The serviceable nozzles displayed the most uniform distribution of fuel around the spray at all delivery pressures with the maximum streakiness remaining between 6 and 12%. The unknown nozzles displayed the largest variation in localized fuel concentrations at 345 and 517 kPa with values in excess of 30% and an average variation of 17%, but behaved similar to the serviceable nozzles at the higher fuel delivery pressure. The unserviceable nozzles exhibited a wide range of streakiness at all fuel delivery pressures, some representative of the values seen for the serviceable nozzles and some in excess of 35%.

Figure 4.14: T56 Fuel Nozzle Maximum Streakiness Parameter Comparison

Figure 4.15: Max Streakiness for Representative Fuel Nozzle Examples at 517 kPa Fuel Pressure

Figure 4.15 provides a comparison between the three categories of the fuel nozzles at the fuel delivery pressure selected for this research, clearly showing a larger variation in streakiness for the unknown fuel nozzles, with a standard deviation of 0.05 compared to 0.01 and 0.03 for the serviceable and unserviceable nozzles, respectively.

4.3.2 Droplet Size Distributions

When conducting testing using scaled parameters, it is imperative to ensure that the predicted values coincide with the actual experimental values. This will generate valid comparative results for combustion chamber exit temperature distributions. To verify that the predicted droplet D_{32} of $110\ \mu\text{m}$ was accurate for a fuel pressure of 517 kPa in the test rig, the droplet sizes being produced by a serviceable (47M) and unknown (AG4306)

T56-A-15 fuel nozzle were measured at ambient pressure and temperature using an *Artium*TM PDPA. Measurements of the distribution of droplet sizes in terms of D_{32} were taken radially at 3 mm intervals from the spray centre out to 31.8 mm, at an axial distance of 25.4 mm from the nozzle discharge orifice. Figure 4.16 provide histograms of the droplet sizes produced by both nozzles at 25.4 mm from the spray centre. These are summarized in Figure 4.17 using a cumulative distribution of the average D_{32} at radial locations extending from the centre of the spray to the outer regions.

Figure 4.16: Droplet Size Distribution at 517 kPa Fuel Pressure
25.4 mm From Spray Centre

The unknown nozzle produced notably larger droplets closer to the spray centre and smaller droplets at 25.4 mm from the centre of the spray when compared to the serviceable nozzle. It is interesting to note that the D_{32} for both nozzles at all distances from the spray centre remained below the predicted droplet D_{32} of 110 μm , with values at the 25.4 mm location being 74 μm and 68 μm for the serviceable and unknown nozzles,

respectively. The maximum droplet size of all values observed at this location for both nozzles was $110\text{ }\mu\text{m}$. Based on the selected fuel pressure being very close to the fuel pressure required for opening the secondary valve, it is possible that the unknown fuel nozzle was only supplying fuel through the primary port, which would explain the lower D_{32} value at the outer edges of the spray.

Figure 4.17: Cumulative Distribution of Average D_{32} at Various Spray Radial Locations

4.4 Combustor Liner Temperature Distribution

To determine the heat transfer that was occurring at various locations along the combustor liner, several tests were conducted using *Thermal Paint Temperature Technology* (TPTT) KN8 multi-change thermal paint to gain an accurate representation of the combustor liner temperature distribution at the selected testing parameters. TPTT KN8 is an acrylic paint that is temperature sensitive and is designed to create an irreversible visual representation of the temperature contour patterns to which materials

are exposed at various operating conditions. The paint calibration details are provided in Table 4.2. The paint was first sprayed evenly over the entire surface of the combustion liner and allowed to dry for 30 minutes. The liner was then placed in an oven at 523K for 30 minutes to dry off any solvents and fuse the paint to the surface.

Table 4.2: TPTT KN8 Paint Calibration Temperatures in Degrees Kelvin

Color Name	Temperature [K]	Paint Calibration
Violet	488
Fawn	658
Cream	745
Blue	833
Semi Glaze	1161
Glaze	1183

Once the liner had cooled, it was placed in the test rig, as shown in Figure 4.18, combustion was initiated and the test rig was allowed to reach the selected testing parameters. Once the air mass flow and fuel mass flow stabilized, the rig was run for a total of 20 minutes to ensure that the paint had sufficient time to change colour completely. After the completion of the test run, the combustion liner was removed and the temperature contour pattern was examined and recorded using a digital camera. The liner was then media-blasted to remove the paint and the process was repeated for three subsequent tests. The four tests were completed to determine the differences between the liner temperature distributions for nozzles with serviceable and unserviceable spray patterns. Each nozzle was tested twice to confirm the repeatability of the results, which is shown by the similarities between the images in Figure 4.19 for the two tests conducted with the unserviceable nozzle. The liner temperature distributions produced by the unserviceable fuel nozzle were nearly identical between the two tests with only a 5%

increase in the area of the 833 K temperature region on the inner radius of the combustion chamber. Similar results were observed for the two tests conducted with the serviceable nozzle except there was negligible difference between the two temperature distributions.

Figure 4.18: Painted Combustion Liner Installed in the Test Rig

Figures 4.20 and 4.21 provide a comparison between the liner temperature distribution produced by the serviceable and unserviceable nozzles. An examination of the temperature distributions produced by both nozzles clearly showed the effects of film cooling on both the outer and inner radius views of the combustion liner. In the outer radius view of both images in Figure 4.20, there was a clear region in the centre of the transition zone that was significantly cooler. This was a direct result of the film cooling provided from the cooling baffle located in the top of the liner. The air entering these holes serves to reduce the amount of convection from the flame to the liner walls, thus preventing damage from overheating and prolonging component life.

Figure 4.19: (a) Outer Radius Trial 1, (b) Outer Radius Trial 2, (c) Inner Radius Trial 1, and (d) Inner Radius Trial 2 Liner Temperature Distributions for an Unserviceable T56 Fuel Nozzle

Figure 4.20: Outer Radius (a) Serviceable Nozzle and (b) Unserviceable Nozzle Liner Temperature Distributions (dimensions in mm)

Figure 4.21: Inner Radius (a) Serviceable Nozzle and (b) Unserviceable Nozzle Liner Temperature Distributions (dimensions in mm)

Similarly, in the inner radius view of both images in Figure 4.21, the temperature distribution was non-symmetrical partially due to the presence of film cooling and partially due to the air flow distribution within the can. Cool air enters the cooling channel located in the mid-plane of the liner providing film cooling to the liner wall, and enters the transition zone through a small hole located at the 12° position within combustion chamber exit plane. This small hole introduces a jet of cool air, which serves to reduce the gas temperature in this region and decreases the amount of convective heat transfer from the gas flow and flame, respectively, to the liner wall, thus producing a non-symmetrical temperature distribution about the centre axis of the combustor transition zone.

By comparing the temperature distribution resulting from tests of both a representative serviceable (47M) and a representative unserviceable nozzle (AM8FM), it was evident that the serviceable nozzle with a uniform spray distribution produced a more symmetrical temperature distribution about the centre axis as shown by the nearly identical areas of 745 K temperature on either side of the cooling baffle in the outer radius view. The unserviceable nozzle, which had a spray pattern with visible streaks and voids, produced a larger region of 745 K temperatures on the left side of the outer radius as compared to the right indicating a higher flame temperature in this zone and consequently a non-uniform flame temperature distribution. When examining the inner radius view of both cans, the serviceable nozzle produced only a slightly larger region of 833 K temperature and identical regions of 1161 K temperature when compared to the unserviceable nozzle. However, the regions of 833 K on either side of the mid-plane of the transition zone of the serviceable nozzle are much closer in overall area when compared to the unserviceable nozzle, indicating a more uniform temperature distribution and fewer hot zones. Again, the asymmetries of the temperature distributions on both the outer radius and inner radius of the liner were not only directly related to the fuel spray pattern produced by the nozzle, but they were also highly dependent on the air flow distribution in the transition zone. The effect of the off-centre location of the dilution zone holes and the amount of air entering each side of the transition zone on the liner and exit temperature distributions will be discussed in the next Section.

4.5 Combustion Chamber Exit Temperature Distribution

To ensure the consistency of the combustion chamber exit temperature distribution data obtained for each nozzle, the same test procedure was followed for each experiment. The nozzle was first placed in the test chamber and the fuel line was securely attached. The test chamber was then closed and power was supplied to the main electrical control panel, which routed power to the computer and data acquisition unit. The *Labview*TM and the *Si Programmer*TM were initiated and the daily atmospheric pressure was entered into the program, which was used to determine the air density for the air mass flow rate calculations. Power was then supplied to the fuel pump and fuel was circulated throughout the system and returned to the fuel tank. The blower was started and allowed to ramp up to 2800 RPM until the air inlet temperature stabilized. The thermocouple rake was moved to the 62° position and the cooling water was allowed to flow to both the thermocouple rake and exhaust water jacket. The fuel control valve was then switched from the recirculation cycle to allow the fuel to flow to the nozzle. Once the fuel flow was stabilized, the igniter in the test chamber was engaged and a stable and continuous flame produced. The fuel and air mass flow rates were increased until the selected testing parameters of 9.5 g/s and 0.98 kg/s respectively, were reached, resulting in an equivalence ratio of 0.28.

The fuel and air mass flow rates were allowed to stabilize and both programs were triggered to begin collecting data. The program initiated the motion of the thermocouple rake and measurements were taken at 1° increments around the exit passage of the combustion chamber until the thermocouple rake reached the -2° position just outside the combustion chamber exit regions. To obtain each measurement, the stepper motor moved the rake to the desired position and once at the correct location, the thermocouples were allowed to stabilize at the new temperature for 18 s, which was equivalent to three time constants. The data acquisition unit then logged the data, after which the stepper motor was triggered to move to the next position. This procedure was repeated until the thermocouple rake reached the -2° position at which point the fuel flow was no longer routed to the nozzle, and data acquisition ceased.

All the data collected by the data acquisition unit for each test run were exported to an *Excel*[™] spreadsheet, from which the average of the seven seconds of data at each position was determined and used as the temperature at that specific location. The raw data for a representative test run can be found in Appendix D. Using a similar procedure as Kotzer [62] and Pucher *et al.* [72], a 4 x 61 matrix of average temperatures at each location was created, which characterized the combustion chamber exit temperature distribution in polar coordinates. A contour map of the exit temperatures was produced from this matrix using a *Matlab*[®] program originally developed by De Freitas [80], altered by Kotzer [62], and again modified by this author to generate the desired graphical output. Amongst the 4 x 61 matrix, linear interpolations were effected by the *Matlab*[®] routine to produce the contour maps. Although this contour map does provide an excellent representation of the combustion chamber exit temperatures, it is limited radially by the positions of the first and fourth thermocouples. This can be seen in Figure 4.22, which shows a black portion around the perimeter of the generated contour map. This black area is representative of the area of the combustion chamber exit where temperatures could not be measured.

Figure 4.22: Representative Combustion Chamber Exit Temperature Contour Map – 47M

The area represented by the contour map is 4647 mm² and the total area of the combustion chamber exit plane is 6240 mm². In total, 25% of the exit area temperatures were not captured along the margins of the exhaust plane [62]. By not including the temperatures in this region, which would have a mean temperature slightly lower than the

overall mean temperature, a slight increase in the average temperature of the exit plane was expected. Additionally, Figure 4.23 shows the location of the struts and engine thermocouples in the combustion chamber exit plane. The thermocouple rake traverses behind these objects, therefore the temperature recorded in these areas was somewhat decreased, which induced a slight decrease in the calculated average temperature of the exit plane.

Figure 4.23: Definition of Landmark Positions in Combustion Chamber Exit Plane [62]

The combustion chamber exit temperatures were converted to non-dimensional temperatures to allow for comparison between nozzles. This was accomplished using Equation 4.2.

$$\theta = \frac{T_{measured} - T_3}{T_{adiabatic} - T_3} \quad (4.2)$$

The adiabatic flame temperature ($T_{adiabatic}$) for each test run was determined using the process outlined in Appendix A using an *Excel*TM analysis program and T_3 represents the temperature at the combustion chamber inlet. The inputs to this program were the chemical composition of the fuel, the average inlet air temperature, the average inlet air pressure, and the average equivalence ratio for each test run, with an output of an adiabatic flame temperature for that specific run. Since there were slight variations in the air inlet temperature, fuel mass flow, and air mass flow between tests, using the adiabatic

flame temperature to obtain non-dimensional temperatures provides a means of accounting for these variations, ensuring each fuel nozzle was compared using the same scale. The parameters used to calculate the adiabatic flame temperature are listed in Appendix D. It was found that the non-dimensional temperatures ranged from 0.02 to 1.6; therefore twenty equally spaced contour lines were selected to present a standardized format of the combustion chamber exit temperature distributions.

The fuel nozzles studied in the serviceable and unserviceable categories produced similar combustion chamber exit temperature distributions and representative examples of these distributions are presented in Figures 4.24 and 4.25.

Figure 4.24: Combustion Chamber Exit Temperature Profile – Serviceable Nozzle AG3599

Figure 4.25: Combustion Chamber Exit Temperature Profile – Unserviceable Nozzle 123BQ

Figure 4.26: Combustion Chamber Exit Temperature Profile –
Unknown Nozzle 94EX

Several distinctive characteristics of the temperature contour maps for fuel nozzles from these two categories were observed. At the 10° position, centred above the radial mid-plane of the profiles, there was a region of hot gas whose temperature was 20% to 30% above the average non-dimensional temperature. Moving closer towards the centre of the exit plane, still slightly above the radial mid-plane, a large zone of high temperature gas ranging from 30% to 45% above the average non-dimensional temperature was present from 15° to 43°. The hottest exhaust temperatures were centred at approximately 28° and 34° positions reaching values in excess of 50% and 53% respectively, higher than the average non-dimensional temperature. The hot zone at the 34° position was consistently larger than the zone located at the 28° position. A reduction in temperature at the 30° position was noted for all nozzles, which separated these two zones of extremely hot gases. Finally, a region of exhaust gas that was 15% above the average non-dimensional temperature was centred radially at the 53° position.

The fuel nozzles in the unknown category displayed temperature profiles somewhat similar to the other two categories with the exception of the range of temperatures observed and the size of the hot zones. Figure 4.26 provides an example of the exit temperature contour map produced by a nozzle in the unknown category. The hot zone located at the 10° position, was larger and with much higher temperatures ranging from 30% to 40% above the average non-dimensional temperature. The zones of high

temperature gas ranging from 30% to 45% above the average non-dimensional temperature were larger and spanned from 15° to 47°. The highest exhaust temperatures were centred at the 28° and 33° positions reaching 55% to 60% above the average non-dimensional temperature. These hot zones were notably larger than the zones produced by the serviceable and unserviceable nozzles. The region of hot exhaust gas that was centred radially at the 53° position was 22% above the average non-dimensional temperature and was again larger than those observed in the two other nozzle categories.

The temperatures from 0° to 3° and from 57° to 60° positions were markedly lower for all nozzles observed with values 75% below the average non-dimensional temperature due to the presence of the struts, as seen in Figure 4.23. Reductions in temperature were also noted along the outer radius of the exit plane at the 12°, 30°, and 48° positions, which coincided with the location of the engine thermocouples. As a whole, the temperature profiles produced by all three categories of nozzles displayed these trends and were non-symmetrical about the 30° position of the combustion chamber.

To confirm the repeatability of the results obtained, repeat tests were conducted on fuel nozzle 47M, which are presented in Figures 4.27 and 4.28. The two tests demonstrated consistent profiles in all areas, with minimal differences, thereby substantiating the repeatability of the experimental results. The high temperature zones at the 10° position were at the same radial location and were roughly the same size with a difference in area of less than 2%. The zone of temperatures ranging from 30% to 45% were both located from 15° to 43° with the zones being of relatively similar size.

The areas of highest temperature were centred at the 28° and 34° and were the same size. While the zone at the 30° location was extended slightly along the inner radius, this difference equated to only a 1% difference in areas. The location of the high temperature zone at the 53° position was consistent between tests and was of the same magnitude. Overall, the minor differences between the two test runs were so small they were deemed negligible and it was concluded that the test rig provided a consistent platform for determining the effects of fuel nozzle condition on the temperature fluctuations in the combustion chamber exit temperature plane.

Figure 4.27: Combustion Chamber Exit Temperature Profile – 47M Trial 1

Figure 4.28: Combustion Chamber Exit Temperature Profile – 47M Trial 2

Following a similar approach as Kotzer [62], the standard deviation of the temperature distributions at each location for the unknown fuel nozzles was used to determine the variation between the temperature profiles, which were then plotted to produce the contour map shown in Figure 4.29. The regions between the 15° to 28° and 32° to 45° positions had the smallest variation in temperature for all the test results. The zones of largest temperature variation were located from the 2° to 8° and 57° to 58° positions which can be attributed to the fit of the combustion chamber on the struts. The large temperature variation at the 2° to 8° positions can be related to the induction of

cooling air from the dilution zone cooling rib located on the bottom of the exit plane at the 6° position. Depending on any minor air flow fluctuations between the tests, this would cause slight variances in the amount of cooling air admitted at this location, which would account for the dissimilarities in the temperatures observed. Temperature variations were also noted at the outer radius and inner radius of the exit plane on either side of the 12°, 30°, and 48° positions, which correspond to the location of the engine thermocouples. These differences were a result of differences between the amounts of cooling air admitted to the thermocouple cooling channels due to minor fluctuations in the air flow rates between test runs.

Figure 4.29: Standard Deviation of the Unknown Nozzles' Temperature Profiles

The combustion chamber exit temperature profiles observed during this research possessed similar distributions as those measured by Kotzer [62]. The temperature profiles in the exit plane are directly affected by the mixing and air distribution within the combustion chamber. The results obtained by Kotzer [62] and this researcher both show how the airflow within the combustion chamber can affect the temperature distribution experienced by the hot section components. The dome of the T56-A-15 combustion chamber contains swirl vanes that induce a clockwise swirl in the airflow, as observed by Rizk *et al.* [5] and Skidmore [54]. The primary and secondary zones contain symmetrically placed air entrance holes, which would not affect the swirl of the airflow induced by the dome swirl vanes. The final air entrance holes contained in the dilution

zone of the combustion chamber are non-symmetrical with a 20 mm diameter hole located at the 2 o'clock position and a 15 mm diameter hole located at the 11 o'clock position. The addition of the asymmetric dilution air caused the region of hot exhaust gas at the 53° position to have a slightly lower temperature than the hot zone located at 10° position. A similar effect was observed by Kotzer [62]. Additionally, the highest temperatures were consistently observed just to the right of the 30° position, indicating the high temperature core flow was not centred, but was being forced slightly to the right. This observation was theorized by the author to be the result of the swirl in the airflow. This effect was also observed in the combustion chamber liner temperature distributions with the zone of highest wall temperatures being centred between the 30° and 60° positions.

After air was introduced in the dilution zone, the exhaust gases are forced through the constriction in the transition zone of the combustion chamber, or *fishmouth*, which resulted in the hot gases being forced above the centreline of the combustion chamber exit, with cooler zones being located at the root of the exit plane. Additionally, cooling air was introduced through the space between the engine thermocouple ferrules and the combustion chamber, as well as through the engine thermocouple cooling channels, causing cooler regions at the 12°, 30°, and 48° positions both at the inner and outer radius of the combustion chamber exit plane. These phenomena were also observed by Kotzer [62].

To determine the percentage of the exit plane area at specified intervals of non-dimensional temperature, histograms were created for each fuel nozzle test. A comparative graph is presented in Figure 4.30, which is representative of the results obtained for each nozzle category. These results were obtained using the information determined for the temperature contour maps, along with a *Matlab*® code developed by Kotzer [62], which was modified by this author. The data were categorized using twenty equally spaced bins of non-dimensional temperatures ranging from 0.02 to 1.6, representing the minimum and maximum measured data points.

The results for fuel nozzle 2192 were indicative of the majority of the specimens in the serviceable category. The fraction of exit plane area increased smoothly for these specimens with a rise in non-dimensional temperature until a maximum was reached at a

non-dimensional temperature 44% above the average temperature, after which it decreased rapidly by over 20% of the exit plane area. Approximately 26% of the exit plane area of these specimens ranged from 28% to 44% above the average temperature, which accounted for the largest percentage area at a specific temperature range. The fuel nozzles in the unserviceable category followed a similar pattern of temperature distribution as the serviceable nozzles as shown by the fraction of exit area histogram for nozzle 94EX.

Figure 4.30: Fraction of Exit Area at Specified Temperatures

Two unknown nozzles are represented in Figure 4.30. Fuel nozzle AGH4084 represents the unknown nozzles that were removed from in-service engines, which later tested serviceable at ATESS. Fuel nozzle 48BM represents the unknown fuel nozzles removed from in-service engines, which later tested unserviceable at ATESS. Both nozzles are indicative of the results obtained for these two subcategories within the unknown category of nozzles.

Figure 4.31: Cumulative Distribution of Exit Area at Specified Temperatures

Fuel nozzle AGH4084 followed a similar distribution as the specimens in the serviceable and unserviceable category, but with a slightly higher percentage of exit area at 45% to 60% above the average temperature. Fuel nozzle 48BM did not follow the same pattern of temperature distribution as the serviceable and unserviceable nozzles. Rather, these nozzles had a much smaller exit plane areas below 96% of the average temperature. After this point, the percent of exit plane area rose quickly, reaching a maximum with 24% of the exit plane area ranging from 28% to 44% above the average temperature, which accounted for the largest percentage area at a specific temperature range. The major difference between these nozzles and the other categories was the much larger percentage of exit area, approximately 12%, that was 45% above the average temperature. The cumulative distribution in Figure 4.31 clearly shows the larger exit plane area of lower temperatures of the unknown nozzle, which later tested unserviceable.

To provide a quantifiable comparison between the combustion chamber exit temperature distributions, the pattern factor for each test specimen was determined using

Equation 2.16. Figure 4.32 provides a comparison between each test specimen within the three categories.

Figure 4.32: Pattern Factor Summary for all Nozzle Categories

The combustion chambers in the serviceable category consistently displayed the overall lowest values of pattern factors ranging from 0.48 to 0.54, with a mean value of 0.50 and a standard deviation of 0.02. The fuel nozzles in the unknown category had the widest variation between pattern factors ranging from 0.42 to 0.56, with an average value of 0.52 and a standard deviation of 0.03. The unserviceable fuel nozzles had slightly higher pattern factors than the serviceable nozzles ranging from 0.51 to 0.56, with a mean value of 0.54 and a standard deviation of 0.02.

Chapter 5

Discussion

5.1 General Introduction

This Chapter will provide discussion on the experimental test rig setup and test methodology used to obtain valid and comparable results of the fuel nozzle test specimen combustion chamber exit temperature profiles. Additionally, qualitative and quantitative analysis of the effects of spray cone angle, spray symmetry and streakiness on pattern factor will be performed to gain a better understanding of which spray characteristics had the greatest influence on pattern factor. The final results obtained will be validated that they were representative of those expected for a T56-A-15 combustion section.

5.2 Experimental Test Rig

A combustion chamber sector rig was used as the primary means of determining the effects of fuel nozzle condition on the combustion chamber exit temperature distribution and pattern factor (PF). Therefore, during this experimentation it was of great importance to obtain scaled results of a realistic engine operating condition for the comparison of the combustion chamber exit temperature profiles produced by fuel nozzles of varying condition. Typical cruise at 7620 m (25000 ft) was selected as the engine operating condition for this research. For the experimentation rig testing, three parameters were used to scale the operating conditions: Mach number, equivalence ratio, and fuel droplet size (D_{32}). All calculations associated with the scaling process can be found in Appendix C.

5.2.1 Mach Number (Mass Flow Parameter)

The mixing processes in the combustion chamber play an important role in determining the temperature traverse quality at the turbine inlet. Therefore, a Mass Flow Parameter (MFP), which is essentially equivalent to the Mach Number, at the entrance of the combustion chamber was matched between the test rig and the engine to ensure the

mixing processes remained consistent. In general, any changes in the mixing processes can translate into increased or decreased evaporation rates and degradation in the mixing of the products of combustion with the dilution air, resulting in an undesirable combustion chamber exit temperature profile. Taylor [78] created a model to determine the air mass flow rate at the entrance of a T56-A-15 engine at various operating conditions. At the selected operating condition, for a single combustion chamber, the ambient temperature and pressure were predicted to be 454 K and 357 kPa, with an air mass flow rate of 1.23 kg/s. Using these values, a MFP ($\dot{m}_a\sqrt{T}/P$) for the engine was determined to be 7.36×10^5 (kg·K^{0.5}/s·Pa). Measured data from sample test results for fuel nozzle 47 M, with an air inlet temperature and pressure of 309 K and 118 kPa, and an air mass flow rate of 0.981 kg/s, were used to determine the test rig MFP, giving a value of 7.31×10^5 (kg·K^{0.5}/s·Pa). The MFPs were matched, with less than a 1% difference, indicating that the mixing processes within the test rig closely modeled those of the actual engine, providing an operating platform that was representative of a real engine for examining combustion chamber exit temperature profiles.

5.2.2 Equivalence Ratio

The test rig used in this study was designed for scaled down flow conditions, which required less fuel flow to conserve the fuel-to-air ratio of the real engine operating conditions. To match the fuel-to-air ratio in the combustion chamber between the test rig and the engine, the equivalence ratio was conserved. At the selected engine operating condition, the equivalence ratio can vary from 0.24 to 0.38 depending on payload, as taken from the USAF C130 flight manual [77]. An equivalence ratio of 0.28 was used for this research, representing a military cruise setting. Sample test results from fuel nozzle 47 M were used to verify the selected and actual equivalence ratios were matched. With a fuel flow rate of 9.49 g/s and an air flow rate of 0.979 kg/s, the test equivalence ratio was determined to be 0.283, representing less than a 1% difference between the selected and actual value, thus confirming that the fuel-to-air was equivalent between the test rig and the engine. However, it is recognized that the test rig exit temperatures were less than that of the engine due to the reduced inlet air temperature and pressure in the test rig.

5.2.3 Droplet Sauter Mean Diameter (D_{32})

The final parameter that was conserved between the test rig and the engine was the D_{32} , which ensured that the droplet evaporation time was matched. Using Lefebvre's equation for calculating D_{32} (Equation 2.10), at the selected operating condition, the engine D_{32} from the primary and secondary ports were found to be $39\text{ }\mu\text{m}$ and $69\text{ }\mu\text{m}$. Accounting for the difference in fuel flow rates between the primary and secondary ports, the combined engine D_{32} was determined to be $42\text{ }\mu\text{m}$. At the experimental operating condition, which was at a much lower fuel flow rate than the engine, the test rig combined D_{32} was determined to be $110\text{ }\mu\text{m}$. Guy [79] predicted a similar value of $104\text{ }\mu\text{m}$ for an ambient pressure test rig with operating parameters set to model a cruise condition. However, since the predicted value for the test rig was over two times the value for the engine, droplet size tests were performed using a PDPA to confirm these predictions. At the experimental fuel delivery pressure, it was found that the combined D_{32} for a serviceable nozzle was $74\text{ }\mu\text{m}$. These measurements were conducted at ambient pressure and temperature without any air flowing through the nozzle shroud. It has been reported by Wang and Lefebvre [19] that increases in ambient pressure from standard atmospheric pressure results in an increase in D_{32} . Additionally, Rizk *et al.* [5] noted that the addition of shroud air caused a slight decrease in the droplet size at low fuel delivery pressures near ignition (70 kPa) and an increase at higher pressures. Therefore, it was concluded that the combined D_{32} would increase in the test rig where sufficient shroud air is present and the ambient pressure is higher, making the measured value closer to the predicted value of $110\text{ }\mu\text{m}$. Larger droplets increase the evaporation rates of the fuel, thus affecting ignition, flame stability, combustion efficiency, and emissions. However, Odgers *et al.* [29] determined that most studies overestimate the evaporation time in a real combustor and the droplets are actually broken up into finer droplets, which become entrained in the recirculating flow, allowing sufficient time for evaporation. They concluded there was no measureable effect of fuel droplet size on combustion efficiency for droplets ranging from $14\text{ }\mu\text{m}$ to $150\text{ }\mu\text{m}$, indicating that the actual diameter of the droplets and their evaporation rates in a combustor rarely have an effect on the overall combustion efficiency. The final conclusion surrounding the values predicted for the combined engine D_{32} and the combined rig D_{32} was that both these values were well

below $150\ \mu\text{m}$; therefore, if they were not exactly matched, there was negligible effect on combustion efficiency and pattern factor.

5.2.4 Experimental Test Rig Improvements

During the process of determining the required test rig fuel mass flow rate and air mass flow rate to model a military cruise operating condition, it was discovered that the current combustion chamber sector rig was not capable of delivering the required air flow and the test thermocouples could not withstand the high temperatures expected with the selected equivalence ratio. Several improvements were carried out to expand the testing capabilities of this rig. A foam insert was added to the inlet duct of the test rig to redirect the air flow to the test section, which resulted in a 9% increase in the available air mass flow rate. Without this improvement, the selected air mass flow rate of $0.98\ \text{kg/s}$ was not achievable. Additionally, the temperatures expected at the selected equivalence ratio were above the maximum operating temperature of the Type-K thermocouples installed in the thermocouple rake. This was confirmed with several tests at the selected operating condition, which resulted in complete degradation of the Type-K thermocouples. Type-R thermocouples were installed, which increased the maximum operating temperature of the test rig by over $100\ \text{K}$. These new thermocouples were also more resistant to oxidation, which resulted in prolonged life and they did not require frequent replacement. The final improvement to the test rig was the automation of the test procedure with the installation of a stepper motor and modifications to the *LabVIEWTM* software. The process automation increased repeatability and consistency between tests by removing the human error associated with the movement of the thermocouple rake.

Using scaled operating parameters to model closely a real engine operating condition along with the improvements to the test rig provided a stable operating platform for obtaining representative combustion chamber exit temperature distributions produced by nozzles with varying conditions. During experimentation, the combustion chamber inlet air temperature ranged from $312\ \text{K}$ to $319\ \text{K}$, representing only a 2% difference between the tests. The air and fuel mass flow rates remained stable and consistent between the test runs, with standard deviations below $0.001\ \text{kg/s}$ and $0.08\ \text{g/s}$, respectively. The fuel mass flow rates ranged from 9.46 to $9.57\ \text{g/s}$ between test runs, representing only a 1%

difference between test runs. The air mass flow rates ranged from 0.978 to 0.985 kg/s, representing a variation of 0.7%. In total, these minor fluctuations in operating conditions translated to only 0.8% of the full-scale conditions. Test condition measurement uncertainties were determined from the sample calculations found in Appendix C and were found to be less than 1% of the full-scale values. Therefore, it was concluded that the test rig was a stable operating platform for obtaining repeatable results with minimal fluctuations in the operating conditions between tests.

During the course of this testing, it was found that the range of operating conditions that could be examined using this test rig was very limited. The primary deficiency of this rig was the maximum available output of the blower. As a result of this limitation, the maximum possible airflow for testing was 1.10 kg/s, which in turn restricted the fuel flow. At the maximum air flow, any increases in fuel flow above the value selected for this testing resulted in temperatures at the maximum operating limit of the Type-R thermocouples. The fuel flow rate of 9.5 g/s selected for this research occurred at a fuel delivery pressure of 517 kPa (75 psi), which is very close to the pressure required to open the secondary valve and is in the lower operating range for this nozzle. As a result, the fuel flow through the secondary port was minimal and any nozzle deficiencies or blockages in the swirlplates or passages of the secondary port would be less visible in the spray. Figure 5.1 provides an example of the spray patterns produced by an unknown nozzle (AG4309) at fuel delivery pressures of 517 kPa (75 psi) and 620 kPa (90 psi).

Figure 5.1: Fuel Spray Patterns for Unknown Nozzle – AG4309

When considering such figures, vertical dark striations are apparent. These are artefacts of the OSPA images and were not considered during the analysis of the streakiness index for each spray pattern. Examining the specific images shown in Figure 5.1, the streaks and non-uniformities were only slightly visible in the spray pattern at the fuel delivery pressure 517 kPa. However, as the fuel delivery pressure was increased to 620 kPa, the spray clearly became less symmetrical and displayed a larger concentration of fuel streaks. This observation was noted for several of the other fuel nozzle spray patterns. Since several of the nozzles displayed streaks that intensified as the fuel delivery pressure increased, indicating the spray from the secondary orifice was greatly influencing the spray pattern, it is recommended that further research be conducted at higher fuel delivery pressures to further examine the effects of fuel spray patterns with streaks and voids on the combustion chamber exit temperature profile. Due to the airflow limitations of the test rig, it is recommended that the fuel spray patterns from both the primary and secondary ports be examined separately. A large majority of the unknown nozzles displayed significant non-uniformities in their spray pattern at a fuel delivery pressure of 345 kPa (50 psi). It is recommended that this fuel delivery pressure be used for determining the effects of the fuel nozzle condition on pattern factor. For the testing of the fuel spray pattern produced by the secondary port, it is recommended that the primary port be blocked forcing a higher fuel delivery pressure through the secondary port only. Fuel delivery pressures in excess of 620 kPa should be used to ensure any streaks occurring as a result of blockages in the secondary orifice and swirlplates are intensified.

5.3 Fuel Nozzle Condition

The correlation between the different categories of fuel nozzles examined and the resulting pattern factor indicated that the fuel nozzle condition greatly influenced the combustion chamber exit temperature profile. The nozzles taken directly from in-service engines displayed the highest pattern factors and had the largest deviations from the overall mean value of 0.50. It was unclear however what caused these large deviations from the mean PF value. Therefore, to gain a better understanding of which spray

characteristics had the greatest influence on pattern factor, qualitative and quantitative analysis of the results were performed.

The optical patternation system used for examining fuel nozzles with varying condition demonstrated an excellent ability to capture the required data of the fuel spray pattern. To examine the fuel nozzle condition, qualitative assessments of the spray patterns were performed and allowed for the identification of characteristic differences between the spray patterns produced by the fuel nozzles in the *unknown*, *serviceable* and *unserviceable* categories. Table 5.1 provides representative examples of the spray patterns produced by nozzles in each category.

Table 5.1: Representative Fuel Spray Patterns

Nozzle	Fuel Delivery Pressure		
	345 kPa	517 kPa	620 kPa
48BM (<i>unknown</i>)
AGM 291 (<i>serviceable</i>)
AM8FM (<i>unserviceable</i>)

Analysis of these images showed that the unknown fuel nozzles displayed spray patterns that were the least symmetrical and contained large streaks and voids. These nozzles were removed from in-service engines and had visible build-up of carbon in the nozzle passages. Therefore, it was expected that their spray patterns would exhibit a

large amount of non-uniformities. The serviceable fuel nozzles were cleaned and tested by ATESS and were expected to display the most symmetrical and uniform spray patterns. The analysis of their spray patterns confirmed this assumption. An interesting finding occurred during the qualitative analysis of the unserviceable fuel nozzles. It was initially expected that these nozzles would display the most non-uniformities; however, this was not the case. Upon examining their spray patterns, it was found that the majority of the unserviceable fuel nozzles displayed spray patterns similar to the serviceable fuel nozzles with the exception of a few, which exhibited streaks and voids at fuel delivery pressures of 345 kPa and 620 kPa. This observation is considered to be the result of only partial removal of the majority of carbon and gum deposits from the nozzle passages and swirlplates during the cleaning and testing process at ATESS. It is possible that the unserviceable fuel nozzles that displayed visible streaks and voids had internal damage or still had partial blockages.

An additional finding was that these streaks seemed to be less visible at a fuel delivery pressure of 517 kPa. This observation was related to a noticeable decrease in cone angle at this pressure. Larger cone angles were observed at 345 kPa and 620 kPa, which can be clearly seen by examining the serviceable fuel nozzle spray patterns in Table 5.1. Lefebvre [1] suggests that the primary spray cone angle is often designed to be slightly wider than the secondary spray cone angle, allowing the two sprays to combine almost immediately after leaving the nozzle. Due to the low fuel delivery pressure being supplied to the secondary port at 517 kPa, any non-uniformity in the spray pattern from the secondary port would be minimized by the spray from the primary port. Increases in pressure would amplify any non-uniformity in the spray from the secondary port. Based on these findings, the images provided by the optical patternator proved to be very useful in the qualitative assessment of the fuel nozzle condition and these observations may not have been detected if another test method was used.

Although the investigation of the spray images proved to be very useful as an initial analysis tool for the nozzle condition, a quantitative assessment was performed to allow for correlation between the nozzle condition and the combustion chamber exit temperature profiles. The images were analyzed using *Nexum* Research Corporation image analysis software and three parameters were extracted for nozzle condition

comparison: spray cone angle, Maximum Out of Round and Maximum Streakiness. These parameters were used to determine the effect of nozzle condition on pattern factor. Table 5.2 provides a comparison of the values obtained for representative fuel nozzles in each category at the test fuel delivery pressure of 517 kPa.

Table 5.2: Quantitative Comparison of Nozzle Condition and Pattern Factor

Nozzle	Condition	Cone Angle (deg) ₁	Max Out of Round (%) ₂	Max Streakiness (%) ₃	Non-Dimensional Max Temp (K)	Non-Dimensional Average Temp (K)	PF ₄
AM8FM	Unknown	75.3	17.2	18.7	1.60	1.03	0.55
AGH3055	Unknown	77.4	15.2	17.3	1.52	1.02	0.56
48BM	Unknown	75.5	14.5	21.3	1.59	1.05	0.55
AG4306	Unknown	76.2	17.1	19.6	1.51	0.97	0.54
2192	Serviceable	79.3	8.1	10.0	1.52	1.00	0.49
AGM291	Serviceable	77.7	7.1	10.2	1.50	0.99	0.50
VN1AMM0335	Serviceable	83.0	7.6	10.9	1.49	1.01	0.48
VN1AMT1839	Serviceable	81.6	7.4	11.9	1.48	1.00	0.48
48BM	Unserviceable	78.0	9.5	15.0	1.53	0.99	0.53
AM8FM	Unserviceable	80.4	14.6	17.8	1.50	0.99	0.52

1 – 75 to 95 degrees is the nominal range [8].

2 – A value of 0.0 indicates no deviations in the spray pattern from perfectly round.

3 – A value of 0.0 indicates no streaks in the spray pattern.

4 – PF values ranging from 0.48 to 0.50 are considered low for the nozzles examined.

It is important to mention that only the worst case nozzles, those in the unknown and unserviceable categories that exhibited visible streaks, voids, and non-uniform spray patterns during the qualitative study were used for the quantitative analysis of the effects fuel nozzle condition on the pattern factor. Additionally, it should be pointed out that the fuel nozzles in the serviceable category displayed a typical value of 8% and 10% for the Maximum Out of Round and Maximum Streakiness parameters, respectively. These numbers were taken as the nominal values representative of an ideal spray distribution. Table 5.2 also provides the maximum and average temperature of the combustion chamber temperature profile for each test specimen.

To establish the effect of fuel nozzle condition on the combustion chamber exit temperature distributions and pattern factor, qualitative and quantitative comparisons were conducted between the spray characteristic parameters and the fraction of exit plane area at a specified non-dimensional temperature. The non-dimensional temperatures

were divided into four categories: under 0.76, 0.76 to 1.18, 1.18 to 1.43, and over 1.43, which represented low, medium, high, and very high temperature zones in the exit plane.

5.3.1 Spray Cone Angle Effect on Pattern Factor

The spray cone angle is an important spray characteristic because it dictates the degree of exposure of the fuel droplets to the surrounding air. Increasing the spray cone angle increases the droplets' exposure, thus improving atomization and mixing and reducing droplet evaporation times. A qualitative analysis of the effect of spray cone angle on the combustion chamber exit temperature was performed by comparing the quantitatively determined spray angle to the combustion chamber exit temperature distribution. An example of this comparison for an unknown and serviceable fuel nozzle can be seen in Figures 5.2 and 5.3.

These figures are representative of the qualitative assessments for nozzles in these categories. By comparing these figures, it was found that as the spray angle increased the area of exit plane at low and high temperatures increased and the area of the exit plane at the very high temperatures decreased. The large decrease in the very high temperature regions of the exit has been highlighted in Figures 5.2 and 5.3 to clearly show this change. To confirm this analysis, a quantitative comparison between the effects spray cone angle on the fraction of combustion chamber exit plane area at specified non-dimensional temperatures was completed.

Figure 5.2: Qualitative Assessment Example for Unknown Nozzle – 48BM

Figure 5.3: Qualitative Assessment Example for Serviceable Nozzle – VN1AMT1839

Figures 5.4 to 5.7 display the correlations between the fuel nozzle spray cone angle at the experimental fuel delivery pressure of 517 kPa and the fraction of the exit plane area at the specified non-dimensional temperature ranges.

Figure 5.4: Correlation of Spray Cone Angle to Fraction of Exit Plane Area for a Non-Dimensional Temperature Under 0.76

Figure 5.5: Correlation of Spray Cone Angle to Fraction of Exit Plane Area for a Non-Dimensional Temperature Between 0.76 and 1.18

Figure 5.6: Correlation of Spray Cone Angle to Fraction of Exit Plane Area for a Non-Dimensional Temperature Between 1.18 and 1.43

Figure 5.7: Correlation of Spray Cone Angle to Fraction of Exit Plane Area for a Non-Dimensional Temperature Over 1.43

From examining the trends in Figures 5.4 to 5.7, the initial qualitative analysis was confirmed. Figures 5.4 and 5.6 demonstrate that as the spray cone angle increased, so did the fraction of the exit plane at the low and high temperatures. A 10% increase in the spray cone angle equated to increases of 5% and 4% in exit plane of the low and high temperature zones. The fraction of the exit plane at medium temperatures was relatively unaffected by an increase in spray cone angle, as evident by the slope of Figure 5.5.

Examining Figure 5.7 revealed the largest effect of spray cone angle on the fraction of exit plane area at a specified non-dimensional temperature. A 10% increase in the spray cone angle led to decreases in the exit area at very high temperatures of over 15%. The test specimens with the lowest spray cone angles had the smallest areas at temperatures in the low and high zones and the largest in the very high zones. This can be related to degradation in the mixing of the products of combustion and the dilution air within the combustion chamber as seen by the reduction of the lower temperature areas and a less concentrated region of very high temperature gases. The test specimens with larger spray angles had the largest areas at temperatures in the low and high zones and the

smallest in the very high zones. This change in the exit temperature distribution can be attributed to the increased exposure of the spray to the surrounding air, improving atomizing, mixing, and evaporation rates, resulting in a more compact higher temperature core flow and cooler regions at the root of the combustion chamber exit plane. This was a more desirable temperature profile since it provides cooler temperatures at the root of the turbine blades where the physical stress is the highest.

The combustion chamber exit temperature profiles are an excellent means of providing an indication of the location of hot zones, which could cause damage to downstream components. The hot zones produced by the nozzles in the unknown category were noticeably larger than those produced by the serviceable and unserviceable nozzles, and had the highest maximum temperatures overall. To provide a quantifiable comparison between the effects of varying spray cone angles on the combustion chamber exit temperature distributions, the pattern factors for the representative test specimens were determined and are displayed in Figure 5.8.

Figure 5.8: Correlation of Spray Cone Angle to Pattern Factor

From examining the trend in Figure 5.8, it was concluded that the spray cone angle does have a measurable influence on the pattern factor. Although the maximum spray cone angle was only 10% larger than the minimum value, this translated into a 15% reduction in the pattern factor, giving a more desirable temperature profile for the downstream turbine blades. Any further reduction in the minimum spray cone angles observed during this research would lead to further degradation in the mixing processes and increased evaporation times, potentially resulting in delayed combustion. This has been shown to cause hot streaks in the combustion chamber exit temperature profiles, as observed by Guy [79]. Therefore, it was concluded that fuel nozzle spray cone angle does have a significant effect on the temperature distribution experienced by the nozzle guide vanes and turbine blades and should continue to be examined as part of the fuel nozzle condition inspection and testing process.

5.3.2 Maximum Out of Round Parameter Effect on Pattern Factor

Although the analysis of the spray cone angle led to the conclusion that this parameter had an influence on pattern factor, it was still unclear if it was the main parameter responsible for causing the unknown and unserviceable nozzles to have higher pattern factors than the serviceable nozzles. Therefore, qualitative and quantitative assessments of the effects of the Maximum Out of Round and Maximum Streakiness parameters on pattern factor were also performed. Spray symmetry is an important spray characteristic because it ensures an even distribution of fuel within the combustion chamber, eliminating pockets of highly concentrated fuel that could come into contact with the liner wall causing excessive soot formation or could later combust and translate into hot streaks. To determine the spray uniformity for each test specimen, the Maximum Out of Round parameter was used. This parameter provided a quantitative indicator of the maximum deviation of the measured spray from a perfectly symmetrical and circular spray. The nozzles that had been cycled through the cleaning and testing process, referred to as the serviceable and unserviceable test specimens, exhibited nearly perfectly round and symmetrical sprays, with the exception of a few of the unserviceable nozzles. The unknown nozzles that came directly from in-service T56-A-15 engines had a large range of deviations from perfectly round, exhibiting highly skewed spray patterns. The

quantitatively determined Maximum Out of Round for each nozzle spray pattern was used to compare the combustion chamber exit temperature distributions between nozzles of varying condition. Representative examples of the qualitative assessments for a serviceable and an unknown fuel nozzle can be seen in Figures 5.9 and 5.10. The *Nexum* software generated both a perfectly round perimeter around the spray, based on the average spray diameter, and a perimeter of the actual spray, which are shown in Figures 5.9 and 5.10 by the lines.

*Maximum Out
of Round = 7.1%*

Figure 5.9: Qualitative Assessment Example for Serviceable Nozzle – AGM291

*Maximum Out
of Round = 17.2%*

Figure 5.10: Qualitative Assessment Example for Unknown Nozzle – AM8FM

Carrying out a qualitative analysis of these figures revealed that as the spray became less symmetrical and more out of round, the region of lower temperatures greatly decreased and the region of very high temperatures greatly increased. The large increase in the very high temperature regions of the exit plane has been highlighted in Figures 5.9 and 5.10 to clearly show this change. To confirm this analysis, a quantitative comparison between the effects of the Maximum Out of Round parameter on the fraction of combustion chamber exit plane area at specified non-dimensional temperatures was carried out. Figures 5.11 to 5.14 display the correlations between the fuel nozzle spray pattern Maximum Out of Round parameter at the experimental fuel delivery pressure of 517 kPa and the fraction of the exit plane area at the specified non-dimensional temperature ranges.

Figure 5.11: Correlation of Maximum Out of Round to Fraction of Exit Plane Area for a Non-Dimensional Temperature Under 0.76

Figure 5.12: Correlation of Maximum Out of Round to Fraction of Exit Plane Area for a Non-Dimensional Temperature Between 0.76 and 1.18

Figure 5.13: Correlation of Maximum Out of Round to Fraction of Exit Plane Area for a Non-Dimensional Temperature Between 1.18 and 1.43

Figure 5.14: Correlation of Maximum Out of Round to Fraction of Exit Plane Area for a Non-Dimensional Temperature Over 1.43

Examining the trends in Figures 5.11 to 5.14 revealed the effects of spray symmetry on the combustion chamber exit temperature profiles and confirms the results of the qualitative comparisons. As the Maximum Out of Round parameter of the fuel sprays increased by 58%, the regions of low and high temperatures saw decreases of 5% and 9% and the region of very high temperatures experienced the largest change with increases of over 14%. Test specimens with the lowest deviations from round had the largest areas at temperatures in the low and high zones and the smallest in the very high zones. This confirmed the effective mixing of the hot combustion products with the cool dilution air. The test specimens with the largest deviations from round had the smallest areas at temperatures in the low and hot zones and the largest in the very high zones. This change in the exit temperature distribution can potentially be related to the distribution of fuel within the primary zone. Non-uniformities in the spray could result in regions of excessive or moderate fuel concentrations, which could translate into poor atomization,

longer evaporation times and delayed combustion. To quantifiably compare the effects of the Maximum Out of Round parameter on the combustion chamber exit temperature profiles, the pattern factors for the representative test specimens were determined and are displayed in Figure 5.15.

Figure 5.15 confirms that non-uniformities in the spray symmetry had a negative influence on the pattern factor. It was found that a 58% increase in the Maximum Out of Round parameter resulted in a 10% increase in the pattern factor, giving a less desirable temperature profile for the turbine blades. It was expected that such a large increase in this parameter would have a more significant effect on the pattern factor; however, this was not observed. Therefore, it was concluded that the fuel nozzle spray roundness and symmetry had minimal influence on the temperature distribution experienced by the nozzle guide vanes and turbine blades when compared to effects of spray cone angle.

Figure 5.15: Correlation of Maximum Out of Round Parameter to Pattern Factor

5.3.3 Maximum Streakiness Parameter Effect on Pattern Factor

During combustion, localized concentrations of fuel, also known as *fuel streaks*, can generate streams of high temperature combustion products that propagate the entire length of the combustion liner into the turbine section of the engine [1]. To determine the presence of streaks in the fuel spray patterns of the test specimens, the Maximum Streakiness parameter was used. This parameter compared the brightest intensity measured, i.e. largest fuel concentration, to the overall spray intensity average. A qualitative analysis of the effect of streaks in the spray patterns on the combustion chamber exit temperature was performed by comparing the Maximum Streakiness parameter to the combustion chamber exit temperature distribution. Figures 5.16 and 5.17 provide representative examples of the qualitative assessments performed for the fuel nozzles in serviceable and unknown categories with the fuel streaks highlighted by circles.

The qualitative analysis of the spray patterns revealed that the nozzles in the serviceable and unserviceable categories exhibited the least amount of intensity variations or streakiness with the exception of a few of the unserviceable nozzles. The unknown nozzles that came directly from in-service engines had a larger range of variations in the Maximum Streakiness parameter with the highest values observed.

Figure 5.16: Qualitative Assessment Example for Serviceable Nozzle – AGM291

Figure 5.17: Qualitative Assessment Example for Unknown Nozzle – 48BM

By comparing Figures 5.16 and 5.17, it was found that as the Maximum Streakiness of the fuel sprays increased, the area of exit plane at low and high temperatures decreased and the area of the exit plane at the very high temperatures increased. To confirm this analysis, a quantitative comparison between the effects of fuel streaks on the fraction of combustion chamber exit plane area at specified non-dimensional temperatures was completed. Figures 5.18 to 5.21 display the correlations between the fuel nozzle Maximum Streakiness parameter at the experimental fuel delivery pressure of 517 kPa and the fraction of the exit plane area at the specified non-dimensional temperature ranges.

Figures 5.18 to 5.21 presented clear trends for the effects of fuel streaks on the combustion chamber exit temperature profiles and confirms the results of the qualitative comparisons. As the Maximum Streakiness parameter of the fuel sprays increased by over 53%, the regions of low and high temperatures experienced decreases of 4% and 12% and the region of very high temperatures saw the largest change with increases of over 16%.

Test specimens with the least variations from the average intensity (i.e. minimal streaks) had the largest areas at temperatures in the low zone and the smallest in the very high zones. The test specimens with highly streaky sprays had the smallest areas at temperatures in the low zone and the largest in the very high zones. This change in the exit temperature distribution can be related to the localized fuel rich zones (streaks) produced by blocked nozzles.

Figure 5.18: Correlation of Maximum Streakiness to Fraction of Exit Plane Area for a Non-Dimensional Temperature Under 0.76

Figure 5.19: Correlation of Maximum Streakiness to Fraction of Exit Plane Area for a Non-Dimensional Temperature Between 0.76 and 1.18

Figure 5.20: Correlation of Maximum Streakiness to Fraction of Exit Plane Area for a Non-Dimensional Temperature Between 1.18 and 1.43

Figure 5.21: Correlation of Maximum Streakiness to Fraction of Exit Plane Area for a Non-Dimensional Temperature Over 1.43

Streaks cause the fuel to combust and burn beyond the intended primary and secondary zones producing large regions of very high temperatures that cannot be sufficiently thinned by the dilution air [1]. If there is still un-combusted fuel (most severe streaking cases) in the dilution zone, the dilution air can actually cause the fuel to combust in this region. This was confirmed by Guy [79], who observed excessive hot streaks in combustion chamber exit temperature distributions that were produced by implanted faulty nozzles. This is an undesirable situation that can lead to thermal stressing of the nozzle guide vanes and turbine blades. Similarly to the previous spray characteristics examined, the effects of the Maximum Streakiness parameter on pattern factor were quantifiably compared for the representative test specimens and are displayed in Figure 5.22.

Figure 5.22 confirms that the presence of fuel streaks in the fuel nozzle spray pattern has a negative influence on the pattern factor. It was found that a 53% increase in the Maximum Streakiness parameter resulted in increases in the pattern factor by over 14%, giving a less desirable temperature profile for the turbine blades.

Figure 5.22: Correlation of Maximum Streakiness Parameter to Pattern Factor

Similarly to the Maximum Out of Round Parameter, it was expected that such a large increase in this parameter would have a more significant effect on the pattern factor, but this was not observed. It is important to reiterate that during the qualitative analysis of the fuel spray patterns it was found that several of the nozzles displayed streaks that intensified as the fuel delivery pressure increased above the selected testing pressure, indicating the spray from the secondary orifice was greatly influencing the spray pattern. The selected fuel delivery pressure was within the lower operating range of this fuel nozzle and is typically only experienced during start-up conditions. The fuel delivery pressure for a typical military cruise condition is over 50% higher; therefore any blockages or defects in the fuel nozzle would be magnified in the fuel spray pattern at this higher pressure and could have a more profound influence on the pattern factor. Although the variation observed between the Maximum Streakiness parameter were minimal between the fuel nozzles examined, it was concluded that at higher fuel delivery pressures the spray patterns having a high streakiness index could have a larger negative influence on the temperature distribution experienced by the nozzle guide vanes and turbine blades. This observation also applies to the Maximum Out of Round parameter. Therefore, fuel nozzle spray patterns should continue to be assessed for symmetry and the presence of streaks and voids during the fuel nozzle condition inspection and testing processes.

5.4 Pattern Factor

To confirm that the values determined for the pattern factor during this investigation were representative of those expected for a T56-A-15 combustion chamber, a comparison was conducted with the values measured by Kotzer [62]. To this author's knowledge, investigations involving the exit temperature distributions and resulting pattern factors of the T56-A-15 combustion chamber are limited to the research conducted by Rizk *et al.* [5], Mulligan [35], and Kotzer [62]. Mulligan [35] examined the exit temperature profiles at the exit plane of the turbine of a complete T56-A-15 engine using infrared thermography. This method resulted in high resolution and no disturbances in the exit plane of the turbine section, but cannot be used for comparison with the results of this research since the combustion chamber exit temperature profiles were never captured.

Rizk *et al.* [5] utilized 10 equally spaced rakes containing three thermocouples each, resulting in a total of 30 temperature measurements, to investigate the exit temperatures leaving a single T56-A-15 combustion chamber in a sector rig. However, their measurements did not include the effects of the engine thermocouples and struts on the exit temperature profile of the combustion chamber. The engine thermocouples and struts act as shields for the thermocouple rake, redirecting the exhaust gases, and reducing the amount of heat transfer from flame radiation, thus resulting in lower temperatures behind these obstructions. Therefore, the results of the current research could not be validated using the results obtained by Rizk *et al.* [5]. Since the setup initially employed by Kotzer [62] along with modeling of a standard cruise operating condition were maintained for this research, his results were used for comparison to those obtained during this study. Kotzer [62] observed maximum and average temperatures of 1561 K and 1158 K. The mean of the maximum and average temperatures measured during this experimentation were 1646 K and 1192 K, respectively. The testing parameters of this experimentation involved a higher fuel and air mass flow rate than that examined by Kotzer [62]. This resulted in a higher equivalence ratio, which would lead to a larger temperature rise in the combustion chamber. Therefore, it was expected that the maximum and average temperatures observed would be slightly higher than those observed during Kotzer's research. The pattern factors observed during this research varied from 0.42 to 0.56, while Kotzer [62] measured values spanning from 0.40 to 0.60. Since these ranges demonstrated similar results between the two studies, it can be concluded that the results obtained during this research for the examination of the effects of fuel nozzle condition on pattern factor are representative of the typical values expected for T56-A-15 combustion chambers.

5.5 Summary

Based on the trends presented in Figures 5.8, 5.15, and 5.22, the spray characteristics produced by in-service fuel nozzles with visible carbon build-up had a negative effect on the turbine inlet temperature distributions. A reduction in spray cone angle was concluded to be the largest contributor to increases in pattern factor. Although the results obtained for the effect of asymmetries and streaks in the fuel spray also showed a

negative effect on pattern factor, additional research is required into these spray non-uniformities to provide a more thorough comparison between the fuel nozzle condition and combustion chamber exit temperature distributions. The fuel nozzles examined during this study produced noticeably skewed sprays with higher streakiness index at higher fuel delivery pressures than the one examined. To provide more conclusive results on the effect of fuel nozzle condition on the combustion chamber exit temperature distributions and pattern factor, further testing at higher fuel delivery pressures is required.

Chapter 6

Conclusions and Recommendations

6.1 Conclusions

The turbine of any gas turbine engine is responsible for power generation and any damages to this component can result in significant degradations in aircraft performance. Damage, either due to erosion or cracks, is often found on the nozzle guide vanes and first stage turbine blades of gas turbine engines and require extensive maintenance equating to increased operational down-time and high material and manpower costs. To better understand the cause of this damage, several studies have been conducted on the effects of mixing and combustion chamber geometry on the combustion chamber exit temperature profile, which is experienced by the hot section components. At the time of this study, there had been no known investigations of the effects of fuel nozzle spray patterns on the temperature profiles entering the turbine of gas turbine engines.

The objective of this study was to investigate the effects of fuel nozzle condition on the combustion chamber exit temperature profile entering the nozzle guide vanes and first stage turbine blades. To perform this research, an optical patternator was used to quantify the spray patterns from a series of T56-A-15 fuel nozzles with a range of service lives. An ambient inlet air pressure and temperature combustion chamber sector rig constructed with T56-A-15 engine components from the diffuser to the turbine inlet was used to capture the combustion chamber exit temperature distributions for each test specimen. Two major improvements over past configurations of the test rig were completed including the addition of a foam insert in the inlet duct to increase the available air mass flow rate through the test section and the replacement of Type-K thermocouples with Type-R thermocouples to increase the experimental testing temperature range. An operationally representative engine condition of typical cruise at 463 km/hr (250 kts) at 7620 m (25,000 ft) was modeled by matching the Mach number, equivalence ratio, and droplet size (D_{32}) at the combustion chamber entrance of the test rig. Using scaled values of a real engine operating condition ensured the mixing and heat and mass transfer

processes were matched as closely as possible. The scaling of the engine condition led to an air mass flow rate of 0.98 kg/s and fuel mass flow rate of 9.5 g/s within the combustion chamber, yielding an equivalence ratio of 0.28.

The population of fuel nozzles selected included 17 specimens, which were classified as having an unknown condition having been removed from CC-130 aircraft undergoing preventive maintenance, with service lives ranging from 136 to 1000 hours. These specimens tested either serviceable or unserviceable once they were cycled through the testing and cleaning process. An additional six fuel nozzles used in recent experiments by Guy [79] to examine combustive and non-combustive conditions were also examined and classified as having an unknown condition. One newly refurbished fuel nozzle was used as a basis of comparison for the serviceable fuel nozzles.

The main conclusion of this research was that deviations of the fuel nozzle spray pattern from a nominal distribution correlated to an increase in pattern factor. It was found that reductions in the spray cone angle, variations in the spray roundness, and streak intensities observed in the spray patterns of the fuel nozzles removed from in-service engines resulted in degraded mixing processes and increased fuel droplet evaporation times within the combustion chamber. As a result, large regions of very high temperature core flow and smaller regions of lower temperatures within combustion chamber exit plane were produced, subjecting the nozzle guide vanes and turbine blades to higher temperatures at their roots and tips, which could lead to cracks as a result of excessive stressing at these points. During the qualitative analysis of the unknown fuel nozzles, large carbon deposits were found on the nozzle face and in the nozzle passageways, which were concluded to contribute greatly to variations in the fuel nozzle condition and cause changes in pattern factor. The spray characteristic with the most measurable influence on pattern factor was the spray cone angle, with a 10% decrease resulting in a 15% rise in pattern factor. The effects of spray roundness and streak intensity had slightly less significant influences on pattern factor. A 58% increase in the Maximum Out of Round parameter only equated to a 10% increase in the pattern factor, while a 53% increase in the Maximum Streakiness parameter resulted in increases in the pattern factor by 14%, giving a less desirable temperature profile for the turbine blades. The majority of the fuel nozzles that had been tested and cleaned at ATESS presented

fairly well-mixed exit temperature profiles with smaller regions of very hot temperature flow and larger areas of cooler flow, resulting in the lowest average pattern factor of 0.50, which represents the uniformity in temperature distribution in the combustion chamber exit plane.

The results obtained during this study were consistent with the results of past research on the test article T56-A-15 combustion rig. The combustion chamber sector rig proved to be a very stable operating platform for this research. The pattern factors observed during this research varied from 0.42 to 0.56, while values ranging from 0.40 to 0.60 had been previously measured by Kotzer [62]. Since similar results were demonstrated between the two studies, the findings obtained during this research for the examination of the effects of fuel nozzle condition on pattern factor were representative of the typical values expected for a T56-A-15 combustion chamber.

6.2 Recommendations

During the course of this research, a number of recommendations were made for improvements to the test apparatus or for future exploration into the effects of fuel nozzle condition on the combustion chamber exit temperature distributions.

Although, the sample population of test specimens was quite large, totalling over 40, testing an increased number of fuel nozzles taken directly from in-service engines over an extensive range of service lives would further substantiate the results obtained during this research. Several trends including increases in carbon deposits, decreases in cone angle, increases in the Maximum Out of Round parameter, increases in streak intensities, and increases in pattern factor were all observed for the fuel nozzles in the unknown category which had relatively long service lives. However, very few fuel nozzles with low service lives were examined. Therefore, it is recommended that a wider range of nozzles with similar service lives be investigated to ensure the observed trends are accurate.

During this research, the assessment of the fuel nozzle condition was limited to a visual inspection of the amount of carbon deposits in the openings of the primary and secondary ports of the test specimens. Previous studies have indicated that the swirlplates within the nozzle become blocked with coke, causing substantial fuel streaking and damage to hot section components. It is recommended that future research

be conducted to examine the amount of coke deposits on the fuel nozzle swirlplates as a function of service life and if there are any effects on the combustion chamber exit temperature distributions.

An engine condition near maximum cruise was modeled for this research, which required scaling of the test conditions to ensure the mixing, and heat and mass transfer processes were matched as closely as possible to the real condition. Due to the limitation of the maximum blower speed of the combustion chamber sector rig, the maximum possible air mass flow rate was limited to 1.10 kg/s. An air mass flow of 0.98 kg/s was selected for this research and a fuel mass flow rate of 9.5 g/s, equating to an equivalence ratio of 0.28. The selected fuel mass flow rate was found to occur at 517 kPa (75 psi), which is just after the opening of the secondary port. During the optical patternation testing, it was found that a large number of the fuel nozzles tested displayed spray patterns with streaks that intensified as the fuel delivery pressure surpassed the selected test fuel pressure, indicating the carbon deposits in the secondary orifice were greatly influencing the spray pattern. To examine further the effects of fuel spray patterns with streaks and voids on the combustion chamber exit temperature profile, research should be conducted at higher fuel delivery pressures using the test rig. Since the maximum possible airflow of this rig is limited, the fuel spray patterns from both the primary and secondary ports may need to be examined separately. To accomplish this it is recommended that a fuel delivery pressure of 345 kPa (50 psi) be used for determining the effects of the primary orifice condition on pattern factor since a large majority of the nozzles displayed significant non-uniformities in their spray pattern at this fuel pressure. For testing the condition of the secondary orifice, it is highly recommended that a blower with a greater capacity be acquired to allow for increased air mass flow, thus an increased fuel flow rate at fuel injection pressures above 620 kPa to ensure any streaks occurring as a result of blockages in the secondary orifice and swirlplates are intensified. If it is not possible to increase the air mass flow rate of the test rig, it is recommended that the primary port be blocked forcing a higher fuel delivery pressure through the secondary port only. Although this is not representative of the actual spray pattern, it will give an indication of the effects of fuel sprays with high streak intensities on pattern factor.

The PDPA testing conducted during this research was used only as a means of verifying if the predicted D_{32} was accurate at the selected fuel delivery pressure. However, during this testing, notable differences were found between the droplet sizes produced by a serviceable and unknown nozzle. The author was unable explore these differences further, due to limitations of the PDPA and several difficulties encountered when attempting to focus the intersection of the laser beams on streaks or voids. It is recommended that further testing be conducted using a PDPA with a larger range of droplet diameters to determine the effects of voids and streaks on D_{32} .

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Appendix A

Combustion Review

A.1 Chemical Reaction

When examining gas turbine combustion, it is important to first have a basic understanding of the chemical processes involved in order to better understand the characteristics of combustion, like the adiabatic flame temperature and pollutant formation. For a mixture reacting at high temperatures, the major species break down or dissociate into minor species. Consider the following reaction of a generic hydrocarbon with air, assuming N_2 is inert:

and with dissociation of N_2

where N' represents the molar concentration of the species. These chemical reactions can be broken down into two phases. First, there is the heat release phase during which the ideal combustion products (typically CO_2 , H_2O , and O_2) dissociate and react to form H_2 , OH , CO and H . This phase neglects any N_2 dissociation since the concentration of N_2 will change very little and the concentrations of NO , N and O will be very small. The second phase of the reaction involves the formation of NO , N and O in the combustion products. Dissociation of the products can be modeled by application of the chemical equilibrium theory. To determine the adiabatic flame temperature of a mixture and the concentration of pollutants produced during the chemical reaction, the composition of the combustion products must be known. The chemical equilibrium theory basically states that the concentrations of the reactants and the products remain the same. For any reaction such as:

to be at chemical equilibrium the rates of the forward and reverse reactions must be equal [13]. In this chemical reaction, A and B are the reactant chemical species, C and D are the product species, and a , b , c , and d are the stoichiometric coefficients of the respective reactants and products.

For an equilibrium reaction to occur, the temperature and pressure are held constant, and both the First and Second Law of Thermodynamics must be satisfied [13]. The First Law accounts for the principle of energy conservation, and centres on the summation of the enthalpies of both the reactants and products being equal. The enthalpy of a system, H , is defined as the sum of the internal energy of the system, U , plus the system's volume, V , multiplied by the pressure exerted by the system on its surroundings, P , and is shown in the below equation:

$$H = U + PV \quad (\text{A.4})$$

To satisfy the First Law of Thermodynamics the following equation must be true:

$$H_{\text{reac}} - H_{\text{prod}} = 0 \quad (\text{A.5})$$

In order for the Second Law to be satisfied, the Gibbs free energy function obtains a minimum at equilibrium. Gibbs free energy is defined as:

$$G \equiv H - TS \quad (\text{A.6})$$

and a summation of the Gibbs free energy for the individual species, gives the Gibbs free energy of the mixture as shown in Equation A.7:

$$G_{\text{mix}} = \sum_i N_i [\bar{G}_i] \quad (\text{A.7})$$

where N_i is the molar concentration of the species. As H , T , and S are state functions, which means they are functions of any two independent thermodynamic parameters, the Gibbs free energy is also a state function. Therefore, Equations A.6 and A.7 are a function of pressure and temperature. At a given static temperature and using the thermodynamic relationship for enthalpy, Equation A.6 can be differentiated, and using the *Ideal Gas Law* becomes:

$$\bar{G}_i = \bar{G}_i^o + R_u T \ln \frac{P_i}{P^o} \quad (\text{A.8})$$

where the superscript o, represents standard-state condition, P^o is the reference pressure and P_i is the partial pressure. Substituting Equation A.8 into Equation A.7 gives the Gibbs function for a mixture of ideal gases:

$$G_{\text{mix}} = \sum_i N_i [\bar{G}_i] = \sum_i N_i \left[\bar{G}_i^o + R_u T \ln \frac{P_i}{P^o} \right] \quad (\text{A.9})$$

For a fixed temperature and pressure, the equilibrium condition is such that the derivative of Equation A.9 with respect to time will be equal to zero. A thorough derivation of this condition is provided in Turns [13]. By assuming a standard-state or *reference* pressure of 1 atm, the following equation for chemical equilibrium at constant temperature and pressure is obtained:

$$\Delta G^o = -R_u T \ln K_P \quad (\text{A.10})$$

where the equilibrium constants (K_P) of the reaction is defined as:

$$K_P = \frac{\prod_{\text{prod}} \chi_i^{N_i}}{\prod_{\text{reac}} \chi_i^{N_i}} \left(\frac{P}{P^o} \right)^{\sum_{\text{prod}} N_i - \sum_{\text{reac}} N_i} = \exp \left(\frac{-\Delta G_T^o}{R_u T} \right) \quad (\text{A.11})$$

By applying the atom balance equations for the reactants, the equations required for determining the number of moles for the C, H and O can be found. These equations can

then be used to determine the expressions for the number of moles of the major species in the products (CO_2 , H_2O and O_2). Using these equations, the expression for the total moles in the products can be found along with the mole fractions of the species in the products. By using the following equilibrium reactions for the dissociation of the products:

the equilibrium constants for each reaction can be found, along with the expressions for the standard-state Gibbs function change [13]. It can be seen that there are three unknowns (N'_{CO} , N'_{H_2} , N'_{OH}) in Equation A.1 that must be found in order to determine the adiabatic flame temperature. Referring back to Equation A.4, for a constant pressure process, the heat supplied to the system is equal to the increase in enthalpy [13]. The system's enthalpy is comprised of the enthalpy due to a change in temperature from a reference point, and the enthalpy stored as chemical energy. Therefore, the heat released as the products are formed can be determined. As the products of combustion are formed, the presence of high temperatures can cause the products of combustion to dissociate, greatly altering the final mixture composition, enthalpy change and adiabatic flame temperature [13]. The adiabatic flame temperature is the temperature which the flame would attain if the net energy released by the chemical reaction that converts the fresh mixture into combustion products were fully utilized in heating the products. At high-temperature flames, dissociation of the combustion products occurs to a significant extent and absorbs a large amount of the heat released from the reaction. At low temperatures of combustion of a stoichiometric or lean fuel-air mixture, it would be expected that only H_2O and CO_2 would be formed in the products. However, these products themselves are unstable and partly revert (or dissociate) to simpler molecular and atomic species and radicals like CO , H_2 , O , H and OH [13]. The energy absorbed by

this dissociation is considerable and its effect is to reduce substantially the adiabatic flame temperature [13].

Similarly, for the second phase of the reaction with the dissociation of N_2 , as shown in Equation A.2, the same process as described above is used to determine the number of moles of NO, N, and O. By applying the following equilibrium reactions:

the three unknowns (N'_{NO} , N'_N , and N'_O) can be determined as detailed above using the same method.

The adiabatic flame temperature can be determined by iteratively solving Equations A.5 and A.9 by systematically guessing the values of the unknowns described above until the conditions of both the First and Second Law of Thermodynamics are satisfied. This is a time-consuming process and it can be quite difficult to accurately predict the adiabatic flame temperature of an actual combustion reaction. Several simulation and computer programs have been developed over the past decade by Turns [13], Olikara and Borgman [81], and numerous others, to make this task somewhat less difficult and less time-consuming. When examining the models used to compute the flame temperature, it is clear that several assumptions are made which may cause the results to be somewhat limited to certain scenarios. The main assumption of using Equations A.5 and A.9 to determine the flame temperature is that the reaction occurs at constant pressure; however it is known that there is a slight pressure increase in a combustion chamber, but this increase is so small that it is often deemed negligible [1]. The next assumption is that the number of species that are considered to be in the products of combustion. Typically, eleven species are considered: H, O, N, H_2 , OH, CO, NO, O_2 , H_2O , CO_2 and N_2 [13]. Numerous sources, including Turns [13], and Gordon and McBride [82] suggest that since the molar quantities of some of these species in the mixture are so minuscule, they

will have little to no contribution to the enthalpy change and attention should be paid to only dissociation reactions of the major species being H_2O and CO_2 . The level of accuracy required will indefinitely determine the number of species examined.

There are three primary factors that affect the adiabatic flame temperature: the equivalence ratio, inlet air temperature and inlet air pressure. When examining reactions that are lean or have an equivalence ratio less than stoichiometric, it is known that there is incomplete combustion due to the presence of CO and H_2 in the products, which basically indicates that not all the fuel carbon was converted to CO_2 and not all of the fuel hydrogen was converted to H_2O [13]. As the reaction approaches stoichiometric, combustion is more complete, resulting in an increase in the mole fractions of CO_2 and H_2O . If the temperature is high enough, the major species will dissociate to form minor species in the products. For the heat release phase, the dissociation of N_2 is typically neglected and only slightly decreases as the reaction approaches stoichiometric [13]. Another major species of lean combustion is O_2 and as the reaction approaches stoichiometric, the mole fraction decreases. This is due to the increase in CO_2 and H_2O in the products, which requires the reaction of the O_2 with CO and H_2 , thus decreasing the mole fractions of O_2 in the products. *Le Châtelier's Principle* explains this phenomenon.

Le Châtelier's Principle states that any system that is initially in a state of equilibrium when subjected to change, like increasing pressure or temperature, will shift in composition in order to minimize the effect of the imposed change [13]. For an increase in pressure at a fixed temperature, the reaction will favour the side with less volume (i.e. fewer gas molecules occupy less volume) and the equilibrium will shift in the direction to produce fewer moles. This will be explained using equation A.14.

Changing the pressure will have an effect on this equilibrium reaction because there are only two molecules on the left side, but three on the right. An increase in pressure would move the position of equilibrium to the left.

Because this is an all gas equilibrium, the equilibrium constant, K_P , will be used to explain the shift:

$$K_p = \frac{(X_{CO} \times P)^2 (X_{O_2} \times P)^1}{(X_{CO_2} \times P)^2} \quad (A.18)$$

where P is pressure and X is the mole fraction of the species. It can be seen that most of the pressures cancel out with the exception of one in the numerator of the Equation A.18. K_p must stay constant, so in order to compensate for the pressure term, the mole fraction of CO_2 would have to increase and the mole fractions of CO and O_2 would need to decrease. Therefore, the position of equilibrium has moved to the left to keep K_p constant. This phenomenon is also true for most of the equilibrium reactions shown in Equations A.12 to A.16. Equation A.17 has an equal amount of moles on each side of the equation and therefore changing the pressure should have no effect on this reaction.

Unlike changes in pressure where K_p is kept constant, the equilibrium constants change as the temperature is increased or decreased. When the pressure is fixed and the temperature is increased, the position of equilibrium will move in such a way as to reduce the temperature again. This will be accomplished by favouring the reaction which absorbs heat (endothermic). By again examining equation A.5, it is known that heat is absorbed when CO_2 breaks down into CO and O_2 . Therefore, increasing the temperature produces a shift to the right and the products are favoured. In order to rebalance the equation, less CO_2 will be formed, and the equilibrium mixture will contain more CO and O_2 . Thus, there will be an increase in the equilibrium constant. This phenomenon is also true for most of the equilibrium reactions shown in Equations A.12 to A.16.

By applying *Le Châtelier's Principle*, it is known that as the pressure is increased at a fixed inlet air temperature, the adiabatic flame temperature also increases due to the increased suppression of the dissociation of the major species into minor species. At higher pressures, there is less dissociation of the products resulting in less heat absorption and an increase in the flame temperature. This is also true for changes in temperature. As the inlet air temperature is increased at a fixed pressure, the adiabatic flame temperature also increases due to an increase in the absolute enthalpies of both the products and of the reactants. Increasing the air inlet temperature causes a shift in the equilibrium equations towards the products, thus promoting the dissociation of the major species.

Having reviewed the theory behind chemical equilibrium, one might ask *why is this important when examining the exit temperature distribution of a T56-A-15 combustion*

chamber? Since the air inlet temperature and pressure for the experimental setup was fixed, only the equivalence ratio could be varied. An equivalence ratio of 0.28 was selected for this research, which coincided with an actual operating condition of this engine. This condition was very lean and would ultimately result in incomplete combustion, resulting in less dissociation of the products and an increase in the adiabatic flame temperature. It is known that adiabatic flame temperature has great influence over the rate of chemical reaction and thus the combustion efficiency; therefore it is an important parameter that can be used to examine experimental data. The equivalence ratio, inlet temperature and pressure were fixed for all experiments; therefore, the adiabatic flame temperature was used to obtain non-dimensional temperatures at each point to account for any fluctuations in the air mass flow, fuel mass flow and air inlet temperature between tests and ensure the combustion chamber exit temperature distributions were compared on an even scale. This concept was discussed in more detail in Chapter 4.

A.2 Chemical Reaction Rates

Chemical kinetics deals primarily with the experimental determination of reaction rates from which rate laws and rate constants are derived. Estimates of these rates along with the rates of droplet evaporation, t_e , and flame diffusion, t_d , are often used to determine the total time required for combustion, t_c , to occur, which is shown in the Equation A.19.

$$t_c = t_e + \Delta t_d + \Delta t_{cr} \quad (A.19)$$

The delta terms are present for the chemical reaction, t_{cr} , and diffusion times due to the fact that these processes begin immediately following the formation of fuel vapour. If one of these processes has a much slower rate than the others, then the total combustion time will be dependent on that individual process [29]. The main factors that influence the reaction rate include: the physical state of the reactants, the concentrations of the reactants, the temperature at which the reaction occurs, and whether or not any catalysts are present in the reaction [13]. Various methodologies exist for determining these rates with numerous assumptions that are taken into consideration.

A chemical reaction will proceed at a rate that is directly proportional to the concentrations of the two reacting species. Referring back to Equation A.1 for the reactants and products contained in the equilibrium reaction, the reaction rate (RR) is given by:

$$RR = -\frac{1}{a} \frac{dn_A}{dt} = -\frac{1}{b} \frac{dn_B}{dt} = -\frac{1}{c} \frac{dn_C}{dt} = -\frac{1}{d} \frac{dn_D}{dt} \quad (\text{A.20})$$

As the concentration of the reactants increase, the rate of reaction will also increase due to an increase in the number of collisions between the reacting species, as explained by the *Molecular Collision Theory* [83].

In chemical kinetics, the rate equation is often used to associate the rate of the reaction with the concentration of each reactant such that it is proportional to the product of the reactant species concentrations raised to their respective stoichiometric coefficients, and is expressed as:

$$\frac{dn_A}{dt} = -k_f n_A^a n_B^b \quad (\text{A.21})$$

where k_f is defined as the forward reaction rate constant. This constant is affected by numerous factors; the most significant for the combustion process is its dependence on the reaction temperature. If the reaction is occurring at a higher temperature, there will be a greater number of collisions between molecules, which will deliver more energy into the system and increase the rate of the reaction. The number of successful collisions and energy that results are dependent on both the *Activation Energy*, E_A , which expresses the amount of energy that must be overcome in order for a reaction to occur, and the geometry of the collisions between the reacting molecules. The effects of temperature and activation energy, which are found experimentally, on the reaction rate constant, are described by the *Arrhenius Equation* shown in Equation A.22 [13].

$$k_f = \mathcal{A} \exp(-E_A/R_u T) \quad (\text{A.22})$$

where $\mathcal{A}$ is also experimentally determined and is known as the *pre-exponential factor*.

It is known that the chemical reaction rate can limit the rates of combustion in low pressure and low temperature situations due to reduced number of collisions between molecules. It can be accurately predicted using the above equations [84]. This rate is often used in determining the combustion efficiency for situations where the overall heat release is controlled by chemical kinetics using widely used methods, such as the *burning velocity* and *stirred reactor models*. These models are extensively described in reputable combustion texts, such as Lefebvre [1] and Turns [13]. In combustion applications, the rate of chemical reaction is quite high. It has been determined by numerous researchers like Chin and Lefebvre [28], and Odgers *et al.* [29], that the efficiency of gas turbine combustion is rarely affected or controlled by the rate of chemical reactions, but rather by the rate of droplet evaporation and burning. The rate of droplet evaporation and burning are very dependent on the rates of heat and mass transfer occurring during the combustion process.

A.3 Heat and Mass Transfer

Having reviewed the chemical processes involved and how the energy or heat of combustion is formed, it is now important to understand the fundamentals behind how this heat and mass are transferred during the combustion process. Most combustion models currently used examine the mixing process and subsequent heat and mass transfer, using the fundamental conservation equations of mass, energy, species, and momentum, completing the analysis with varying turbulence models to account for the heat transfer that is occurring during the process [85]. By gaining an understanding of these basic first principles of fluid mechanics and heat transfer, the complex phenomenon of combustion can be described on a more simplistic level.

Although flame properties are an important part of the combustion process, this topic is quite broad and given its complexity it will only be briefly discussed. Only the major concepts and current areas of research in combustion will be presented as they relate to heat and mass transfer in gaseous flows.

Before examining how heat and mass transfer affect droplet evaporation and burning, it is important to have a general understanding of the different modes that exist. Heat

transfer can occur either through conduction, convection or radiation. Conduction heat transfer is energy transport due to molecular motion and interaction and the rate of conductive heat transfer is governed by *Fourier's law* [86]. Fourier determined that the one-dimensional heat transfer per unit area or heat flux, q_x'' , is proportional to the temperature gradient, $T_2 - T_1$, and is given by:

$$q_x'' = -k \frac{dT}{dx} = -k \frac{T_2 - T_1}{\Delta x} \quad (\text{A.23})$$

where k is a transport property known as thermal conductivity and is a characteristic of the material.

Convective heat transfer is comprised of energy transport due to molecular diffusion and advection, which is the energy transferred by the bulk motion of the fluid. The term convection is usually used when referring to the cumulative transport and the term advection when referring to transport due to bulk fluid motion [13]. Newton described convective heat flux as being proportional to the difference between the surface and fluid temperatures, T_s and T_∞ , respectively [86]. The temperature difference usually occurs across a thin layer of fluid adjacent to the solid surface, otherwise known as the thermal boundary layer. The one-dimensional convection heat transfer rate, q_{conv}'' , is known as *Newton's Law of cooling* and is expressed as:

$$q_{conv}'' = h(T_s - T_\infty) \quad (\text{A.24})$$

where h is the convective heat transfer coefficient and depends on the fluid properties and fluid velocity.

Radiative heat transfer is energy transport due to emission of electromagnetic waves from a surface of liquid or gas. While the transfer of energy by conduction and convection requires a transfer medium, radiation does not [86]. The heat transfer by radiation originates from the thermal energy of matter bounded by the surface and the rate at which energy is released from the surface per unit area is known as the *emissive power*, E . The radiation heat transfer also depends on the material properties and surface

geometry and is represented by the *emissivity*, ε , of the material. The radiative heat flux emitted by a real surface is given by

$$E = \varepsilon \sigma T_s^4 \quad (\text{A.25})$$

There are modes of mass transfer that are analogous to convection and conduction heat transfer, and therefore when discussing specifically mass transfer, it is important understand the concept of diffusion. Diffusion is defined as the spontaneous net movement of particles from an area of high concentration to an area of low concentration in a given volume of fluid (either liquid or gas) down the concentration gradient [86]. The rate equation of mass diffusion is known as *Fick's Law*, and for the transfer of species A , in a mixture of A and B , is expressed as:

$$j_A = -\rho \mathcal{D}_{AB} \nabla m_A \quad (\text{A.26})$$

where j_A is the diffusive mass flux of species A , ρ is the mixture density, and ∇m_A is the gradient in species A mass fraction. The most important transport property in the above equations is the mass diffusivity, $\mathcal{D}_{AB}$, which is proportional to the relative velocity of the diffusing particles, which depends on the temperature, viscosity of the fluid and the size of the particles. Mass transport properties are particularly important when examining droplet evaporation.

The rates of heat and mass transfer are related to the *Laws of Conservation* and to briefly summarize, if a fuel and an oxidizer were to undergo a chemical reaction, as shown in Equation A.1, it would eventually lead to the formation of products and overall heat release. During chemical reactions, heat and individual species are created and/or destroyed, although momentum remains unaffected at the molecular level. Therefore, when examining heat and mass transfer during combustion processes, the momentum conservation equation will be the same as that for a chemically inert flow and is given by:

$$\bar{u} \frac{\partial}{\partial x} (\rho_g \bar{u}) + \bar{v} \frac{\partial}{\partial r} (\rho_g \bar{u}) = -\frac{\partial p}{\partial x} + \bar{\nabla} \cdot (\mu_g (\bar{\nabla} \bar{u})) \quad (\text{A.27})$$

The terms on the left-hand side of the equation represent the momentum flow by axial, $\bar{u}$, and radial, $\bar{v}$ convection per unit volume, respectively [87]. The first term on the right-hand side represents the pressure gradient, $\partial P / \partial x$, in the fluid and the last term represents the axial viscous force per unit volume, $\mu_g (\nabla \bar{u})$.

Using the same notation, the mass conservation or continuity equation is given by:

$$\frac{\partial}{\partial x}(\rho_g \bar{u}) + \frac{1}{r} \frac{\partial}{\partial r}(r \rho_g \bar{v}) = 0 \quad (\text{A.28})$$

which represents the net rate of mass flow out of the control volume.

The conservation of species and energy during the chemical reaction is extremely important since it is the destruction and subsequent creation of unit mass of one species that is related to the amount of heat release as well as changes in the concentrations of the rest of the species. Consequently, each species requires a species conservation equation.

Diffusion of species A , $\dot{m}_A'''$, is directly related to the concentration gradients and for axisymmetric geometry; the corresponding species conservation equation is given by:

$$\frac{1}{r} \frac{\partial}{\partial r}(r \rho_g \bar{v} Y_A) + \frac{1}{r} \frac{\partial}{\partial x}(\rho_g \bar{u} Y_A) - \frac{1}{r} \frac{\partial}{\partial r} \left[r \rho_g D_{AB} \frac{\partial Y_A}{\partial r} \right] = \dot{m}_A''' \quad (\text{A.29})$$

where Y_A is the mass fraction of species A . The first, second and third terms on the left-hand side of the equation represent the mass flow of species A per unit volume due to radial convection, axial convection and molecular diffusion in the radial direction. In Equation A.29, it was assumed that the axial diffusion was negligible in comparison to the radial diffusion so it was not included.

The relationship most commonly used for describing the energy conservation for chemical reactions is known as the *Shvab-Zeldovich energy equation*, which was named in recognition of the researcher who first developed it [87]. Before deriving this equation several assumptions must be made about the species, a key assumption being that the *Lewis number* is unity.

The *Lewis number*, Le , is related to both the *Prandtl number*, Pr , and the *Schmidt number*, Sc , which respectively provide a measure of relative effectiveness of momentum

and thermal energy, and momentum and mass transport, respectively, by diffusion in the velocity and thermal boundary layers. In simpler terms, the *Lewis number*, is the ratio of characteristic thermal diffusion rate of the mixture to the characteristic mass diffusion rate of the i^{th} species as shown in Equation A.30:

$$Le = \frac{Sc}{Pr} = \frac{\left(\frac{\nu}{D_{AB}}\right)}{\left(\frac{\nu}{\alpha}\right)} = \frac{\alpha}{D_{AB}} \quad (A.30)$$

where α is the fluid thermal diffusivity and ν is the kinematic viscosity. By making the assumption that $Le=1$, the energy equation is greatly simplified, which is sufficiently explained in Turns [13]. Due to its length and complexity, it will not be reproduced here. The *Shvab-Zeldovich energy equation* in axisymmetric form is given by:

$$\frac{1}{r} \frac{\partial}{\partial x} (r \rho_g \bar{u} \int c_p dT) + \frac{1}{r} \frac{\partial}{\partial r} (r \rho_g \bar{v} \int c_p dT) - \frac{1}{r} \frac{\partial}{\partial r} \left[r \rho_g D_{AB} \frac{\partial \int c_p dT}{\partial r} \right] = - \sum h_{f,i}^{\circ} \dot{m}_i''' \quad (A.31)$$

where $\bar{u}$ and $\bar{v}$ are the momentum axial and radial flow respectively, $\dot{m}_i'''$ is the mass diffusion of the species, ρ_g is the fluid density, c_p is the specific heat at constant pressure and $h_{f,i}^{\circ}$ is the enthalpy of formation. The first, second and third terms on the left-hand side of the equation represent the rate of sensible enthalpy transport per unit volume by axial convection, radial convection and radial diffusion, respectively. The term on the right-hand side of the equation represents the rate of enthalpy production by chemical reaction per unit volume.

The *Arrhenius number*, Ar , provides an indication of the sensitivity of the chemical reaction to temperature variations through the relationship shown in Equation A.22. The reaction rate and therefore the rate of consumption and production of species, and thus the energy or heat released during the chemical reaction are dependent upon this factor. Therefore, the rate of droplet evaporation and combustion are also dependant on this factor.

A.4 Droplet Evaporation and Burning

Before proceeding to a review of droplet evaporation, it is important to describe the relationship between diffusion and premixed flames. A premixed flame is a flame in which the oxidizer has been mixed with the fuel before it reaches the flame front. This creates a thin flame front since all of the reactants are readily available [88]. In a diffusion flame, the fuel and oxidizer are initially separated and they are transported through diffusion and convection to the flame region where mixing and reaction occur. The subsequent combustion heat and products generated are then transported away from the flame region.

The evaporation of fuel droplets in air involves heat and mass transfer where the heat of evaporation is transferred to the droplet surface (wall) by conduction and convection from the surroundings and vapour is transferred by convection and diffusion into the air stream [13]. This concept can easily be seen in Figure A.1.

Figure A.1: Schematic Showing Droplet Evaporation and Combustion Model (adapted from [88])

The rate of the droplet evaporation is dependent on pressure, temperature, velocity of the droplets relative to that of the air stream, and the transport properties of the fuel (temperature, volatility, initial diameter of droplets) [89]. If the droplet surface

temperature is known, a droplet evaporation model can be formed to describe the mass transfer that is occurring. However, most combustion environments have a high ambient temperature and the droplet surface temperature is usually unknown. Therefore, several assumptions must be made in order to analyze droplet evaporation. The most widely used model considers a single droplet evaporating in a hot gas. The following assumptions, originally derived by Godsave [90] and Spalding [33] and taken from Turns [13], are used in order to simplify calculations by eliminating the mass-transfer aspects:

- the droplet evaporates in quiescent surroundings;
- the evaporation process is steady state;
- the fuel is single component;
- binary diffusion is occurring;
- the droplet is spherical in shape;
- the thermophysical properties of the fuel are constant; and
- the droplet temperature is uniform.

It is assumed that the droplet surface temperature will be close to the boiling point of the fuel and hence, the evaporation rate is controlled by the heat transfer rate from the ambient air to the droplet surface rather than the mass transfer rate. The heat from the ambient environment supplies the energy necessary to vapourize the fuel, and the fuel vapour then diffuses from the droplet surface to the ambient surroundings. The droplet mass loss causes the droplet radius to decrease with time and the droplet surface region is always saturated with vapour. This process is known as diffusion-induced convection and therefore the rate-limiting processes are heat and mass diffusion, and convection [89].

Using these assumptions, the mass evaporation rate, $\dot{m}$, and the droplet radius history, $r_s(t)$, can be found by writing a gas-phase mass conservation equation, a gas-phase energy equation, a droplet gas-phase interface energy equation, and a droplet liquid mass conservation equation. It should be reiterated that all of the following equations apply to quiescent conditions. The gas-phase mass conservation equation is given by Equation A.32:

$$\dot{m} = \dot{m}_f = \rho \bar{v} 4\pi r^2 = \text{constant} \quad \text{and} \quad \frac{d(\rho \bar{v} r^2)}{dr} = 0 \quad (\text{A.32})$$

where r is a variable radius. A complex derivation of the gas-phase energy equation, which is adequately shown by Law [87], gives the temperature distribution in the gas phase as seen below:

$$T(r) = \frac{(T_\infty - T_{boil}) \exp(-Z\dot{m}/r) - T_\infty \exp(-Z\dot{m}/r_s) + T_{boil}}{1 - \exp(-Z\dot{m}/r_s)} \quad (\text{A.33})$$

where

$$Z \equiv c_{pg}/4\pi k_g \quad (\text{A.34})$$

and T_∞ , represents the surrounding air temperature, T_{boil} is the fluid boiling temperature, r_s is the droplet radius at the liquid-vapour interface, c_{pg} is the specific heat of the fluid at constant pressure, and k_g is the thermal conductivity of the fluid. This equation is necessary in order to determine the surface energy balance and using the value obtained for the heat transferred to the droplet surface, the interface (surface) energy balance can be obtained.

Heat, conducted to the interface from the ambient gas, is then used to vapourize the fuel, with no heat flowing into the droplet interior [87]. Using the surface energy balance and the derivative of Equation A.32, the droplet-gas-phase interface energy equation can be rearranged to give the evaporation rate, as shown in Equation A.35:

$$\dot{m} = \frac{4\pi r_s k_g}{c_{pg}} [B_q + 1] \quad (\text{A.35})$$

where

$$B_q = c_{pg}(T_\infty - T_{boil})/h_{fg} \quad (\text{A.36})$$

where h_{fg} is the heat of formation, and B_q is known as the Spalding mass transfer number and the subscript q indicates that it is based on the heat transfer considerations

alone. When considering only the mass transfer for the evaporation process, the mass evaporation rate ($\dot{m}$) becomes:

$$\dot{m} = 4\pi r_s \rho D_{AB} \ln[1 + B_Y] \quad (\text{A.37})$$

where B_Y is the mass transfer number based on the mass transfer considerations alone. Now that the evaporation rate has been estimated, the droplet lifetime can be determined. By equating the instantaneous evaporation rate:

$$\dot{m} = \frac{d}{dt} \left(\frac{\rho \pi D^3}{6} \right) \quad (\text{A.38})$$

where D is the droplet diameter, with Equation A.35 and rearranging, it can be shown that:

$$\frac{dD^2}{dt} = -\frac{8k_g}{\rho c_{pg}} \ln[B_q + 1] \quad (\text{A.39})$$

which shows that the time derivative of the square of the droplet diameter is constant and therefore, D^2 varies linearly in time at a rate K , which is given by:

$$K = \frac{8k_g}{\rho c_{pg}} \ln[B_q + 1] \quad (\text{A.40})$$

K is also known as the *evaporation constant* and is often expressed as λ_{ev} . By integrating Equation A.39 and using Equation A.40, a general relationship expressing the variation of the droplet diameter with time, known as the D^2 Law [13] can be found and is given by:

$$D^2(t) = D_o^2 - Kt \quad (\text{A.41})$$

This law can be used to determine the time it takes for a droplet of an initial diameter (D_o) to evaporate as shown in Figure A.2.

Figure A.2: The D^2 Law for Droplet Evaporation [13]

An extensive review of various scholarly research papers and current journal articles has revealed that heat and mass transfer relating to droplet evaporation and burning rates is widely studied using the above model with particular attention being paid to the effects of different types of fuel, droplet size and shape, and ambient conditions on the evaporation and burning rates. Some of the most evident conclusions have been that lighter and more volatile fuels evaporate faster than heavier, less volatile fuels as discussed by Rizk *et al.* [5]. Additionally, as the temperature of the surrounding conditions is increased, the evaporation time decreases. At lower temperatures, smaller droplet sizes are preferred since there is not sufficient heat in the flow to quickly heat-up large droplets to steady-state and commence evaporation, as found by Wang and Lefebvre [19]. However, the disadvantages of using heavier fuels can be counteracted by increasing the pressure at which the fuel is introduced into the combustor via the use of an atomizer, thus introducing smaller droplets and expediting the heat and mass transfer processes, as discussed by Rizk *et al.* [5].

In gas turbine combustors, several major assumptions for the above droplet evaporation and burning method do not hold true, the first being that the evaporation time is based upon single droplet experiments of a uniform size. It is known that fuel nozzles produce sprays with droplets in a range of sizes and distributions, as discussed by Wang and Lefebvre [19]. It has been shown through numerous experiments that the mean drop size has a much more significant effect on evaporation time than the size distribution of

the spray [14, 19, 31]. The next assumption that does not hold true is that the evaporation process is occurring at steady state in quiescent surroundings. There is significant mixing in the combustion chamber and the fuel droplets are introduced into an environment comprised of varying thermal and velocity gradients designed to promote better atomization, mixing and combustion. The droplets entering the combustor are subjected to these rapid changes in the environment and are no longer in steady state. The importance of atomization and chamber mixing on combustion will be examined in the following sections.

Extending the droplet evaporation and burning calculations to incorporate convection involves a simple alteration to Equations A.32 to A.40, replacing the surrounding temperature by the adiabatic flame temperature and the droplet surface temperature by the boiling point [89].

Equations A.32 to A.40 assume that there is no relative velocity between the droplet and the free stream. This is obviously not a true representation of what is actually occurring in a gas turbine combustor and forced convection must be incorporated into the droplet evaporation and burning models. This concept uses *film theory*, which is the replacement of the heat and mass transfer boundary conditions (BC) as described by Turns [13], which are typically:

$$BC\ 1 \quad T_r = T_\infty \quad as \quad r \rightarrow \infty \quad (A.42)$$

$$BC\ 2 \quad T_r = T_{boil} \quad at \quad r = r_s \quad (A.43)$$

with the same conditions moved radially to the so-called film radius, δ_M for species and δ_T for thermal energy [89]. By applying this theory, steeper concentration and temperature gradients are found and therefore increased mass and heat transfer rates at the droplet surface [89]. It can therefore be deduced that the presence of convection increases droplet burning rates and decreases burning times. The film radii are defined in terms of the *Nusselt number*, Nu (which is the non-dimensional temperature gradient at the droplet surface), for heat transfer and the *Sherwood number*, Sh (which is the concentration gradient at the surface). For the case with forced convection, it is assumed

that $Nu=Sh$ [13] and the average *Nusselt number* is found using the following empirical correlation:

$$\overline{Nu} = \bar{h} \frac{D}{k_a} = 2 + 0.6Re_d^{0.5}Pr^{0.33} = Nu(1 + 0.3Re_d^{0.5}Pr^{0.33}) \quad (A.44)$$

where the *Reynolds number*, $Re = \rho DV_{rel}/\mu_a$, is based on the droplet diameter and the relative velocity and the *Prandtl number*, $Pr = c_{pg}\mu_a/k_a$, is determined for the conditions of the surrounding air [89]. The subscript a is used to represent the properties of air. Using the equations from the quiescent conditions, the burning rate of the fuel and evaporation constant are now given by:

$$\dot{m}_F = \frac{2\pi D}{c_{pg}} (1 + 0.3Re_d^{0.5}Pr^{0.33}) \ln[B_q + 1] \quad (A.45)$$

$$\lambda_{ev} = \frac{8}{\rho c_{pg}} (1 + 0.3Re_d^{0.5}Pr^{0.33}) \ln[B_q + 1] \quad (A.46)$$

Using the Equations A.42 to A.46, evaporation and burning models can be developed to explore the influence of droplet size and ambient conditions on droplet lifetime and subsequently, the combustion rates of numerous systems including gas turbine combustors.

Appendix B

Schematics and Instrumentation Data

Table B.1: Combustion Chamber Sector Rig Components

System	Component	Manufacturer	Description	Model Number
Fuel	Fuel Pump	Rocky Mountain Industries	Max Output of 9.5 L/min	RM1P2003429
	Electric Motor	WEG	1 Hp	XCOO504F
Air	Electric Motor	WEG	40 Hp	IP23
	Variable Speed Drive	ABB		ACS800-U1-0050-7
	Centrifugal, 2-stage	National Turbine Corp	Cast iron	NT8802-2-0-0-0AD
	Venturi Flow Meter	Lambda Square Inc.	Fibreglass reinforced plastic	8 in Venturi 2300
Instrumentation (Differential) (Static) (Fuel)	Turbine Flow Meter	FTI Flow Technology	0-10 V output	FT4-8AEXS-LEA-5
	Pressure Transducer	Omega	0-25 inH ₂ O, 1-5 V output	PX 653
	Pressure Transducer	Omega	0-300 psig, 4-20mAmp DC	PTX 1240
	Pressure Transducer	Omega	0-5 psig, 0-5 V output	PX 4202
	Thermocouple	Omega	Type K, exposed junction	TJ36-CAXL-18E-6-CC-XISB
	Data Acquisition System	Hewlett Packard	Data acquisition/control unit	3852A

Appendix C

Sample Calculations

C.1 Mass Flow Parameter Matching Calculation

To ensure the mixing processes between the combustion chamber sector rig and the engine were matched, a Mass Flow Parameter at the entrance of the combustion chamber was determined for the engine operating condition of 463 km/hr (250 knots) at 7620 m (25 000 ft). Taylor [78] created a model to determine the air mass flow rate at the entrance of a T56-A-15 engine at various operating conditions. At the selected operating condition, he found that for a single combustion chamber of a T56-A-15 engine operating at 463 km/hr (250 knots) at 7620 m (25 000 ft), the ambient temperature and pressure would be 454 K and 357 kPa, and the air mass flow rate would be 1.23 kg/s. Using these values, the Mass Flow Parameter for the engine inlet is:

$$\left(\frac{\dot{m}_a \sqrt{T}}{P}\right)_{engine} = \frac{(1.23 \text{ kg/s}) \sqrt{454 \text{ K}}}{357 \text{ 000 Pa}} = 7.36 \times 10^5 \frac{\text{kg} \cdot \text{K}^{0.5}}{\text{s} \cdot \text{Pa}} \quad (\text{C.1})$$

Measured values from fuel nozzle 47M at a randomly selected time of 1096 s, taken from Tables D.10 and D.11 are:

$$\dot{m}_a = 0.981 \text{ kg/s}; T_{inlet} = 309 \text{ K}; P_{ambient} = 100 \text{ 500 Pa}; P_{static} = 17606 \text{ Pa} \quad (\text{C.2})$$

The Mass Flow Parameter at the entrance of test combustion chamber in the test rig was calculated to be:

$$\left(\frac{\dot{m}_a \sqrt{T}}{P}\right)_{rig} = \frac{\left(\frac{0.98 \text{ kg/s}}{2}\right) \sqrt{309 \text{ K}}}{(100 \text{ 500 Pa} + 17606 \text{ Pa})} = 7.31 \times 10^5 \frac{\text{kg} \cdot \text{K}^{0.5}}{\text{s} \cdot \text{Pa}} \quad (\text{C.3})$$

The air mass flow rate was divided by two in Equation C.3 since the air entering the test section is split between the test combustion chambers and the two half dummy combustion chambers. Comparing the values in Equations C.1 and C.3, the Mass Flow Parameters are nearly identical, with less than a 1% difference.

C.2 Sauter Mean Diameter (D_{32}) Calculations

There exist a vast number of equations derived from numerous experiments to calculate the droplet D_{32} from a pressure swirl atomizer. Lefebvre's [1] correlation has been used in this research since it is dimensionally correct for a dual orifice pressure swirl atomizer and is given as:

$$D_{32} = 2.25\sigma_f^{0.25}\nu_f^{0.25}\dot{m}_f^{0.25}\Delta P_f^{-0.5}\rho_a^{-0.25} \quad (C.4)$$

where σ and ν are the surface tension and viscosity of the fuel, respectively, $\dot{m}_f$ is the fuel mass flow rate, ΔP_f is the fuel injection pressure, and ρ_a is the inlet air density. The fuel properties used in these calculations were taken from [68] for Jet A-1 at room temperature. The rig fuel mass flow rate was taken from the measured values for fuel nozzle 47M, as contained in Table D.11. The engine fuel mass flow rate was taken from [87]. Fuel pressures for both the rig and the engine were taken from the fuel mass flow rate curve for fuel nozzle 47M.

According to the T56-A-15 Inspection Manual [7], the fuel nozzle secondary port opens at a fuel delivery pressure of 448 kPa (65 psi). At the selected operating condition, the engine fuel flow rate to a single can was determined to be 27 g/s from [77] equating to a fuel delivery pressure of 965 kPa (140 psi), making the primary and secondary port pressures 965 kPa (140 psi) and 517 kPa (75 psi), respectively. These values are identical to those determined by Kotzer [62] for the same operating condition. At these fuel delivery pressures, the combined D_{32} for the droplets being produced by the primary and secondary ports was determined to be 42 μm . The same process used to determine the engine combined D_{32} was used to determine the rig combined D_{32} . Therefore only the process for determining the combined D_{32} of the rig will be shown.

In order to determine the air density entering the combustion chamber for the test rig, the ideal gas law and the measured values for fuel nozzle 47M found in Table D.11 were used. Inserting these values into Equation C.4 gives the D_{32} for the rig primary and secondary ports.

$$D_{32}^{primary} = 2.25 \left(0.026 \frac{kg}{m^3}\right)^{0.25} \left(0.0129 \frac{m^2}{s}\right)^{0.25} \left(0.0068 \frac{kg}{s}\right)^{0.25} (517\,000\,Pa)^{-0.5} \left(1.3 \frac{kg}{m^3}\right)^{-0.25} \quad (C.5)$$

$$= 110\,\mu m \quad (C.6)$$

$$D_{32}^{primary} = 2.25 \left(0.026 \frac{kg}{m^3}\right)^{0.25} \left(0.0129 \frac{m^2}{s}\right)^{0.25} \left(0.0027 \frac{kg}{s}\right)^{0.25} (138\,000\,Pa)^{-0.5} \left(1.3 \frac{kg}{m^3}\right)^{-0.25} \quad (C.7)$$

$$= 172\,\mu m \quad (C.8)$$

To calculate the combined D_{32} for the test rig, the primary and secondary D_{32} values must be amalgamated. This involves taking into consideration the difference in fuel flow rates between the primary and secondary ports. The first step is determining the volumetric flow rates ($\dot{V}_f$):

$$\dot{V}_f^{primary} = \frac{\dot{m}_f}{\rho_f} = \frac{0.0068\,kg/s}{804\,kg/m^3} = 8.46 \times 10^{-6} \frac{m^3}{s} \quad (C.9)$$

$$\dot{V}_f^{secondary} = \frac{0.0027\,kg/s}{804\,kg/m^3} = 3.36 \times 10^{-6} \frac{m^3}{s} \quad (C.10)$$

where ρ_f is the fuel density and The next step is to determine the surface area (A_s) and volume of a spherical droplet (V_s) from the primary and secondary ports:

$$A_s^{primary} = \pi D_s^2 = \pi (110\,\mu m)^2 = 3.7 \times 10^{-8} m^2 \quad (C.11)$$

$$A_s^{secondary} = \pi (172\,\mu m)^2 = 9.6 \times 10^{-8} m^2 \quad (C.12)$$

$$V_s^{primary} = \frac{\pi D_s^3}{6} = \frac{\pi (110\,\mu m)^3}{6} = 7.0 \times 10^{-13} m^3 \quad (C.13)$$

$$V_s^{secondary} = \frac{\pi (172\,\mu m)^3}{6} = 2.8 \times 10^{-12} m^3 \quad (C.14)$$

and the rate of droplet formation ($\dot{d}_s$):

$$\dot{d}_s^{primary} = \frac{\dot{V}_f^{primary}}{V_s^{primary}} = \frac{8.46 \times 10^{-6} \frac{m^3}{s}}{7.0 \times 10^{-13} m^3} = 1.22 \times 10^7 \frac{droplets}{s} \quad (C.15)$$

$$\dot{d}_s^{secondary} = \frac{\dot{V}_f^{secondary}}{V_s^{secondary}} = \frac{3.36 \times 10^{-6} \frac{m^3}{s}}{2.8 \times 10^{-12} m^3} = 1.3 \times 10^6 \frac{droplets}{s} \quad (C.16)$$

producing a combined total droplet area and volume per second of:

$$\begin{aligned} \dot{A}_{total} = & (3.7 \times 10^{-8} m^2) \left(1.22 \times 10^7 \frac{droplets}{s} \right) + (9.6 \times 10^{-8} m^2) \left(1.3 \times 10^6 \frac{droplets}{s} \right) \end{aligned} \quad (C.17)$$

$$= 0.6 \frac{m^2}{s} \quad (C.18)$$

$$\begin{aligned} \dot{V}_{total} = & (7.0 \times 10^{-13} m^3) \left(1.22 \times 10^7 \frac{droplets}{s} \right) + (2.8 \times 10^{-12} m^3) \left(1.3 \times 10^6 \frac{droplets}{s} \right) \end{aligned} \quad (C.19)$$

$$= 1.17 \times 10^{-5} \frac{m^3}{s} \quad (C.20)$$

The combined D_{32} is the ratio of the droplet formation rates by volume to surface area. Therefore, the experimentation rig D_{32} is:

$$\frac{\dot{V}_{total}}{\dot{A}_{total}} = \frac{\pi D_{32}^{combined^3}}{\pi D_{32}^{combined^2}} \quad (C.21)$$

$$D_{32}^{combined} = 6 \left(\frac{\dot{V}_{total}}{\dot{A}_{total}} \right) = 6 \left(\frac{1.17 \times 10^{-5} \frac{m^3}{s}}{0.6 \frac{m^2}{s}} \right) = 110 \mu m \quad (C.22)$$

C.3 Calibration Curves Used in *LabVIEW*TM

Calibration curves for all pressure transducers, the turbine flow meter and the sweeper arm potentiometer were created to translate the instrument outputs into the display values in *LabVIEW*TM. The voltage outputs for all pressure transducers were converted to pressure readings using the calibration data provided by the manufacturers. The calibration curves for the static pressure transducer, differential pressure transducer, and the fuel pressure transducer are shown in Figures C.1, C.2, and C.3, where the equations shown were calculated using *Excel*TM.

The turbine flow meter voltage output was converted to a volumetric flow rate, as seen in Figure C.4, using the manufacturer's calibration data. Using a fuel density of 804 kg/m^3 [68], the volumetric flow rate was then converted to a fuel mass flow rate, as shown in Figure C.5.

The sweeper arm potentiometer calibration curve shown in Figure C.6 was published in Kotzer [62] during the commissioning of the combustion chamber sector rig. A digital protractor was attached to the sweeper arm and the output voltage and corresponding angular positions were recorded for numerous radial positions.

Figure C.1: Static Pressure Transducer Calibration Curve

Figure C.2: Differential Pressure Transducer Calibration Curve

Figure C.3: Fuel Pressure Transducer Calibration Curve

Figure C.4: Turbine Flow Meter Calibration Curve – Volumetric Flow Rate

Figure C.5: Turbine Flow Meter Calibration Curve – Fuel Mass Flow Rate

Figure C.6: Sweeper Arm Potentiometer Calibration Curve [72]

C.4 Fuel Mass Flow Rate Calculation

The selected operating parameters of an air mass flow rate of 0.49 kg/s and an equivalence ratio of 0.28 were used to determine fuel mass flow rate. As per the findings of Blazowski [52], the carbon-to-hydrogen ratio of Jet A-1 was taken as $\text{CH}_{1.94}$. Neglecting dissociation of the products, the stoichiometric equation is:

The fuel and air mass for this mixture are determined as:

$$m_f = (1 \text{ mol} \times 12.011 \text{ g/mol}) + (1.94 \text{ mol} \times 1.008 \text{ g/mol}) = 13.97 \text{ g}_{\text{fuel}} \quad (\text{C.24})$$

$$m_a = 2(1.49 \text{ mol} \times 15.99 \text{ g/mol}) + 2 \left(5.602 \text{ mol} \times 14.00 \frac{\text{g}}{\text{mol}} \right) = 204.5 \text{ g}_a \quad (\text{C.25})$$

Using these values, the stoichiometric fuel-air ratio is then calculated:

$$f_{st} = \frac{13.97 \text{ } g_{fuel}}{204.5 \text{ } g_a} = 0.0683 \frac{g_{fuel}}{g_a} \quad (C.26)$$

Using Equation C.26 and the measured values from fuel nozzle 47M, taken from Tables D.11, the fuel-air ratio is determined as:

$$f = \phi f_{st} = (0.282)0.0683 \frac{g_{fuel}}{g_a} = 0.0193 \quad (C.27)$$

Using the selected air mass flow rate and the value determined in Equation C.27, the fuel mass rate is calculated as:

$$f = \frac{\dot{m}_f}{\dot{m}_a} \quad (C.28)$$

$$\dot{m}_f = (0.0193) \left(\frac{0.984}{2} \frac{kg}{s} \right) \left(\frac{1000 \text{ } g}{1 \text{ } kg} \right) = 9.5 \frac{g}{s} \quad (C.29)$$

C.5 Measurement Uncertainty

Estimating uncertainty is an essential requirement for all measurements because it gives an indication of the accuracy of the measured value when compared to the *true value*. The true value ideally matches the magnitude of the physical property of interest and it is the determination of this value that is the driving force for conducting the measurement [91]. Uncertainties in the measured values can be the result of errors that arise during instrument calibration, data acquisition, data reduction, or errors associated with the measuring instruments themselves. There are two main types of errors that can cause a measured value to be inaccurate: *bias* errors and *precision* errors.

Bias errors are systematic errors meaning that they are an estimate of the difference between the mean of repetitive measurements and the true value of the property being measured. Precision errors are considered random errors and are an estimate of the standard deviation of the measured values obtained in repetitive tests under the same conditions [91]. Both of these errors contribute to the uncertainty of a measurement, u , which is determined using Equation C.30:

$$u = \sqrt{b^2 + p^2} \quad (C.30)$$

where b and p are, respectively, the bias and precision limits of the measurement. Uncertainty calculations for the independently measured properties and derived properties are provided in the subsequent sections. The Type-R thermocouples were calibrated every five test runs. Prior to carrying out each test run, all instruments were zeroed. This reduced the possibility of any factors, like data acquisition errors or instrument errors, systematically affecting the measurement of the test parameters across the test run. As a result, bias errors were deemed negligible and were not examined during the uncertainty analysis of the test parameters.

The uncertainty calculations performed in the subsequent sections were determined using Taylor series expansion. In general terms, it represents a function as an infinite sum of the terms calculated from the value of its derivatives [91]. It provides a means of expressing the deviation of an output from its nominal value in terms of deviations of its inputs from their nominal values. It contains higher order powers of deviations and higher order derivatives of the function with respect to each input. If the deviations are relatively small, the higher powers become negligible. For this analysis, it can be seen that the uncertainties determined using only the first order approximation are quite small. Additionally, the majority of the functions examined varied linearly and in those cases, the higher order terms would be significantly smaller and have a negligible effect on the overall uncertainty calculation. Therefore, these terms were not included in the uncertainty analysis provided in the following sections.

C.6 Fuel Mass Flow Rate Uncertainty

The fuel mass flow rate measurements were carried out to determine the approximate fuel delivery pressure required for each nozzle to deliver a fuel mass flow rate of 9.5 g/s. The uncertainty of the time measurement was ± 0.25 s to account for any error associated with human response time and the uncertainty of the digital scale was ± 5 g. The fuel mass flow was calculated by:

$$\dot{m}_f \approx \frac{\Delta m_f}{\Delta t} \quad (\text{C.31})$$

The uncertainty of the fuel mass flow rate is given by:

$$u_{\dot{m}_f} = \sqrt{\left[\left(\frac{\partial \dot{m}_f}{\partial m_f}\right)(u_{m_f})\right]^2 + \left[\left(\frac{\partial \dot{m}_f}{\partial t}\right)(u_t)\right]^2} \quad (\text{C.32})$$

$$u_{\dot{m}_f} = \sqrt{\left(\frac{u_{m_f}}{\Delta t}\right)^2 + \left(\frac{-m_f u_t}{\Delta t^2}\right)^2} \quad (\text{C.33})$$

At 690 kPa (100 psi), 1115g of fuel was collected in 60 s for fuel nozzle 47M. Inserting these values into Equation C.32 gives the uncertainty for this reading:

$$u_{\dot{m}_f} = \sqrt{\left(\frac{5g}{60s}\right)^2 + \left(\frac{-(1115g)(0.25s)}{(60s)^2}\right)^2} = 0.1 \text{ g/s} \quad (\text{C.34})$$

C.7 Air Mass Flow Rate Calculation

To determine the air mass flow rate, the measured inlet static pressure and temperature were first used to calculate the air density continuously at the sampling frequency. The differential pressure measured using the venturi flow meter was then used to determine the air mass flow rate. A derivative of Bernoulli's equation was used to calculate the air mass flow rate as seen in Equation C.35. A full derivation of this equation can be found in White [92]. In this equation, the terms r_1 and r_2 represent the venturi inlet and throat radii, respectively:

$$\dot{m}_a = C_d A_2 \sqrt{\frac{2\rho_a \Delta P_{venturi}}{1 - \left(\frac{r_2}{r_1}\right)^4}} \quad (\text{C.35})$$

where C_d is the discharge coefficient, A_2 is the throat area, and $\Delta P_{venturi}$ is the measured differential air pressure. Values obtained from the manufacturer for the venturi flow meter [93] were as follows:

$$C_d = 0.995; \quad r_1 = 0.10135 \text{ m}; \quad r_2 = 0.06058 \text{ m} \quad (\text{C.36})$$

Measured values from fuel nozzle 47M at 545 s, taken from Tables D.10 and D.11 were:

$$\Delta P_{venturi} = 2390 \text{ Pa}; P_{static} = 17\,575 \text{ Pa}; P_{atm} = 100\,500 \text{ Pa}; T_{inlet} = 308 \text{ K} \quad (C.37)$$

The air density was calculated as:

$$\rho_a = \frac{P_a}{RT_{inlet}} = \frac{(P_{atm} + P_{static})}{RT_{inlet}} = \frac{(100\,500 \text{ Pa} + 17\,575 \text{ Pa})}{(287 \frac{\text{J}}{\text{kgK}} 309 \text{ K})} = 1.33 \frac{\text{kg}}{\text{m}^3} \quad (C.38)$$

Substituting the values from Equations C.36, C.37, and C.38 into Equation C.35, the air mass flow rate was calculated.

$$\dot{m}_a = (0.995)(\pi(0.06058 \text{ m})^2) \sqrt{\frac{2(1.33 \frac{\text{kg}}{\text{m}^3})2390 \text{ Pa}}{1 - \left(\frac{0.06058 \text{ m}}{0.10135 \text{ m}}\right)^4}} = 0.979 \text{ kg/s} \quad (C.39)$$

The value calculated in *LabVIEW*TM and shown in Table D.11 was 0.979 kg/s.

C.8 Air Mass Flow Rate Uncertainty

Using Equation C.35, the air mass flow rate uncertainty can be determined by separating the constants as follows:

$$\dot{m}_a = C_d A_2 \sqrt{\frac{2\rho_a \Delta P_{venturi}}{1 - \left(\frac{r_2}{r_1}\right)^4}} = \frac{C_d A_2 \sqrt{2}}{\sqrt{1 - \left(\frac{r_2}{r_1}\right)^4}} \sqrt{\left(\frac{P_{static}}{RT_{inlet}}\right) \Delta P_{venturi}} = \frac{C_d A_2 \sqrt{2}}{\sqrt{R \left[1 - \left(\frac{r_2}{r_1}\right)^4\right]}} \sqrt{\frac{P_{static} \Delta P_{venturi}}{T_{inlet}}} \quad (C.40)$$

The first term is a constant and can be defined as $\hat{C}_d$ and is calculated as follows:

$$\hat{C}_d = \frac{C_d A_2 \sqrt{2}}{\sqrt{R \left[1 - \left(\frac{r_2}{r_1}\right)^4\right]}} = \frac{(0.995)(\pi(0.06058 \text{ m})^2) \sqrt{2}}{\sqrt{287 \frac{\text{J}}{\text{kgK}} \left[1 - \left(\frac{0.06058 \text{ m}}{0.10135 \text{ m}}\right)^4\right]}} = 1.0248 \times 10^{-3} \text{ msK}^{0.5} \quad (C.41)$$

The air mass flow rate can then be expressed simply as:

$$\dot{m}_a = \hat{C}_d \sqrt{\frac{P_{static} \Delta P_{venturi}}{T_{inlet}}} \quad (C.42)$$

Equation C.42 can now be used to determine the uncertainty of the air mass flow rate.

$$u_{\dot{m}_a} = \sqrt{\left[\frac{\partial \dot{m}_a}{\partial P_{static}} u_{P_{static}}\right]^2 + \left[\frac{\partial \dot{m}_a}{\partial \Delta P_{venturi}} u_{\Delta P_{venturi}}\right]^2 + \left[-\frac{\partial \dot{m}_a}{\partial T_{inlet}} u_{T_{inlet}}\right]^2} \quad (C.43)$$

It is recognized that a covariance exists between the pressure measurements and the inlet temperature and this was not incorporated into Equation C.43. According to the Ideal Gas Law, pressure varies linearly with temperature, so the second derivatives of this relationship would be zero. Additionally, the inlet air temperature did not change during the individual experiments and there were minimal deviations between the different test runs. Therefore, the covariance between the inlet air temperature and pressure was considered to have negligible effect on the mass flow rate uncertainty calculations and was not included in Equation C.43.

Measured values from fuel nozzle 47M, taken from Tables D.10 and D.11 were used to determine the uncertainties for the three portions of Equation C.43. The uncertainties of the differential and static pressure transducers were taken as 0.25% of their respective full scale readings of 6 221 Pa and 34 474 Pa as given in the manufacturer's calibration data. The thermocouple uncertainty measurement was 2 K.

$$\frac{\partial \dot{m}_a}{\partial P_{static}} u_{P_{static}} = \frac{\hat{C}_d u_{P_{static}}}{2} \sqrt{\frac{\Delta P_{venturi}}{P_{static} T_{inlet}}} = \frac{(1.0248 \times 10^{-3} \text{ msK}^{0.5})(86.2 \text{ Pa})}{2} \sqrt{\frac{2390 \text{ Pa}}{(17575 \text{ Pa})(309 \text{ K})}} \quad (C.44)$$

$$= 9.27 \times 10^{-4} \text{ kg/s} \quad (C.45)$$

$$\frac{\partial \dot{m}_a}{\partial \Delta P_{venturi}} u_{\Delta P_{venturi}} = \frac{\hat{C}_d u_{\Delta P_{venturi}}}{2} \sqrt{\frac{P_{static}}{\Delta P_{venturi} T_{inlet}}} \quad (C.46)$$

$$= \frac{(1.0248 \times 10^{-3} \text{ msK}^{0.5})(15.6 \text{ Pa})}{2} \sqrt{\frac{17575 \text{ Pa}}{(2390 \text{ Pa})(309 \text{ K})}} \quad (C.47)$$

$$= 1.24 \times 10^{-3} \text{ kg/s} \quad (C.48)$$

$$\frac{\partial \dot{m}_a}{\partial T_{inlet}} u_{T_{inlet}} = -\frac{\hat{C}_d u_{T_{inlet}}}{2} \sqrt{\frac{P_{static} \Delta P_{venturi}}{T_{inlet}^3}} \quad (C.49)$$

$$= \frac{(1.0248 \times 10^{-3} \text{ msK}^{0.5})(2 \text{ K})}{2} \sqrt{\frac{(2390 \text{ Pa})(17575 \text{ Pa})}{(309 \text{ K})^3}} \quad (\text{C.50})$$

$$= -1.22 \times 10^{-3} \text{ kg/s} \quad (\text{C.51})$$

Inserting these uncertainties into Equation C.43 gives the measurement uncertainty for the air mass flow rate.

$$u_{\dot{m}_a} = \sqrt{\left[9.27 \times \frac{10^{-4} \text{ kg}}{\text{s}}\right]^2 + \left[1.24 \times \frac{10^{-3} \text{ kg}}{\text{s}}\right]^2 + \left[-1.22 \times \frac{10^{-3} \text{ kg}}{\text{s}}\right]^2} \quad (\text{C.52})$$

$$= 0.002 \text{ kg/s} \quad (\text{C.53})$$

C.9 Equivalence Ratio Calculation

The equation used in *LabVIEW*TM to determine the equivalence ratio is as follows:

$$\phi = \left(\frac{\dot{m}_f}{\left(\frac{\dot{m}_a}{2}\right)}\right) \left(\frac{1}{f_{st}}\right) \quad (\text{C.54})$$

Only half of the airflow enters the test combustion chamber, therefore the air mass flow rate is divided by 2. Using the experimental values for fuel nozzle 47M, taken from Tables D.10 and D.11, the stoichiometric fuel-air ratio from Equation C.27, the equivalence ratio was:

$$\phi = \left(\frac{0.00947 \text{ kg/s}}{\left(\frac{0.979 \text{ kg/s}}{2}\right)}\right) \left(\frac{1}{0.0683}\right) = 0.283 \quad (\text{C.55})$$

The value recorded in Table D.12 is 0.283.

C.10 Equivalence Ratio Uncertainty

The uncertainty for the equivalence ratio is given as:

$$u_\phi = \sqrt{\left(\frac{\partial \phi}{\partial \dot{m}_f} u_{\dot{m}_f}\right)^2 + \left(\frac{\partial \phi}{\partial \dot{m}_a} u_{\dot{m}_a}\right)^2} \quad (\text{C.56})$$

Using the experimental values for fuel nozzle 47M, taken from Tables D.10 and D.5, along with uncertainty for the air mass flow rate contained in Equation C.53 and a fuel mass flow rate uncertainty of 0.5% of the measured value, as stated by the manufacturer, the uncertainty of the equivalence ratio was calculated as:

$$u_{\phi} = \sqrt{\left(\frac{2u_{\dot{m}_f}}{\dot{m}_{afst}}\right)^2 + \left(-\frac{2\dot{m}_f u_{\dot{m}_a}}{\dot{m}_a^2 f_{st}}\right)^2} = \sqrt{\left(\frac{2\left(4.74 \times 10^{-5} \frac{kg}{s}\right)}{\left(0.979 \frac{kg}{s}\right)(0.0683)}\right)^2 + \left(-\frac{2\left(0.00947 \frac{kg}{s}\right)\left(0.002 \frac{kg}{s}\right)}{\left(0.979 \frac{kg}{s}\right)^2 (0.0683)}\right)^2} \quad (C.57)$$

$$= 0.002 \quad (C.58)$$

Appendix D

Data

Table D.1: Unknown Nozzle Condition Images

Nozzle	Unknown	Notes
AGM291		TSI – 341.1 hours
AGL824		Cleaned nozzle used in Guy's [90] research
VN1AMM1301		TSI – 136.6 hours
VN1AMT1839		TSI – 136.6 hours
2192		TSI – 380.7 hours
AGH3055		TSI – 1003.3 hours

Table D.1: (cont): Unknown Nozzle Condition Images

Nozzle	Unknown	Notes
AGH4084		TSI - 1003.3 hours
4298		TSI - 510.3 hours
AG4306		Dirty nozzle used in Guy's [90] research
AG4309		Dirty nozzle used in Guy's [90] research
AG56522		Cleaned nozzle used in Guy's [90] research
AG7553		Refurbished nozzle used in Guy's [90] research

Table D.1: (cont): Unknown Nozzle Condition Images

Nozzle	Unknown	Notes
123BQ		TSI – 410.4 hours
14FN		TSI – 943.8 hours
31AG		TSI – 800.2 hours
AG347G		TSI – 942.8 hours
48BM		TSI – 824.4 hours
83BJ		Refurbished nozzle used in Guy's [90] research

Table D.1: (cont): Unknown Nozzle Condition Images

Nozzle	Unknown	Notes
94EX		TSI – 410.4 hours
AG3599		TSI – 943.8 hours
AGK944		TSI – 510.3 hours
AM8FM		TSI – 824.4 hours
VN1AMM0335		TSI – 932.2 hours

Table D.2: Serviceable Nozzle Condition Images

Nozzle	Serviceable	Nozzle	Serviceable
AGM291		31AG
VN1AMM1301		47M
VN1AMT1839		AG3599
2192		AGK944
AGH4084		VN1AMM0335
14FN

Table D.3: Unserviceable Nozzle Condition Images

Nozzle	Unserviceable
AGH3055
4298
123BQ
AG347G
48BM
94EX
AM8FM

Table D.4: Spray Characteristic Data at 345 kPa

Nozzle	Category	Spray Diameter [cm]	Spray Cone Angle [°]	Max Out of Round	Max Streakiness
AGM291	Unknown	6.19	80.7	0.089	0.327
AGL824	Unknown	6.63	84.5	0.06	0.085
VN1AMM1301	Unknown	6.80	90.6	0.078	0.184
VN1AMT1839	Unknown	6.76	90.2	0.081	0.19
2192	Unknown	5.83	77.3	0.086	0.183
AGH3055	Unknown	6.06	79.4	0.116	0.199
AGH4084	Unknown	6.04	82.9	0.09	0.278
4298	Unknown	6.03	79.2	0.136	0.31
AG4306	Unknown	5.59	83.7	0.115	0.249
AG4309	Unknown	5.71	76.1	0.226	0.247
AGF6522	Unknown	5.73	76.3	0.154	0.274
AG7553	Unknown	6.89	86.8	0.071	0.066
123BQ	Unknown	6.41	82.7	0.063	0.076
14FN	Unknown	5.91	81.4	0.121	0.126
31AG	Unknown	5.62	78.3	0.184	0.401
AG347G	Unknown	5.92	78.1	0.08	0.238
48BM	Unknown	5.81	80.4	0.134	0.22
83BJ	Unknown	6.58	84.1	0.087	0.062
94EX	Unknown	6.06	83.0	0.073	0.218
AG3599	Unknown	6.36	82.2	0.086	0.081
AGK944	Unknown	6.02	82.6	0.112	0.11
AM8FM	Unknown	5.72	79.4	0.161	0.303
VN1AMM0335	Unknown	7.37	90.6	0.081	0.151
AGM291	Serviceable	5.54	83.2	0.062	0.098
VN1AMM1301	Serviceable	6.55	94.3	0.109	0.063
VN1AMT1839	Serviceable	6.20	90.7	0.077	0.076
2192	Serviceable	5.43	81.9	0.065	0.102
AGH4084	Serviceable	5.60	84	0.09	0.09
14FN	Serviceable	6.06	89.1	0.1	0.064
31AG	Serviceable	4.68	78.4	0.096	0.088
47M	Serviceable	6.63	77.1	0.062	0.084
AG3599	Serviceable	6.48	93.7	0.06	0.074
AGK944	Serviceable	6.29	91.6	0.081	0.068
VN1AMM0335	Serviceable	6.36	92.2	0.057	0.068
AGH3055	Unserviceable	5.58	83.7	0.057	0.109
4298	Unserviceable	6.47	93.6	0.109	0.195
123BQ	Unserviceable	5.73	85.5	0.079	0.113
AG347G	Unserviceable	5.57	83.6	0.082	0.119
48BM	Unserviceable	5.63	84.2	0.078	0.084
94EX	Unserviceable	5.77	85.9	0.088	0.079
AM8FM	Unserviceable	6.03	89	0.136	0.31

Table D.5: Spray Characteristic Data at 517 kPa

Nozzle	Category	Spray Diameter [cm]	Spray Cone Angle [°]	Max Out of Round	Max Streakiness
AGM291	Unknown	6.10	79.8	0.092	0.127
AGL824	Unknown	6.24	81.1	0.077	0.087
VN1AMM1301	Unknown	6.37	82.3	0.069	0.123
VN1AMT1839	Unknown	6.44	83.1	0.078	0.193
2192	Unknown	5.69	75.9	0.135	0.104
AGH3055	Unknown	5.95	78.4	0.152	0.172
AGH4084	Unknown	5.81	76.7	0.113	0.144
4298	Unknown	5.74	76.4	0.146	0.178
AG4306	Unknown	5.92	76.2	0.171	0.196
AG4309	Unknown	5.72	77.2	0.117	0.14
AGF6522	Unknown	5.56	74.7	0.12	0.145
AG7553	Unknown	6.74	85.5	0.051	0.086
123BQ	Unknown	6.28	81.5	0.058	0.08
14FN	Unknown	6.01	78.7	0.083	0.104
31AG	Unknown	5.57	74.2	0.145	0.253
AG347G	Unknown	5.60	75.1	0.091	0.085
48BM	Unknown	5.70	75.5	0.145	0.213
83BJ	Unknown	6.33	81.9	0.062	0.081
94EX	Unknown	5.92	77.8	0.124	0.223
AG3599	Unknown	5.95	78.4	0.081	0.087
AGK944	Unknown	6.10	79.6	0.08	0.161
AM8FM	Unknown	5.67	75.3	0.172	0.187
VN1AMM0335	Unknown	6.89	86.7	0.067	0.15
AGM291	Serviceable	5.49	77.7	0.071	0.102
VN1AMM1301	Serviceable	6.29	84.0	0.086	0.074
VN1AMT1839	Serviceable	6.05	81.6	0.074	0.119
2192	Serviceable	5.18	79.3	0.081	0.1
AGH4084	Serviceable	5.37	78.5	0.082	0.09
14FN	Serviceable	6.02	81.4	0.091	0.1
31AG	Serviceable	4.49	76.2	0.121	0.123
47M	Serviceable	6.32	79.6	0.077	0.072
AG3599	Serviceable	6.26	83.7	0.092	0.091
AGK944	Serviceable	6.22	83.2	0.108	0.101
VN1AMM0335	Serviceable	6.20	83.0	0.076	0.109
AGH3055	Unserviceable	4.94	76.6	0.066	0.085
4298	Unserviceable	5.50	83.1	0.074	0.119
123BQ	Unserviceable	5.23	79.9	0.094	0.104
AG347G	Unserviceable	4.80	74.8	0.083	0.105
48BM	Unserviceable	5.07	78.0	0.095	0.15
94EX	Unserviceable	5.16	79.1	0.092	0.106
AM8FM	Unserviceable	5.22	80.4	0.146	0.178

Table D.6: Spray Characteristic Data at 620 kPa

Nozzle	Category	Spray Diameter [cm]	Spray Cone Angle [°]	Max Out of Round	Max Streakiness
AGM291	Unknown	8.09	85.6	0.094	0.086
AGL824	Unknown	8.92	91.2	0.06	0.104
VN1AMM1301	Unknown	6.72	90.2	0.071	0.144
VN1AMT1839	Unknown	6.83	91.3	0.07	0.177
2192	Unknown	7.57	81.9	0.154	0.09
AGH3055	Unknown	7.94	84.6	0.091	0.11
AGH4084	Unknown	7.82	83.8	0.149	0.117
4298	Unknown	6.94	92.4	0.092	0.139
AG4306	Unknown	8.25	86.8	0.133	0.116
AG4309	Unknown	8.30	87.2	0.247	0.104
AGF6522	Unknown	9.42	94.4	0.152	0.093
AG7553	Unknown	9.07	92.2	0.097	0.094
123BQ	Unknown	7.05	93.3	0.096	0.15
14FN	Unknown	6.58	88.8	0.237	0.168
31AG	Unknown	7.50	81.4	0.127	0.087
AG347G	Unknown	7.96	84.7	0.065	0.078
48BM	Unknown	6.82	91.2	0.133	0.133
83BJ	Unknown	8.75	90.2	0.154	0.123
94EX	Unknown	6.78	90.8	0.147	0.272
AG3599	Unknown	6.79	90.9	0.105	0.155
AGK944	Unknown	7.04	93.3	0.08	0.147
AM8FM	Unknown	6.38	86.7	0.238	0.273
VN1AMM0335	Unknown	7.98	91.4	0.063	0.162
AGM291	Serviceable	6.99	82.9	0.055	0.119
VN1AMM1301	Serviceable	7.18	78.9	0.074	0.124
VN1AMT1839	Serviceable	6.87	76.4	0.065	0.142
2192	Serviceable	6.65	79.6	0.072	0.108
AGH4084	Serviceable	7.01	83.0	0.068	0.101
14FN	Serviceable	7.44	80.9	0.106	0.109
31AG	Serviceable	6.72	80.3	0.065	0.108
47M	Serviceable	7.87	84.3	0.075	0.085
AG3599	Serviceable	7.83	83.8	0.079	0.104
AGK944	Serviceable	7.86	84	0.076	0.122
VN1AMM0335	Serviceable	6.76	80.7	0.059	0.104
AGH3055	Unserviceable	7.00	91.5	0.159	0.117
4298	Unserviceable	7.80	100.3	0.167	0.196
123BQ	Unserviceable	7.43	96.9	0.148	0.108
AG347G	Unserviceable	6.73	88.6	0.166	0.126
48BM	Unserviceable	7.15	94.3	0.177	0.115
94EX	Unserviceable	7.14	94.2	0.226	0.225
AM8FM	Unserviceable	7.44	96.9	0.306	0.369

Table D.7: Unknown Spray Patterns at Different Delivery Pressures

Nozzle	Fuel Delivery Pressure		
	345 kPa	517 kPa	620 kPa
VN1AMT1839
VN1AMM1301
AGM291
2192
123BQ
94EX

Table D.7: (cont): Unknown Spray Patterns at Different Delivery Pressures

Nozzle	Fuel Delivery Pressure		
	345 kPa	517 kPa	620 kPa
AGK944
4298
31AG
AM8FM
48BM
VN1AMM0335

Table D.7: (cont): Unknown Spray Patterns at Different Delivery Pressures

Nozzle	Fuel Delivery Pressure		
	345 kPa	517 kPa	620 kPa
AG347G
14-FN
AG3599
AGH3055
AGH4084
AGL824

Table D.7: (cont): Unknown Spray Patterns at Different Delivery Pressures

Nozzle	Fuel Delivery Pressure		
	345 kPa	517 kPa	620 kPa
83-BJ

Table D.8: Serviceable Spray Patterns at Different Delivery Pressures

Nozzle	Fuel Delivery Pressure		
	345 kPa	517 kPa	620 kPa
47M
VN1AMT1839
VN1AMM1301
AGM291
2192

Table D.8: (cont): Serviceable Spray Patterns at Different Delivery Pressures Continued

Nozzle	Fuel Delivery Pressure		
	345 kPa	517 kPa	620 kPa
AGK944
31AG
VN1AMM0335
14-FN
AG3599
AGH4084

Table D.9: Unserviceable Spray Patterns at Different Delivery Pressures

Nozzle	Fuel Delivery Pressure		
	345 kPa	517 kPa	620 kPa
123BQ
94EX
4298
AM8FM
48BM
AG347G
AGH3055

Table D.10: Fuel Nozzle 47M – Experimentation Data (Part 1)

Time (s)	Thermocouple				Rake Position (deg)	Inlet Air Temperature (K)
	1 (K)	2 (K)	3(K)	4 (K)		
0.0	322.2	311.4	309.1	326.7	61	307.4
27.6	346.9	326.1	320.4	360.6	60	307.8
28.8	347.5	326.6	320.7	360.4	60	307.8
30.0	347.4	326.7	320.4	359.8	60	307.9
31.3	347.5	326.9	320.1	359.0	60	307.9
32.5	347.8	327.2	320.4	359.5	60	307.9
50.2	455.4	397.2	376.8	422.6	59	307.4
51.4	456.3	397.6	377.8	421.9	59	307.4
52.7	455.9	398.3	376.8	419.5	59	307.5
53.9	456.1	398.2	376.9	421.5	59	307.5
55.1	456.9	398.4	377.3	423.7	59	307.6
72.8	598.4	546.6	533.8	547.7	58	307.4
74.1	601.4	547.2	534.5	548.8	58	307.5
75.3	603.0	548.5	535.1	549.2	58	307.6
76.6	604.9	549.4	536.2	550.3	58	307.7
77.8	606.1	551.6	538.1	550.9	58	307.8
95.5	760.3	769.0	814.3	755.9	57	308.0
96.7	765.0	772.3	816.4	758.1	57	307.1
97.9	763.0	770.3	814.2	757.6	57	307.3
99.2	766.1	771.5	815.4	759.7	57	307.4
100.4	763.5	769.9	814.4	758.4	57	307.5
118.1	908.7	942.6	1032.0	914.1	56	307.2
119.3	910.9	946.0	1036.4	917.0	56	307.3
120.5	910.4	948.2	1038.6	914.2	56	307.0
121.8	904.6	944.3	1036.4	913.9	56	307.5
123.0	909.5	946.4	1038.2	915.4	56	307.6
140.7	1030.8	1091.4	1190.8	1013.1	55	307.6
141.9	1031.4	1089.7	1188.4	1013.3	55	307.6
143.2	1035.6	1095.9	1194.7	1014.2	55	307.7
144.4	1040.3	1096.6	1192.8	1013.2	55	308.1
145.6	1036.0	1093.7	1194.4	1015.0	55	308.2
163.3	1187.2	1277.8	1320.5	1106.7	54	308.2
164.5	1179.0	1270.8	1318.3	1106.6	54	308.2
165.8	1182.7	1271.4	1315.3	1104.4	54	308.2
167.0	1185.2	1271.6	1314.7	1107.4	54	308.5
168.3	1191.9	1281.1	1318.5	1108.6	54	308.5
186.0	1153.1	1284.3	1345.9	1006.9	53	308.5
187.2	1161.7	1293.8	1351.4	1009.8	53	308.5
188.4	1153.8	1283.2	1345.6	1008.2	53	308.5
189.7	1168.9	1299.2	1349.8	1009.1	53	308.7
190.9	1169.2	1297.5	1345.2	1007.8	53	308.6
208.6	1037.4	1183.5	1281.4	942.0	52	308.6
209.8	1044.7	1190.8	1283.7	943.4	52	308.6
211.1	1045.0	1191.3	1284.8	943.4	52	308.6

Table D.10: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 1)

Time (s)	Thermocouple				Rake Position (deg)	Inlet Air Temperature (K)
	1 (K)	2 (K)	3(K)	4 (K)		
212.3	1043.3	1189.8	1284.5	943.0	52	308.7
213.5	1032.8	1179.3	1282.6	943.2	52	308.7
231.2	1039.4	1182.1	1274.2	938.6	51	308.7
232.5	1042.5	1185.2	1275.7	938.7	51	308.7
233.7	1043.3	1186.7	1278.9	940.0	51	308.7
234.9	1038.9	1182.5	1277.6	939.0	51	308.8
236.2	1035.6	1181.8	1275.7	937.3	51	308.7
253.8	912.4	1057.9	1220.3	1043.4	50	308.7
255.1	912.3	1057.1	1214.7	1040.4	50	308.7
256.3	913.8	1058.3	1215.1	1040.4	50	308.7
257.5	913.7	1057.5	1213.3	1039.2	50	308.8
258.8	910.5	1054.4	1211.6	1038.0	50	308.7
276.5	866.1	1002.1	1181.8	1133.5	49	308.8
277.7	867.3	1003.1	1182.6	1133.7	49	308.7
278.9	866.4	1002.7	1180.6	1131.2	49	308.7
280.2	866.5	1002.4	1182.4	1134.6	50	307.7
281.4	864.3	1000.2	1182.2	1134.6	49	308.1
299.1	928.2	1084.8	1215.9	1175.8	48	308.2
300.4	928.0	1082.9	1214.4	1176.2	48	308.2
301.6	925.2	1079.8	1211.5	1173.6	48	308.2
302.8	926.6	1081.0	1211.9	1173.5	48	308.2
304.1	929.5	1084.1	1214.0	1175.3	48	308.5
321.8	1033.8	1270.8	1343.0	1167.5	47	308.5
323.0	1033.6	1270.3	1340.7	1161.7	47	308.5
324.2	1036.8	1273.1	1342.8	1162.0	47	308.5
325.5	1037.2	1272.9	1342.7	1163.9	47	308.5
326.7	1041.0	1278.1	1346.6	1170.7	47	308.7
344.4	1146.0	1404.5	1436.1	1223.2	46	308.6
345.6	1150.5	1409.7	1441.4	1226.9	46	308.6
346.9	1146.7	1405.8	1437.7	1226.7	46	307.7
348.1	1144.0	1401.7	1434.6	1225.1	46	308.1
349.3	1144.3	1399.5	1432.8	1228.5	46	308.2
367.0	1236.7	1476.7	1466.4	1287.4	46	308.2
368.3	1236.4	1474.2	1463.8	1283.4	46	308.2
369.5	1236.0	1470.9	1457.4	1276.5	46	308.2
370.8	1235.9	1474.8	1461.6	1275.1	46	308.5
372.0	1231.9	1471.2	1461.0	1278.0	46	308.5
389.7	1273.0	1495.9	1477.4	1399.3	45	308.5
390.9	1273.2	1495.4	1476.9	1398.3	45	308.5
392.2	1271.7	1491.4	1472.8	1395.6	45	308.5
393.4	1270.3	1488.2	1471.3	1389.3	45	308.7
394.6	1269.5	1490.4	1473.1	1393.5	45	308.6
412.3	1289.8	1507.2	1485.5	1418.4	44	308.6
413.6	1289.2	1508.2	1489.0	1421.3	44	306.9
414.8	1292.2	1506.8	1486.0	1418.0	44	306.9

Table D.10: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 1)

Time (s)	Thermocouple				Rake Position (deg)	Inlet Air Temperature (K)
	1 (K)	2 (K)	3(K)	4 (K)		
416.0	1292.6	1505.4	1483.4	1416.4	44	307.0
417.3	1290.6	1503.1	1483.0	1417.5	44	307.0
434.3	1287.0	1499.4	1497.9	1438.9	44	307.5
435.5	1289.5	1501.9	1499.7	1439.9	44	307.6
436.8	1288.5	1501.0	1500.2	1436.8	44	307.6
438.1	1288.7	1502.9	1499.8	1433.1	44	307.6
439.4	1287.6	1500.7	1499.0	1435.1	44	307.7
456.4	1285.4	1497.4	1513.3	1452.3	43	308.1
457.7	1285.7	1495.3	1508.7	1449.8	43	308.2
459.0	1282.7	1492.0	1506.3	1448.4	43	308.2
460.3	1283.4	1493.2	1507.3	1448.6	43	308.2
461.5	1279.7	1490.7	1510.5	1450.2	43	308.2
478.6	1260.8	1480.3	1524.1	1465.7	42	308.5
479.9	1262.9	1481.5	1523.9	1468.3	42	308.5
481.2	1261.6	1479.6	1522.3	1467.5	42	308.5
482.4	1263.7	1481.1	1523.0	1467.6	42	308.5
483.7	1265.2	1479.9	1523.2	1469.9	42	308.5
500.8	1231.2	1458.0	1546.3	1509.6	41	308.7
502.1	1230.3	1459.5	1549.0	1509.9	41	308.6
503.4	1229.8	1458.4	1547.8	1511.8	41	308.6
504.7	1229.3	1456.4	1547.4	1511.4	41	308.6
505.9	1235.7	1462.6	1548.4	1512.8	41	308.6
523.0	1218.3	1448.0	1572.3	1544.5	40	308.7
524.3	1216.9	1445.1	1570.1	1546.0	40	308.7
525.6	1215.6	1445.2	1568.3	1537.4	40	308.7
526.9	1214.0	1443.0	1570.5	1536.2	40	308.7
528.1	1212.6	1441.8	1566.3	1534.4	40	308.7
545.2	1195.6	1422.7	1586.4	1563.5	39	308.8
546.5	1195.6	1424.9	1587.9	1563.5	39	308.7
547.7	1196.9	1426.2	1588.9	1565.5	39	308.7
549.0	1195.8	1424.5	1586.6	1557.9	39	308.7
550.3	1194.4	1422.9	1582.8	1555.9	39	308.7
567.4	1177.4	1397.5	1599.9	1579.2	38	308.8
568.6	1178.0	1397.6	1600.1	1580.8	38	308.7
569.9	1176.8	1396.5	1599.7	1579.3	38	308.8
571.2	1174.1	1393.5	1600.2	1578.6	38	308.7
572.5	1171.5	1389.7	1597.9	1582.0	38	308.7
590.4	1183.3	1399.6	1615.8	1597.5	37	308.7
591.6	1186.1	1400.8	1616.5	1600.6	37	308.7
592.8	1187.1	1402.1	1616.8	1600.4	37	308.7
594.1	1188.3	1404.5	1619.8	1605.7	37	308.8
595.3	1186.7	1401.2	1616.6	1601.9	37	308.8
613.1	1226.6	1459.9	1639.8	1617.8	36	308.9
614.4	1226.4	1459.5	1638.1	1612.9	36	308.9
615.6	1228.5	1460.7	1640.5	1619.7	36	308.9

Table D.10: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 1)

Time (s)	Thermocouple				Rake Position (deg)	Inlet Air Temperature (K)
	1 (K)	2 (K)	3(K)	4 (K)		
616.9	1227.2	1460.3	1638.8	1614.3	36	309.0
618.1	1224.9	1458.2	1637.5	1612.1	36	309.0
635.9	1241.1	1499.5	1654.0	1622.8	36	309.2
637.1	1238.9	1496.5	1655.7	1625.6	36	309.2
638.4	1239.1	1497.6	1657.0	1628.2	36	309.2
639.6	1238.6	1498.8	1657.8	1627.2	36	309.2
640.9	1241.6	1503.5	1656.2	1623.6	36	309.2
658.7	1236.3	1518.0	1668.7	1621.5	35	309.3
659.9	1236.7	1519.8	1669.8	1624.4	35	309.3
661.2	1237.9	1522.1	1669.2	1624.4	35	309.3
662.4	1239.0	1524.8	1667.4	1616.9	35	309.3
663.6	1237.9	1520.3	1667.9	1618.4	35	309.3
681.4	1285.6	1568.9	1677.4	1596.4	34	309.3
682.7	1282.1	1563.3	1673.4	1594.3	34	309.3
683.9	1280.6	1560.3	1674.9	1595.4	34	309.3
685.2	1280.6	1562.1	1676.5	1596.3	34	309.3
686.4	1281.8	1563.0	1675.7	1594.4	34	309.3
704.2	1370.1	1624.1	1680.4	1549.7	33	309.4
705.5	1373.1	1626.5	1682.4	1554.4	33	309.4
706.7	1374.3	1628.1	1679.6	1545.7	33	309.4
708.0	1374.2	1628.0	1681.2	1549.7	33	309.4
709.2	1375.4	1629.5	1684.3	1552.4	33	309.4
727.0	1375.8	1591.1	1641.9	1513.5	32	309.2
728.3	1374.9	1587.3	1639.2	1510.1	32	309.2
729.5	1375.0	1587.4	1640.2	1511.9	32	309.2
730.8	1374.5	1585.3	1637.3	1513.4	32	309.2
732.0	1373.6	1585.8	1637.3	1511.2	32	309.2
749.8	1308.4	1495.4	1576.7	1524.9	31	308.7
751.0	1309.7	1496.6	1578.4	1529.1	31	308.6
752.3	1311.7	1497.4	1578.3	1529.7	31	308.6
753.5	1308.7	1493.8	1573.6	1521.8	31	308.6
754.8	1308.0	1493.1	1572.9	1520.4	31	308.5
772.6	1310.4	1504.0	1589.4	1540.5	30	308.3
773.8	1313.0	1506.7	1589.6	1540.9	30	308.3
775.0	1313.0	1506.8	1590.4	1541.0	30	308.3
776.3	1310.9	1504.2	1589.2	1539.5	30	308.3
777.5	1312.0	1505.9	1591.2	1543.3	30	308.3
795.4	1372.6	1592.0	1645.9	1565.9	30	308.6
796.6	1371.5	1592.7	1647.8	1572.0	30	308.7
797.8	1372.2	1594.6	1648.1	1571.7	30	308.7
799.1	1372.9	1594.4	1647.9	1571.0	30	308.7
800.3	1373.3	1593.1	1645.7	1567.5	30	308.7
818.1	1354.9	1597.5	1658.5	1563.0	29	308.7
819.4	1352.0	1596.8	1658.5	1561.5	29	308.7
820.6	1353.8	1596.4	1658.3	1559.2	29	308.8

Table D.10: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 1)

Time (s)	Thermocouple				Rake Position (deg)	Inlet Air Temperature (K)
	1 (K)	2 (K)	3(K)	4 (K)		
821.9	1352.2	1592.8	1655.3	1557.7	29	308.8
823.1	1354.3	1593.8	1656.5	1559.9	29	308.8
840.9	1237.8	1483.3	1634.8	1566.8	28	308.7
842.2	1239.6	1487.1	1635.9	1566.3	28	308.7
843.4	1241.4	1491.4	1638.2	1568.9	28	308.7
844.6	1242.0	1493.2	1636.4	1565.7	28	308.7
845.9	1239.8	1488.6	1635.5	1566.5	28	308.7
863.7	1122.0	1372.5	1607.2	1581.8	27	308.4
864.9	1122.9	1373.9	1607.8	1580.0	27	308.4
866.2	1123.0	1375.2	1609.2	1583.1	27	308.4
867.4	1122.1	1373.3	1608.3	1581.2	27	308.4
868.7	1121.3	1370.3	1606.9	1584.1	27	308.4
886.5	1061.8	1318.2	1588.0	1574.1	26	308.0
887.7	1062.1	1321.1	1591.4	1576.1	26	308.0
889.0	1063.6	1322.7	1590.9	1575.3	26	308.0
890.2	1061.4	1316.8	1590.0	1578.8	26	308.0
891.4	1061.3	1318.4	1590.3	1575.7	26	308.0
909.2	1048.1	1310.6	1576.3	1561.4	25	307.7
910.5	1048.0	1312.1	1578.5	1564.1	25	307.7
911.7	1046.9	1309.1	1579.8	1568.8	25	307.7
913.0	1048.4	1313.0	1580.4	1563.9	25	307.7
914.2	1048.2	1311.4	1578.8	1560.9	25	307.6
932.0	1065.2	1340.7	1574.7	1537.1	24	307.5
933.3	1062.9	1337.7	1570.5	1534.9	24	307.5
934.5	1060.5	1334.7	1570.5	1538.7	24	307.4
935.8	1061.3	1334.3	1572.4	1542.4	24	307.4
937.0	1060.8	1333.3	1571.5	1541.1	24	307.4
954.9	1069.0	1347.5	1567.5	1529.2	23	307.3
956.1	1068.2	1346.3	1565.5	1525.6	23	307.3
957.4	1069.2	1347.5	1564.7	1525.5	23	307.3
958.6	1067.8	1348.2	1564.6	1522.4	23	307.2
959.9	1065.6	1344.4	1562.9	1526.0	23	307.2
977.7	1071.9	1340.5	1558.1	1511.8	23	307.3
979.0	1071.2	1340.2	1557.0	1514.4	23	307.3
980.2	1070.0	1338.2	1557.8	1516.9	23	307.3
981.5	1071.0	1337.4	1556.4	1517.9	23	307.3
982.7	1067.5	1334.5	1555.1	1517.2	23	307.3
1000.5	1104.0	1341.3	1540.6	1499.0	22	307.4
1001.8	1103.8	1344.4	1542.8	1500.0	22	307.4
1003.0	1107.9	1348.5	1545.6	1501.7	22	307.4
1004.3	1104.3	1342.7	1544.5	1501.4	22	307.5
1005.5	1105.7	1342.0	1542.0	1500.9	22	307.5
1023.4	1162.4	1374.0	1535.5	1491.3	21	307.8
1024.6	1158.4	1369.7	1533.9	1490.7	21	307.8
1025.8	1161.7	1371.3	1532.4	1489.2	21	307.9

Table D.10: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 1)

Time (s)	Thermocouple				Rake Position (deg)	Inlet Air Temperature (K)
	1 (K)	2 (K)	3(K)	4 (K)		
1027.1	1163.9	1372.2	1530.5	1486.0	21	307.9
1028.4	1171.0	1378.3	1531.1	1483.8	21	307.9
1046.2	1258.3	1442.6	1530.5	1481.8	20	308.2
1047.4	1265.7	1448.5	1528.1	1477.8	20	308.2
1048.7	1265.7	1448.3	1527.3	1475.7	20	308.2
1049.9	1268.0	1449.2	1525.1	1472.7	20	308.2
1051.2	1255.8	1440.2	1526.1	1471.0	20	308.2
1069.0	1307.3	1493.0	1523.8	1469.1	19	308.6
1070.2	1307.7	1492.6	1525.2	1470.9	19	308.6
1071.4	1313.5	1496.3	1523.0	1464.8	19	308.6
1072.7	1306.3	1490.0	1525.0	1472.0	19	308.6
1074.0	1302.8	1489.7	1526.3	1474.5	19	308.7
1091.8	1284.7	1501.2	1523.2	1467.5	18	308.8
1093.0	1291.5	1505.4	1526.4	1469.2	18	308.8
1094.2	1287.7	1502.7	1525.6	1467.8	18	308.8
1095.5	1283.3	1499.6	1525.6	1468.3	18	308.8
1096.7	1280.2	1497.7	1522.2	1465.9	18	308.9
1114.5	1233.9	1491.5	1527.4	1464.7	17	308.7
1115.8	1231.7	1490.4	1527.0	1466.2	17	308.7
1117.0	1228.5	1487.0	1525.1	1465.2	17	308.6
1118.3	1228.9	1488.0	1523.1	1461.3	17	308.6
1119.5	1228.7	1488.3	1526.7	1466.8	17	308.6
1137.3	1178.2	1479.3	1532.1	1458.5	16	308.3
1138.6	1175.8	1478.1	1532.1	1458.5	16	308.3
1139.8	1177.6	1479.1	1530.9	1457.1	16	308.3
1141.1	1171.9	1475.0	1534.7	1462.5	16	308.3
1142.3	1171.1	1473.9	1532.4	1459.3	16	308.3
1160.1	1130.9	1450.4	1521.4	1459.2	15	307.9
1161.4	1125.5	1446.0	1521.7	1462.6	15	307.8
1162.6	1122.3	1442.8	1521.9	1465.4	15	307.7
1163.9	1122.6	1442.9	1520.2	1463.7	15	307.7
1165.1	1124.1	1447.1	1525.3	1468.2	15	307.6
1183.0	1032.6	1285.7	1402.0	1385.9	14	308.7
1184.2	1030.5	1284.4	1399.5	1381.0	14	308.6
1185.5	1029.9	1283.7	1398.0	1381.0	14	308.6
1186.7	1026.8	1280.7	1396.5	1380.7	14	308.5
1187.9	1026.2	1279.8	1398.8	1383.2	14	308.4
1205.7	892.0	1118.7	1295.9	1327.5	13	307.3
1207.0	892.3	1117.5	1294.7	1327.1	13	307.3
1208.2	889.6	1116.9	1294.6	1325.6	13	307.2
1209.5	892.0	1120.6	1298.8	1328.8	13	307.1
1210.7	890.5	1118.5	1297.4	1328.4	13	307.0
1228.5	800.6	995.9	1282.5	1281.4	12	308.3
1229.8	803.2	998.8	1285.3	1282.1	12	308.2
1231.0	805.3	1000.0	1284.7	1280.6	12	308.1

Table D.10: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 1)

Time (s)	Thermocouple				Rake Position (deg)	Inlet Air Temperature (K)
	1 (K)	2 (K)	3(K)	4 (K)		
1232.3	806.4	1000.0	1282.5	1278.9	12	308.1
1233.5	805.9	1000.3	1282.4	1278.3	12	308.0
1251.3	810.1	985.7	1372.7	1262.7	11	307.5
1252.5	810.2	986.9	1368.5	1258.5	11	307.4
1253.8	808.6	985.4	1369.6	1260.2	11	307.4
1255.0	810.5	985.1	1370.5	1261.9	11	307.4
1256.3	808.8	983.6	1371.8	1264.2	11	307.4
1274.2	883.8	1056.6	1459.9	1312.6	10	307.3
1275.4	886.6	1059.3	1463.5	1323.2	10	307.3
1276.7	889.4	1059.9	1464.4	1332.0	10	307.3
1277.9	890.8	1060.9	1465.4	1339.7	10	307.3
1279.2	895.6	1067.0	1468.3	1336.7	10	307.3
1297.0	1002.0	1210.0	1470.7	1396.2	9	307.6
1298.2	1004.3	1213.0	1470.9	1394.3	9	307.6
1299.4	1005.8	1214.3	1468.4	1394.0	9	307.6
1300.7	1004.6	1212.3	1467.5	1393.0	9	307.6
1301.9	1007.6	1215.2	1469.0	1394.9	9	307.6
1319.8	1003.3	1194.3	1390.8	1356.1	8	307.7
1321.0	1006.1	1198.0	1389.5	1353.6	8	307.6
1322.2	1006.0	1196.2	1387.2	1350.3	8	307.7
1323.5	1005.5	1192.4	1382.7	1346.7	8	307.6
1324.7	1005.6	1192.9	1384.4	1349.5	8	307.6
1342.5	924.2	1050.8	1251.5	1248.1	7	307.4
1343.8	925.0	1050.1	1250.2	1249.4	7	307.3
1345.0	921.8	1049.3	1249.1	1247.8	7	307.3
1346.3	921.9	1052.5	1251.4	1247.2	7	307.3
1347.5	923.8	1050.0	1243.6	1241.5	7	307.3
1364.2	852.4	943.8	1136.5	1174.9	6	307.5
1365.4	850.6	942.5	1133.4	1173.0	6	307.4
1366.6	850.6	940.7	1129.4	1169.7	6	307.3
1367.9	848.5	938.7	1130.2	1172.0	6	307.2
1369.1	845.4	936.6	1126.7	1168.9	6	307.2
1386.9	785.9	871.8	1043.9	1087.4	5	308.0
1388.2	785.9	870.6	1038.7	1083.9	5	307.9
1389.4	784.2	869.3	1038.7	1084.5	5	307.8
1390.7	784.6	871.8	1044.5	1087.4	5	307.7
1391.9	781.4	869.8	1041.0	1080.6	5	307.6
1409.7	671.1	750.4	844.1	845.0	4	307.5
1411.0	668.4	748.9	842.2	842.1	4	307.4
1412.2	667.2	745.7	839.0	841.0	4	307.2
1413.5	664.8	743.5	836.3	837.5	4	307.1
1414.7	664.3	743.8	836.8	838.1	4	307.0
1432.5	555.4	564.1	622.8	631.6	3	308.3
1433.8	553.3	559.2	618.1	628.6	3	308.2
1435.0	550.2	557.8	617.6	629.5	3	308.1

Table D.10: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 1)

Time (s)	Thermocouple				Rake Position (deg)	Inlet Air Temperature (K)
	1 (K)	2 (K)	3(K)	4 (K)		
1436.2	552.4	557.9	620.3	632.9	3	308.0
1437.5	551.6	553.6	616.5	631.3	3	307.8
1455.3	444.3	401.3	415.3	424.8	2	307.3
1456.6	444.8	399.7	415.0	424.7	2	307.2
1457.8	445.8	399.2	414.5	424.4	2	307.2
1459.1	445.7	398.8	413.2	423.7	2	307.0
1460.3	444.7	398.7	414.5	425.1	2	308.9
1478.2	353.6	330.7	332.1	342.5	1	308.0
1479.4	353.8	330.8	331.9	342.1	1	307.9
1480.6	353.2	330.5	331.6	342.1	1	307.8
1481.9	354.1	330.4	331.6	342.1	1	307.7
1483.1	353.8	330.6	331.6	342.2	1	307.6
1500.9	318.6	317.3	317.2	316.1	0	307.9
1502.2	318.7	317.4	317.2	316.2	0	307.9
1503.4	318.7	317.3	317.1	316.2	0	307.8
1504.7	318.5	317.3	317.1	316.0	0	307.7
1505.9	318.7	317.1	317.1	316.1	0	307.7

Table D.11: Fuel Nozzle 47M – Experimentation Data (Part 2)

Time (s)	Inlet Air Static Pressure (Pa)	Venturi Differential Pressure (Pa)	Fuel Mass Flow (g/s)	Air Mass Flow (kg/s)	Fuel Pressure (Pa)
0.0	17713	2407	9.473	0.979	512587
27.6	17720	2421	9.556	0.984	512075
28.8	17353	2473	9.431	0.977	511093
30.0	17581	2413	9.513	0.974	515654
31.3	17624	2461	9.471	0.980	515868
32.5	17587	2422	9.431	0.977	506992
50.2	17644	2406	9.431	0.975	509746
51.4	17515	2467	9.392	0.982	517042
52.7	17671	2403	9.473	0.988	509040
53.9	17598	2403	9.473	0.976	511388
55.1	17656	2422	9.473	0.981	509552
72.8	17452	2448	9.480	0.979	510343
74.1	17616	2443	9.541	0.987	510715
75.3	17789	2415	9.464	0.981	510560
76.6	17537	2431	9.513	0.982	510490
77.8	17656	2410	9.458	0.987	507626
95.5	17420	2452	9.442	0.976	512117
96.7	17635	2401	9.473	0.976	511044
97.9	17501	2453	9.449	0.978	507015
99.2	17588	2421	9.381	0.981	508024
100.4	17617	2442	9.504	0.971	514534
118.1	17538	2420	9.502	0.978	510878
119.3	17625	2423	9.535	0.973	512057
120.5	17600	2430	9.529	0.974	507540
121.8	17316	2468	9.448	0.978	510043
123.0	17633	2414	9.513	0.977	510881
140.7	17650	2418	9.554	0.982	509464
141.9	17622	2425	9.473	0.971	508321
143.2	17510	2427	9.431	0.986	508665
144.4	17548	2410	9.513	0.981	512251
145.6	17585	2438	9.473	0.979	513035
163.3	17502	2452	9.556	0.984	511002
164.5	17472	2418	9.431	0.977	509009
165.8	17607	2410	9.513	0.974	513378
167.0	17530	2417	9.471	0.980	515051
168.3	17527	2466	9.431	0.977	512854
186.0	17676	2419	9.431	0.975	515387
187.2	17709	2404	9.392	0.982	511569
188.4	17596	2400	9.473	0.988	517120
189.7	17627	2424	9.473	0.976	514914
190.9	17565	2411	9.473	0.981	515119

Table D.11: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 2)

Time (s)	Inlet Air Static Pressure (Pa)	Venturi Differential Pressure (Pa)	Fuel Mass Flow (g/s)	Air Mass Flow (kg/s)	Fuel Pressure (Pa)
208.6	17413	2472	9.480	0.979	511706
209.8	17531	2413	9.541	0.987	513154
211.1	17619	2424	9.464	0.981	514309
212.3	17667	2375	9.513	0.982	516321
213.5	17520	2425	9.458	0.987	511796
231.2	17668	2380	9.442	0.976	510457
232.5	17605	2405	9.473	0.976	517482
233.7	17807	2427	9.449	0.978	516398
234.9	17641	2388	9.381	0.981	513849
236.2	17575	2421	9.504	0.971	510547
253.8	17525	2424	9.502	0.978	514110
255.1	17563	2422	9.535	0.973	518542
256.3	17530	2403	9.529	0.974	513270
257.5	17659	2398	9.448	0.978	509006
258.8	17490	2421	9.513	0.977	514620
276.5	17566	2388	9.554	0.982	512417
277.7	17548	2455	9.473	0.971	512748
278.9	17476	2448	9.431	0.986	512763
280.2	17401	2439	9.513	0.981	515126
281.4	17475	2424	9.544	0.989	507995
299.1	17665	2404	9.453	0.985	509723
300.4	17572	2434	9.555	0.991	515855
301.6	17664	2424	9.522	0.989	510858
302.8	17364	2441	9.514	0.991	510498
304.1	17613	2433	9.594	0.991	516057
321.8	17221	2476	9.552	0.998	515961
323.0	17571	2424	9.555	0.988	516890
324.2	17539	2434	9.594	0.990	517689
325.5	17492	2397	9.514	0.983	514674
326.7	17530	2432	9.555	0.990	507380
344.4	17624	2385	9.482	0.981	510415
345.6	17611	2408	9.593	0.985	514371
346.9	17596	2410	9.554	0.985	514436
348.1	17679	2431	9.534	0.990	509648
349.3	17546	2422	9.500	0.988	512859
367.0	17592	2404	9.594	0.984	512561
368.3	17473	2423	9.513	0.987	514211
369.5	17573	2399	9.473	0.983	513435
370.8	17502	2428	9.513	0.988	515212
372.0	17620	2382	9.513	0.980	512879
389.7	17606	2393	9.513	0.982	511411
390.9	17414	2423	9.533	0.987	513578
392.2	17443	2438	9.446	0.990	513862
393.4	17734	2411	9.513	0.986	510209
394.6	17560	2419	9.473	0.987	512864

D.11: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 2)

Time (s)	Inlet Air Static Pressure (Pa)	Venturi Differential Pressure (Pa)	Fuel Mass Flow (g/s)	Air Mass Flow (kg/s)	Fuel Pressure (Pa)
412.3	17364	2464	9.531	0.995	508293
413.6	17573	2394	9.484	0.982	510079
414.8	17535	2434	9.479	0.989	509030
416.0	17476	2425	9.513	0.987	515917
417.3	17652	2400	9.513	0.983	506077
434.3	17623	2415	9.475	0.986	515137
435.5	17594	2376	9.520	0.978	516243
436.8	17566	2372	9.594	0.977	512577
479.9	17362	2414	9.513	0.976	507561
481.2	17552	2401	9.431	0.976	509981
482.4	17597	2402	9.431	0.981	517932
483.7	17544	2405	9.473	0.977	511649
500.8	17576	2415	9.467	0.981	510108
502.1	17635	2377	9.473	0.981	510030
503.4	17522	2426	9.431	0.978	509293
504.7	17587	2402	9.473	0.981	512334
505.9	17638	2409	9.513	0.985	509614
523.0	17619	2393	9.513	0.981	512936
524.3	17628	2416	9.443	0.981	506098
525.6	17501	2398	9.438	0.983	511731
526.9	17442	2437	9.431	0.984	510708
528.1	17726	2391	9.517	0.974	504683
545.2	17575	2387	9.475	0.983	508419
546.5	17655	2387	9.468	0.976	513441
547.7	17535	2398	9.480	0.983	510265
549.0	17575	2416	9.468	0.982	509738
550.3	17523	2434	9.478	0.978	507512
567.4	17520	2445	9.503	0.975	511791
568.6	17661	2376	9.463	0.985	509123
569.9	17561	2409	9.467	0.978	510439
571.2	17592	2422	9.499	0.978	504593
572.5	17451	2438	9.508	0.981	508021
590.4	17454	2407	9.392	0.983	511781
591.6	17566	2427	9.433	0.977	515281
592.8	17603	2412	9.473	0.979	511157
594.1	17583	2383	9.431	0.974	513234
595.3	17574	2391	9.473	0.985	503608
613.1	17585	2399	9.633	0.979	508955
614.4	17553	2388	9.474	0.980	511121
615.6	17424	2389	9.470	0.987	510801
616.9	17669	2411	9.431	0.984	513903
618.1	17430	2425	9.509	0.986	508401
635.9	17503	2396	9.431	0.980	511755
637.1	17553	2391	9.465	0.979	510664
638.4	17500	2382	9.528	0.977	508176

Table D.11: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 2)

Time (s)	Inlet Air Static Pressure (Pa)	Venturi Differential Pressure (Pa)	Fuel Mass Flow (g/s)	Air Mass Flow (kg/s)	Fuel Pressure (Pa)
658.7	17762	2383	9.393	0.979	511478
659.9	17610	2397	9.513	0.981	512442
661.2	17639	2399	9.464	0.981	509195
662.4	17536	2417	9.490	0.985	504469
663.6	17567	2424	9.464	0.986	508091
681.4	17634	2377	9.431	0.977	505242
682.7	17492	2394	9.431	0.980	511191
683.9	17612	2372	9.431	0.976	509143
685.2	17492	2434	9.475	0.988	511592
686.4	17553	2399	9.501	0.981	507848
704.2	17611	2362	9.496	0.973	504295
705.5	17507	2382	9.473	0.977	512380
706.7	17475	2421	9.441	0.985	513252
708.0	17502	2412	9.473	0.983	510415
709.2	17653	2395	9.474	0.980	513278
727.0	17344	2439	9.549	0.988	505306
728.3	17453	2423	9.513	0.985	509479
729.5	17549	2398	9.473	0.980	512714
730.8	17620	2366	9.513	0.974	513940
732.0	17652	2373	9.473	0.976	512373
749.8	17557	2410	9.473	0.983	512942
751.0	17688	2373	9.513	0.976	506643
752.3	17468	2376	9.431	0.976	513391
753.5	17493	2403	9.431	0.981	515005
754.8	17557	2380	9.473	0.977	507840
772.6	17402	2405	9.467	0.981	506268
773.8	17479	2401	9.473	0.981	507895
775.0	17532	2385	9.431	0.978	506131
776.3	17476	2402	9.473	0.981	508039
777.5	17464	2423	9.513	0.985	507845
795.4	17562	2399	9.513	0.981	512210
796.6	17524	2402	9.443	0.981	507352
797.8	17576	2411	9.438	0.983	510061
799.1	17576	2414	9.431	0.984	508331
800.3	17699	2362	9.517	0.974	509588
818.1	17562	2410	9.475	0.983	510775
819.4	17656	2374	9.468	0.976	511106
820.6	17389	2416	9.480	0.983	504725
821.9	17522	2408	9.468	0.982	512106
823.1	17507	2390	9.478	0.978	509211
840.9	17605	2374	9.503	0.975	513921
842.2	17421	2427	9.463	0.985	514529
843.4	17538	2386	9.467	0.978	508668
844.6	17615	2385	9.499	0.978	509565
845.9	17605	2400	9.508	0.981	506907

Table D.11: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 2)

Time (s)	Inlet Air Static Pressure (Pa)	Venturi Differential Pressure (Pa)	Fuel Mass Flow (g/s)	Air Mass Flow (kg/s)	Fuel Pressure (Pa)
866.2	17542	2393	9.473	0.979	503970
867.4	17652	2366	9.431	0.974	507814
868.7	17513	2422	9.473	0.985	511910
886.5	17568	2394	9.633	0.979	510152
887.7	17529	2397	9.474	0.980	510907
889.0	17474	2433	9.470	0.987	507302
890.2	17537	2431	9.474	0.987	509164
891.4	17550	2383	9.472	0.977	509270
909.2	17649	2348	9.413	0.970	512577
910.5	17541	2400	9.513	0.980	506584
911.7	17527	2400	9.513	0.980	508158
913.0	17615	2379	9.392	0.976	515183
914.2	17529	2404	9.431	0.981	511196
932.0	17483	2415	9.513	0.983	506842
933.3	17539	2379	9.473	0.976	506568
934.5	17578	2386	9.508	0.977	504285
935.8	17571	2393	9.438	0.979	508450
937.0	17641	2405	9.329	0.982	511504
954.9	17244	2456	9.450	0.990	508960
956.1	17485	2387	9.444	0.977	506695
957.4	17598	2408	9.387	0.982	509177
958.6	17614	2401	9.439	0.981	508970
959.9	17492	2422	9.461	0.984	505242
977.7	17545	2392	9.473	0.979	505793
979.0	17379	2423	9.556	0.984	506912
980.2	17555	2383	9.431	0.977	511677
981.5	17546	2372	9.513	0.974	506240
982.7	17491	2403	9.471	0.980	508882
1000.	17645	2381	9.431	0.977	505754
1001.	17600	2374	9.431	0.975	507059
1003.	17623	2406	9.392	0.982	507476
1004.	17435	2444	9.473	0.988	511734
1005.	17583	2381	9.473	0.976	507502
1023.	17585	2406	9.473	0.981	507434
1024.	17489	2397	9.480	0.979	509810
1025.	17368	2439	9.541	0.987	502788
1027.	17535	2406	9.464	0.981	506307
1028.	17483	2413	9.513	0.982	505234
1046.	17346	2439	9.458	0.987	503835
1047.	17448	2381	9.442	0.976	507194
1048.	17563	2378	9.473	0.976	506343
1049.	17357	2396	9.449	0.978	509441
1051.	17493	2405	9.381	0.981	509679
1069.	17618	2353	9.504	0.971	507093

Table D.11: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 2)

Time (s)	Inlet Air Static Pressure (Pa)	Venturi Differential Pressure (Pa)	Fuel Mass Flow (g/s)	Air Mass Flow (kg/s)	Fuel Pressure (Pa)
1074.	17555	2392	9.448	0.978	512598
1091.	17506	2388	9.513	0.977	505131
1093.	17453	2414	9.554	0.982	505570
1094.	17579	2354	9.473	0.971	505348
1095.	17538	2432	9.431	0.986	506842
1096.	17606	2404	9.513	0.981	504735
1114.	17538	2398	9.593	0.979	509940
1115.	17556	2413	9.554	0.982	509976
1117.	17679	2390	9.513	0.978	503750
1118.	17626	2356	9.513	0.971	504880
1119.	17576	2423	9.473	0.984	503572
1137.	17545	2409	9.473	0.982	510333
1138.	17457	2397	9.451	0.979	511721
1139.	17504	2396	9.557	0.979	503189
1141.	17536	2423	9.431	0.984	505407
1142.	17408	2449	9.451	0.989	506441
1160.	17511	2416	9.482	0.983	509055
1161.	17399	2429	9.473	0.985	510141
1162.	17559	2394	9.445	0.979	511142
1163.	17505	2381	9.473	0.976	504947
1165.	17485	2398	9.473	0.979	501935
1183.	17557	2410	9.473	0.982	504288
1184.	17587	2391	9.513	0.978	509542
1185.	17528	2391	9.473	0.978	501937
1186.	17513	2415	9.431	0.983	506219
1187.	17493	2410	9.473	0.981	507209
1205.	17500	2395	9.431	0.978	508282
1207.	17469	2433	9.554	0.986	509733
1208.	17482	2420	9.473	0.983	510126
1209.	17468	2413	9.513	0.982	509834
1210.	17518	2396	9.431	0.979	506607
1228.	17470	2395	9.473	0.978	507951
1229.	17473	2401	9.473	0.979	508264
1231.	17451	2394	9.513	0.978	507106
1232.	17604	2429	9.513	0.986	506059
1233.	17537	2395	9.513	0.978	507204
1251.	17414	2403	9.513	0.979	505754
1252.	17499	2412	9.431	0.982	507015
1253.	17379	2399	9.513	0.978	507871
1255.	17521	2396	9.431	0.978	508182
1256.	17556	2366	9.431	0.972	507023
1274.	17463	2414	9.513	0.982	509348
1275.	17409	2388	9.535	0.976	506579
1276.	17419	2409	9.467	0.980	503385
1277.	17464	2377	9.431	0.974	510651

Table D.11: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 2)

Time (s)	Inlet Air Static Pressure (Pa)	Venturi Differential Pressure (Pa)	Fuel Mass Flow (g/s)	Air Mass Flow (kg/s)	Fuel Pressure (Pa)
1298.2	17480	2394	9.473	0.978	500735
1299.4	17489	2394	9.473	0.978	507114
1300.7	17495	2397	9.513	0.978	505593
1301.9	17448	2422	9.392	0.983	509759
1319.8	17457	2408	9.494	0.981	500660
1321.0	17444	2404	9.450	0.980	508691
1322.2	17405	2421	9.536	0.983	507587
1323.5	17493	2404	9.473	0.980	505164
1324.7	17527	2402	9.418	0.980	509146
1342.5	17367	2420	9.513	0.983	503274
1343.8	17421	2421	9.470	0.983	510646
1345.0	17456	2413	9.506	0.982	514485
1346.3	17434	2410	9.434	0.981	507988
1347.5	17583	2405	9.431	0.981	508047
1364.2	17419	2446	9.440	0.988	503820
1365.4	17513	2404	9.473	0.980	504735
1366.6	17330	2421	9.473	0.983	504177
1367.9	17535	2384	9.593	0.976	504445
1369.1	17452	2426	9.513	0.984	504311
1386.9	17422	2440	9.434	0.987	511336
1388.2	17427	2399	9.434	0.979	506581
1389.4	17491	2425	9.473	0.984	502478
1390.7	17642	2369	9.473	0.974	503926
1391.9	17418	2407	9.392	0.980	503528
1409.7	17410	2415	9.473	0.982	505115
1411.0	17497	2397	9.473	0.978	504637
1412.2	17427	2438	9.392	0.986	506382
1413.5	17568	2388	9.473	0.977	506167
1414.7	17388	2429	9.431	0.985	501345
1432.5	17453	2418	9.473	0.983	498287
1433.8	17522	2385	9.366	0.976	509009
1435.0	17625	2374	9.428	0.974	509650
1436.2	17441	2419	9.434	0.983	504779
1437.5	17456	2426	9.449	0.984	508918
1455.3	17601	2404	9.506	0.980	502310
1456.6	17570	2370	9.473	0.973	503812
1457.8	17567	2407	9.392	0.981	508598
1459.1	17424	2409	9.473	0.981	502046
1460.3	17299	2442	9.431	0.987	502069
1478.2	17426	2424	9.474	0.984	504603
1479.4	17512	2402	9.431	0.980	504975
1480.6	17367	2441	9.431	0.987	507675
1481.9	17288	2438	9.387	0.986	513340
1483.1	17431	2435	9.431	0.986	509117
1500.9	17467	2407	9.431	0.981	507701

Table D.11: (cont.): Fuel Nozzle 47M – Experimentation Data (Part 2)

Time (s)	Inlet Air Static Pressure (Pa)	Venturi Differential Pressure (Pa)	Fuel Mass Flow (g/s)	Air Mass Flow (kg/s)	Fuel Pressure (Pa)
1502.2	17620	2367	9.473	0.974	499230
1503.4	17543	2373	9.392	0.983	506090
1504.7	17500	2388	9.472	0.984	506033
1505.9	17491	2421	9.513	0.980	505534

Table D.12: Experimental Results and Parameters Used for Adiabatic Flame Temperature Calculation of Unknown Specimens

Specimen	Calculated Adiabatic Flame Temperature (K)	Average Chamber Inlet Air Temperature [K]	Average Exit Gas Temperature (K)	Maximum Exit Gas Temperature (K)	Ambient Pressure (atm)	Average Equivalence Ratio	Pattern Factor
AGM291	1190	319	1194	1646	1.000	0.282	0.52
AGL824	1192	316	1181	1652	0.998	0.282	0.54
VN1AMM1301	1187	315	1203	1660	1.005	0.283	0.51
VN1AMT1839	1190	313	1242	1632	1.005	0.284	0.42
2192	1192	319	1198	1626	1.005	0.283	0.52
AG3055	1190	319	1204	1639	1.005	0.284	0.56
AGH4084	1187	318	1189	1660	1.000	0.283	0.54
4298	1186	316	1180	1663	1.000	0.282	0.53
AG4306	1185	313	1162	1631	0.998	0.283	0.55
AG4309	1185	315	1177	1618	0.998	0.284	0.51
AG56522	1193	317	1199	1635	0.998	0.286	0.49
AG7553	1185	315	1187	1648	0.998	0.282	0.53
123BQ	1185	315	1232	1703	1.005	0.283	0.51
14FN	1191	317	1184	1663	1.005	0.283	0.55
31AG	1192	319	1212	1646	0.998	0.281	0.49
AG347G	1183	319	1193	1656	0.998	0.283	0.53
48BM	1180	312	1227	1694	1.005	0.283	0.55
83BJ	1189	315	1189	1652	0.998	0.281	0.53
94EX	1178	315	1215	1679	1.005	0.283	0.52
AG3599	1193	316	1182	1658	0.998	0.283	0.55
AGK944	1185	315	1178	1662	0.998	0.284	0.56
AM8FM	1192	313	1219	1720	1.005	0.283	0.55
VN1AMM0335	1185	318	1192	1637	1.000	0.282	0.51

Table D.13: Experimental Results and Parameters Used for Adiabatic Flame Temperature Calculation of Serviceable Specimens

Specimen	Calculated Adiabatic Flame Temperature (K)	Average Chamber Inlet Air Temp [K]	Average Exit Gas Temperature (K)	Maximum Exit Gas Temperature (K)	Ambient Pressure (atm)	Average Equivalence Ratio	Pattern Factor
AGM291	1185	315	1184	1617	1.002	0.283	0.50
VN1AMM1301	1193	317	1169	1590	0.998	0.285	0.49
VN1AMT1839	1194	318	1193	1618	0.998	0.284	0.48
2192	1187	315	1186	1623	1.002	0.281	0.49
AGH4084	1185	315	1186	1631	0.998	0.285	0.51
14FN	1190	316	1173	1640	1.002	0.284	0.54
31AG	1185	315	1187	1631	1.002	0.283	0.51
47M	1194	318	1206	1642	1.005	0.282	0.52
AG3599	1189	317	1167	1604	1.000	0.284	0.51
AGK944	1185	314	1172	1622	0.998	0.284	0.52
VN1AMM0335	1192	315	1196	1621	0.998	0.285	0.48

Table D.14: Experimental Results and Parameters Used for Adiabatic Flame Temperature Calculation of Unserviceable Specimens

Specimen	Calculated Adiabatic Flame Temperature (K)	Average Chamber Inlet Air Temp [K]	Average Exit Gas Temperature (K)	Maximum Exit Gas Temperature (K)	Ambient Pressure (atm)	Average Equivalence Ratio	Pattern Factor
AG3055	1185	317	1193	1619	0.998	0.284	0.51
4298	1187	318	1175	1627	1.002	0.284	0.56
123BQ	1199	319	1192	1653	1.005	0.283	0.53
AG347G	1189	314	1170	1652	1.002	0.284	0.56
48BM	1184	319	1174	1641	0.998	0.284	0.53
94EX	1188	319	1171	1636	1.005	0.283	0.55
AM8FM	1194	318	1184	1635	1.005	0.283	0.52

Figure D.1: Combustion Chamber Exit Temperature Profile – Serviceable 47M

Figure D.2: Combustion Chamber Exit Temperature Profile – Unknown 14FN

Figure D.3: Combustion Chamber Exit Temperature Profile – Serviceable 14FN

Figure D.4: Combustion Chamber Exit Temperature Profile – Unknown 31AG

Figure D.5: Combustion Chamber Exit Temperature Profile – Serviceable 31AG

Figure D.6: Combustion Chamber Exit Temperature Profile – Unknown 83BJ

Figure D.7: Combustion Chamber Exit Temperature Profile – Unknown AG4306

Figure D.8: Combustion Chamber Exit Temperature Profile – Unknown AG4309

Figure D.9: Combustion Chamber Exit Temperature Profile – Unknown AG56522

Figure D.10: Combustion Chamber Exit Temperature Profile – Unknown AG7553

Figure D.11: Combustion Chamber Exit Temperature Profile – Unknown AGL824

Figure D.12: Combustion Chamber Exit Temperature Profile – Unknown AGM291

Figure D.13: Combustion Chamber Exit Temperature Profile – Serviceable AGM291

Figure D.14: Combustion Chamber Exit Temperature Profile – Unknown VN1AMM1301

Figure D.15: Combustion Chamber Exit Temperature Profile – Serviceable
VN1AMM1301

Figure D.16: Combustion Chamber Exit Temperature Profile – Unknown VN1AMM1839

Figure D.17: Combustion Chamber Exit Temperature Profile – Serviceable
VN1AMM1839

Figure D.18: Combustion Chamber Exit Temperature Profile – Unknown VN1AMM0335

Figure D.19: Combustion Chamber Exit Temperature Profile – Serviceable
VN1AMM0335

Figure D.20: Combustion Chamber Exit Temperature Profile – Unknown AGK944

Figure D.21: Combustion Chamber Exit Temperature Profile – Serviceable AGK944

Figure D.22: Combustion Chamber Exit Temperature Profile – Unknown AG3599

Figure D.23: Combustion Chamber Exit Temperature Profile – Serviceable AG3599

Figure D.24: Combustion Chamber Exit Temperature Profile – Unknown 2192

Figure D.25: Combustion Chamber Exit Temperature Profile – Serviceable 2192

Figure D.26: Combustion Chamber Exit Temperature Profile – Unknown AGH4084

Figure D.27: Combustion Chamber Exit Temperature Profile – Serviceable AGH4084

Figure D.28: Combustion Chamber Exit Temperature Profile – Unknown AG3055

Figure D.29: Combustion Chamber Exit Temperature Profile – Unserviceable AG3055

Figure D.30: Combustion Chamber Exit Temperature Profile – Unknown 4298

Figure D.31: Combustion Chamber Exit Temperature Profile – Unserviceable 4298

Figure D.32: Combustion Chamber Exit Temperature Profile – Unknown 48BM

Figure D.33: Combustion Chamber Exit Temperature Profile – Unserviceable 48BM

Figure D.34: Combustion Chamber Exit Temperature Profile – Unknown 94EX

Figure D.35: Combustion Chamber Exit Temperature Profile – Unserviceable 94EX

Figure D.36: Combustion Chamber Exit Temperature Profile – Unknown 123BQ

Figure D.37: Combustion Chamber Exit Temperature Profile – Unserviceable 123BQ

Figure D.38: Combustion Chamber Exit Temperature Profile – Unknown AG347G

Figure D.39: Combustion Chamber Exit Temperature Profile – Unserviceable AG347G

Figure D.40: Combustion Chamber Exit Temperature Profile – Unknown AM8FM

Figure D.41: Combustion Chamber Exit Temperature Profile – Unserviceable AM8FM