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Characterization of a Novel Porous Injector for Multi-Lean Direct Injection (M-LDI) Combustor

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Characterization of a Novel Porous Injector for Multi-Lean Direct Injection (M-LDI) Combustor

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Abstract

A generic novel injector was designed for multi-Lean Direct Injection (M-LDI) combustors. One of the drawbacks of the conventional pressure swirl and prefilming type airblast atomizers is the difficulty of obtaining a uniform liquid sheet under all operating conditions. Micro-channels are needed inside the injector for uniformly distributing the fuel. The problem of non-uniformity is magnified in smaller sized injectors. The non-uniform liquid sheet causes local fuel rich/lean zones leading to higher NO_x emissions. To overcome these problems, a novel fuel injector was designed to improve the fuel delivery by using a porous stainless-steel material with 30 μm porosity. The porous tube also acted as a prefilming surface. Liquid and gaseous fuels can be injected through the injector.

The current study investigates the aerodynamics, spray quality, fuel-air mixing and emission characteristics of the novel injectors at 4% pressure drop and atmospheric conditions. The injectors have two configurations with different counter-rotating radial-radial swirlers. And the injector 1 has a SN of 0.75 and SN of injector 2 is 0.6. The characteristics of the novel injectors are also compared with a typical airblast injector having a peanut nozzle with flow number of 1. A Central Toroidal Recirculation Zone (CTRZ) and Corner Recirculation Zone (CRZ) are observed from the aerodynamics study.

Spray measurements are carried out at various equivalence ratio conditions without a confinement. D_{10} , D_{32} and $D_{0.5}$ are investigated on Jet-A, GTL and blended fuels. There is no significant influence of fuel types on the spray behavior due to their similar physics properties. The porous injectors generate a fine spray with weighted SMD $\sim 45 \mu\text{m}$ at equivalence ratio of 0.6. Gaseous Fuel-air mixing studies are carried out at different equivalence ratios with and without a

confinement. A fully premixed mixing profile was obtained at 0.43” downstream of the injector exit.

Flame characterization and emission measurements are carried out at elevated inlet air temperature and various equivalence ratio conditions. Natural gas is used as the gaseous fuel, Jet-A, GTL and blended fuels are used as liquid fuels during combustion studies. Symmetric blue flames are observed at fuel lean conditions. Injector 1 has a shorter flame than injector 2 at same condition due to the smaller axial velocity. The effects of inlet air temperature and combustion flame temperature on the NO_x emissions are studied. NO_x emissions increase with inlet air temperature and flame temperature for injector 1. For injector 2, NO_x emissions decrease as inlet air temperature is increased from 400°F to 500°F. NO_x emissions increase as the inlet air temperature rises over 500°F.

Chemkin simulations are carried out to investigate the NO_x emissions with various chemical reaction mechanisms. GRI Mech 3.0 is used to simulate natural gas, two reaction mechanisms from NASA and CNRS are used to simulate Jet-A and GTL fuels. A model with a PSR and a PFR is utilized. The comparison between the simulation and experiments is investigated. The simulation results closely match the experimental data under 1900 K flame temperature, and overpredict at higher flame temperature conditions.

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List of Symbols

D	diameter of the injector exit (inch)
X	X axis (inch)
Y	Y axis (inch)
Z	Z axis (inch)
CTRZ	Central Toroidal Recirculation Zone
CRZ	Corner Recirculation Zone
SN	Swirl Number
J_{tg}	momentum of tangential velocity (N.s)
J_{ax}	momentum of axial velocity (N.s)
U_o	upstream mean velocity (m/s)
U_{mean}	mean axial velocity (m/s)
V_{mean}	mean radial velocity (m/s)
W_{mean}	mean tangential velocity (m/s)
U'	rms of axial velocity (m/s)
V'	rms of radial velocity (m/s)
W'	rms of tangential velocity (m/s)
u	local axial velocity (m/s)
w	local tangential velocity (m/s)
r	radius of the injector (inch)
TKE	Turbulent Kinetic Energy (m^2/s^2)
T.I.	Turbulent Intensity
R	radial distance from the center of the injector (inch)

D_{32}	Sauter Mean Diameter (μm)
D_{10}	mean diameter of the spray (μm)
$D_{0.1}$	10% of total liquid volume in drops of smaller diameter (μm)
$D_{0.5}$	50% of total liquid volume in drops of smaller diameter (μm)
$D_{0.9}$	90% of total liquid volume in drops of smaller diameter (μm)
SMD	Sauter Mean Diameter (μm)
q	Rosin-Rammler distribution parameter
$\Delta p/p$	pressure drop (%)
p	static pressure (Kpa)
ϕ	equivalence ratio
PDI	Particle Doppler Interferometer
LDV	Laser Doppler Velocimetry
$\overline{D_{32}}$	weighted SMD (μm)
D_{32i}	local SMD (μm)
V_i	local volume flux (cm/s)
GTL	Gas-To-Liquid fuel
Y_F	fuel mass fraction
$Y_{normalized}$	normalized fuel mass fraction
Y_{local}	local fuel mass fraction
Y_{rms}	rms of the fuel mass fraction
D.U.	Degree of Unmixedness
Y_{mean}	mean fuel mass fraction
Y_i	local fuel mass fraction

X_{global}	global emission concentration (ppm)
X_i	local emission concentration (ppm)
X_{NO_x}	NO _x concentration in the combustion products (ppm)
X_{CO}	CO concentration in the combustion products (ppm)
X_{CO_2}	CO ₂ concentration in the combustion products (ppm)
A_i	local area (inch ²)
T_{flame}	adiabatic flame temperature (K)
T_3	inlet air temperature (°F)
EINO_x	Emission Index of NO _x
MW	Molar Weight (kg/mol)
RQL	Rich burn, Quick quench, Lean burn combustor
LPP	Lean Premixed Prevaporized combustor
LDI	Lean Direct Injection
PSR	Perfectly Stirred Reactor
PFR	Plug Flow Reactor
ROP	Rate of Production (mol/cm ³ ·sec)
$t_{\text{residence}}$	total residence time (ms)
t_{PSR}	residence time in PSR (ms)
t_{PFR}	residence time in PFR (ms)

Chapter 1 Introduction

1.1 Introduction

The development of next generation aero-engines is predominantly driven by their ability to reduce NO_x to sub ppm levels and their ability to operate on wide range of fuels requirements also termed as fuel flexibility. The emissions resulting from aircraft gas turbine engines is attracting attention due to the significant effect of emissions on environment and human health. Although, emissions produced by aircraft engines are not a significant contributor to the air pollution, they are still the reason for the depletion of the ozone layer in the stratosphere. The photochemical smog caused by emissions is the main reason for the decline of the air quality at ground level near the airports. The exhaust produced by gas turbine engines consists of carbon dioxide (CO₂), water vapor (H₂O), carbon monoxide (CO), unburned hydrocarbons (UHC), nitrogen oxides (NO_x) and particulate matter. The greenhouse gases CO₂ and water vapor can only be reduced by burning less fuel and improving combustion efficiency [1]. To control the discharge of CO₂, the International Civil Aviation Organization (ICAO) proposed CO₂ emissions standard for new aircrafts in the 36-State ICAO Council that applies to the new aircraft combustor designs after 2020 [2]. In the modern gas turbine engines, the CO and UHC level are reduced effectively under lower power conditions. However considerable amounts of NO_x are emitted during takeoff, landing and cruise conditions. Table 1.1[3] shows a typical total mass quantity of emissions formed in a modern subsonic aircraft engine. This table indicates that NO_x is the dominant species in both landing-takeoff, and cruise conditions driving the development of next generation aero-engines.

There are several widespread concepts for NO_x reduction in modern gas turbine combustors, such as Rich burn-Quick quench-Lean burn (RQL), Lean Premixed Prevaporized

(LPP) and Lean Direct Injection (LDI) technologies. NO_x is primarily formed due to the high flame temperature and long residence time in the combustor. For example, RQL concept reduces NO_x emissions by avoiding stoichiometric combustion thereby decreasing the flame temperature. RQL concept has already reached its limits considering stringent current emission standards even though RQL concept has good performance in flame stability. LPP concept on the other hand has issues with auto ignition and flashback occurring under high pressure and temperature conditions in the combustors even though it meets the NO_x emission regulations. A novel LDI concept has been drawing attention which is a promising technology for NO_x reduction. This concept mainly lies on the concept of injecting the fuel directly into the combustion zone with rapid premixing of fuel and air in the combustor. This not only mitigates the auto ignition and flashback but also reduces NO_x as the combustor is always operated at fuel lean conditions.

Category	Percent of Total Emission Mass		
	During Landing-Takeoff	During Climbout	Overall
	Cycle	Cruise/ Descent	
Smoke	-	0.1	0.1
HC	0.6	1.0	1.6
CO	5.4	7.0	12.4
NO_x	7.8	78.1	85.9
Total	13.8	86.2	100.0
	(56.5% NO_x)	(90.6% NO_x)	

Table 1.1 Typical Distribution of Total Emission Mass Quantities Generated during a Flight of an Aircraft Equipped with Modern Engines [3]

The LDI concept also faces challenges such as the fuel flexibility and flame stability. In present scenario, gas turbines are expected to be operated on a wide range of fuels. The fuel composition has tremendous effects on flame characteristics such as flame speed, reactivity, Lean Blow Out (LBO) limits and flashback. The LDI concept is sensitive to the fuel flexibility which needs to be considered during the design phase of the LDI injectors. The atomization characteristics and combustion performances furthermore influence the formation of emissions. Flame instabilities and local hot spots are caused by the poor atomization quality and uneven mixing between fuel and air in both spatial and temporal domains. These factors are dependent on atomizer and injector designs. It is apparent from the above discussion that the injector design is decisive for the success of the LDI concept. For a good injector design, the injector should be able to rapidly mix the fuel and air with good spatial and temporal uniformity. To eliminate the mixing non-uniformity issues, several solutions have been proposed. For example, a highly swirling airflow pattern with counter rotating swirl is utilized for fuel atomization and rapid mixing. However, the conventional large injectors are susceptible to manufacturing tolerances that are significant enough to affect the flame stability in the LDI concept. Moreover, it is difficult to achieve rapid mixing with large injectors. The present day LDI technology is focused towards using multiple small injectors instead of one large injector, called MultiPoint-LDI (M-LDI) concept [4]. This design generates multiple small swirling recirculation zones that enhance fuel atomization, rapid mixing, evaporation and produce stable flame zones. The mixing time required for the fuel and air to achieve good uniformity is reduced that in turn reduces the flame length and the residence time of the species in the combustion zone ultimately restraining NO_x formation. Further, in the M-LDI design, the temperature profile at the combustor exit tends to be uniform to prevent the occurrence of local hot spots. As it is evident, good fuel-air mixing performance can

also be achieved by a thorough injector design. As the injectors are made smaller, it is imperative that the performance of the injector relies on the fuel delivery system. Small injectors have problems with uniform spray formation as there is a minimum flow rate required for conventional atomizers to produce a fully developed spray. The widely-employed fuel delivery system in current gas turbine engines are the airblast injectors. These airblast atomizers fall into two categories. In the first category the airblast injectors have a central pressure swirl atomizer which is used for fuel atomization. For second stage of atomization, airblast atomization methodology is used. These are widely popular in the present gas turbines. In the second category, the airblast atomizers operate in fully airblast mode in the sense that the atomization is primarily carried out by airblast atomization. The airblast atomization assures a fine spray formation and a rapid uniform fuel-air mixing. Airblast atomizers are further divided into two categories, non-prefilming and prefilming atomizers. In non-prefilming atomizers, spray is formed by shearing action between a spread liquid fuel sheet and a high speed coflowing air. For prefilming atomizers, a thin liquid sheet is spread over a flat plate (for example on a venturi of an injector) and a high swirling air stream grazes over the liquid sheet to achieve the fuel atomization[5]. For these injectors to perform effectively in the small injectors, the fuel needs to be spread into a thin sheet at all operating conditions. The goal of the present study is to develop an airblast injector for M-LDI combustors. A novel airblast injector is designed to generate a fine spray by using a prefilming surface made with porous material. The fuel is spread over the porous material through a specially designed fuel feed system. A high velocity axial swirling air grazes over the liquid sheet which in turn interacts with two counter rotating swirling air streams downstream of the porous material to achieve the atomization of the fuel, rapid mixing of the spray with air streams, a stable combustion, and low emissions. This injector can be used with a wide range of GTL and blended fuels. The atomization and

emission characteristics are investigated in the present study. This injector is compared with an airblast injector with a central standard pressure swirl atomizer. The fuel flexibility for both atomizers are also examined. The following section discusses the previous works that are related to the present study.

1.2 Literature Survey

This section discusses the pertinent literature that is reviewed during the course of this study. The literature survey is broadly divided into five sub-sections. The section on liquid sheets breakup discusses the atomization process that occurs on the porous tube in the present injector. The section on fuel flexibility discusses the requirements of the gas turbines to be able to operate on a wide range of fuels. The section on emission regulations discusses the reaction mechanisms that drive the formation of NO_x . The section on gas turbine combustor development discusses the present-day developments in Low NO_x combustors and the section on porous material discusses the applications of porous material in the related areas for the reduction of emissions.

1.2.1 Liquid Jets and Sheets Breakup

The atomization of liquid fuels is broadly classified into either the liquid jet breakup or the liquid sheet breakup. The atomizers are designed in either of the configuration depending on the necessity of the application. In gas turbines injectors, both types of atomization processes are adopted and therefore it is essential to understand both atomization mechanisms. There are some research works carried out on the theoretical and experimental aspects of the liquid jet breakup mechanism. Some earlier research on the liquid jet breakup are studied by Rayleigh [6], Rayleigh considered the method of propagation of small disturbances to predict the critical conditions under which a non-viscous laminar liquid jet breaks up in a quiescent environment under the action of gravitational and surface tension forces. It was concluded that the small disturbances on the surface

of the liquid jet grow exponentially when the wavelength of the disturbance is equal to the circumference of the liquid jet. This type of breakup mechanism is called Rayleigh breakup. The average drop size obtained after the breakup is given by 1.89 times of the diameter of the liquid jet. Tyler [7] proposed and improved the theoretical model developed by Rayleigh on liquid jets breakup. Tyler [7] demonstrated that the effects of the viscosity of the liquid and the influence of the surrounding air are critical for the interfacial wavelength leading to liquid jets breakup. Figure 1.1 shows four regimes of liquid jet break up, identified by Haenlein [8].

Figure 1.1 Breakup regimes of liquid jet

- (a) formation of drops without influence of air; (b) formation of drops under the influence of air; (c) formation of drops due to waviness of the jet; (d) complete disintegration on liquid jet.

Figure 1.2 shows photographic investigation of liquid sheet breakup from single hole nozzle, reported by Dombrowski and Franser [9]. Some earlier works on the liquid sheet breakup were carried out by York [10], Hagerty and Shea [11]. They concluded that there are two types of waves (small disturbances) that are possible on the surface of a liquid sheet. Each surface of the liquid sheet can be in-phase to produce sinuous waves or they can be out-of-phase to produce dilatational waves. Both waves are unstable over the same range of frequencies. They predicted the growth of the instabilities on the waves on the surface. Rangel and Sirignano [12][13]. investigated the effects of the surface tension and density of the fluids on the wavelength of the

waves by applying vortex discretion theory. They obtained and discussed the relation between the wavelength and the ligaments spacing.

Figure 1.2 Typical breakup of liquid sheet from single hole nozzle

Figure 1.3 Nonprefilming and prefilming airblast atomizers [14]

In gas turbine combustors, the most widely used airblast atomizers can be divided into two categories, which are the nonprefilming and the prefilming atomizers as shown in

Figure 1.3[14]. The nonprefilming atomizers produce conical and cylindrical liquid sheets. The air streams flow on either one side of the sheet or on both sides of the liquid sheet. The air streams interact with the liquid sheet to achieve atomization. In this study, a prefilming atomizer is designed and investigated experimentally. In prefilming type of atomizer, the liquid sheet is spread on a surface. There is only one air flow stream interacting with the liquid sheet, until a certain distance on the surface. A primary atomization process occurs on the surface. The secondary air streams interact with the liquid sheet downstream of the surface where the further secondary atomization process takes place. The fundamental study focusing on the liquid sheet breakup with coflowing airstreams is not sufficient to explain the atomization process. Berthoumieu et al. [15] studied the liquid sheet breakup characteristics on a prefilming surface. They also discussed the influence of the liquid sheet thickness and the aerodynamics of the injector on the liquid sheet disintegration. Further, they have found the relation between the ligament spacing on the liquid sheet and aerodynamics of the injector before the liquid sheet breakup. There are similar works carried out by Yule et al. [16][17] that is based on the liquid sheet breakup mechanism on a flat sheet. In his study, the effects of the relative velocities between air and fuel on the primary atomization was investigated. The prior studies discussed above at operating conditions conducive for basic studies. In the present study, the airblast injector is operated at pressure drop similar to engine conditions. At these conditions, the atomization behavior of liquid sheet on the prefilming surface is different from observed by Berthoumieu and Yule. Umesh Bhayaraju and Christoph Hassa [18] studied the effect of static pressure on the liquid sheet breakup. They have concluded that at high velocities of air stream and static pressures a phenomenon called surface stripping dominated the liquid sheet breakup mechanism. The liquid from the surface of

the sheet strips off leading to a very fine atomization. They have obtained a correlation for D_{32} with weber number based on fuel flow rates.

While understanding the atomization is crucial for the atomizer design, for practical applications, the details on the quality of the atomization needs to be quantified. For combustion applications, the atomization is often characterized by D_{32} , Sauter Mean Diameter (SMD). SMD determines the amount of surface area available for a given amount of mass of the fuel for evaporation. There are some available literatures focusing on the correlations of the mean droplet size from experiments for the airblast atomizers at practical operating conditions. The influence of many factors on the droplet size were considered and analyzed. Lefebvre [19] demonstrated that the air impingement angle on the liquid sheet has an effect on the length of the liquid sheet breakup. He proposed that at certain impinging angles, a phenomenon called prompt atomization takes place. A correlation of the SMD based on prompt atomization was obtained. It is represented in the following equation.

$$SMD = 3 \left[\frac{1}{t} + \frac{0.007 \rho_L U_A^2}{4\sigma(1 + \frac{1}{AFR})} \right]^{-1} \quad (1.1)$$

where t is the liquid sheet thickness, ρ_L is the liquid density, U_A is air velocity, σ is the surface tension, AFR is air-fuel ratio.

Jasuja [20] summarized a correlation for SMD based on the properties of air and liquid. He summarized the relation between SMD and properties of air and liquid as follows:

$$\frac{SMD}{l_c} \propto \left(\frac{\sigma}{\rho_a u_a^2 l_c} \right)^{0.5} \left(1 + \frac{1}{AFR} \right) \quad (1.2)$$

where l_c is the characteristic length of the prefilming surface. This equation is utilized for prefilming atomizers. There are many correlations of mean droplet size analysis based on various

considerations and atomizer designs [20]. Brandt [21] and Bhayaraju [14] also provided a correlation utilized for the prefilming airblast atomizer, shown in Equation 1.3.

$$\text{SMD} \propto b_k^{0.3} \cdot \frac{\sigma_k^{0.5}}{\rho_a^{0.3} u_a} \quad (1.3)$$

where b_k is the liquid loading (kg/s/m), σ_k is the surface tension of kerosene (N/m), ρ_a is the density of air (kg/m³), u_a is the relative velocity between liquid sheet and air (m/s).

SMD is often used in combustion applications as it represents the evaporate rate of the spray. Besides SMD (D_{32}), there are some other important representative diameters, such as $D_{0.1}$, $D_{0.5}$, $D_{0.9}$. $D_{0.5}$ is also called Mass Median Diameter (MMD). $D_{0.1}$, $D_{0.5}$, $D_{0.9}$ are the drop diameter such that 10%, 50% and 90% of total liquid volume is in drops of smaller diameter. The drop size frequency curve can be assumed as a Rosin-Rammler distribution, as shown in Figure 1.4. The location of each representative diameter is shown on the curve. The relation between each representative diameter and distribution parameter q is shown in Figure 1.5. It is evident that MMD/SMD is always greater than one, which suggests that $D_{0.5}$ is always greater than D_{32} . It also shows that when $q=3.2584$, $\frac{D_{0.999}-D_{0.5}}{D_{0.5}} = 1$. $D_{0.999}$ represents the drop diameter such that 99.9% of total liquid volume is below this diameter. This indicates that $D_{0.5}$ is half of $D_{0.999}$ at this condition [22][23].

The studies carried out until recently are focused on atomization of Jet-A fuel. However, due to the recent demands from the propulsion and power industry, it has become essential that the aviation gas turbines work on a wide range of liquid fuels. It has also become essential that the aero-derivative land-based gas turbines should be able to operate on both liquid and gaseous fuels. The following section discusses some of the issues related to fuel flexibility.

Figure 1.4 Locations of various representative diameters[22]

Figure 1.5 Relation between Rosin-Rammler distribution parameter q and various spray characteristics [22]

1.2.2 Fuel Flexibility

Fuel flexibility is the ability for combustors to burn a wider range of gaseous and/or liquid fuels without sacrificing the output and efficiency of the system. Switching fuels during the operation of the engine should not affect its operation in terms of shutdown, stability, load transitions, and should not affect combustion efficiency during the entire operation [24]. As the worldwide energy structure is going through significant changes along with the evolvement of present economies, fuel flexibility has become an essential factor in the design and development of next generation gas turbines. The demand for gas turbines to operate on a wide range of fuels is arising not only in aviation sector but also for land-based gas turbines. For aero-derivative gas turbines that are used for the power generation further necessity is trending to operate them on both liquid and gaseous fuels. It is apparent that these requirements stem from the fact that the fossil fuels are depleting and the energy security over fuel supply system seems reasonable for the economies to be self-sufficient. In aviation sector, gas turbines operate primarily on liquid fuels. In this scenario, liquid fuels derived from coal and natural gas are great alternatives for Jet fuel.

Until now, companies such as GE and Siemens [25][26] have been endeavoring on addressing fuel flexibility for gas turbines for decades and gained remarkable achievements on burning a wider variety of gaseous and liquid fuels. GE has been spending efforts on fuel flexibility research for over 50 years [25]. The efforts of GE Energy [27] and GE Oil & Gas [28] led to the manifestation of heavy-duty gas turbine combustors to have the capability of operating over a wider range of gaseous as well as liquid fuels. Some of these Dry Lean NO_x (DLN) combustors operate with LNG based fuels. Due to the high composition of ethane and nitrogen in LNG fuels, the gas turbine engines meet with the challenges in combustor operability to achieve the emission requirements.

Figure 1.6 (a) Emission impact of ethane presenting in natural gas on a typical DLN combustor;
 (b) Effect of ethane in natural gas on flame holding design margin for a typical DLN combustor[27]

Figure 1.6 (a) shows the effects of the presence of ethane in the fuel on emissions for a typical DLN combustor. It shows that NO_x increases significantly for 30% ethane doping in natural gas at a given combustor exit temperature. On the contrary, CO emission decreases significantly at a given exit temperature. The architecture of the combustor and cycle conditions determine the degree of the impact. As the composition of non-methane hydrocarbon increases, the flame speed also increases, which is shown in Figure 1.6 (b). It implies that the maximum ethane volume concentration should not exceed 60% in order to have a stable flame which is a necessary design limit. GE Aviation [29] also launched several aero derivative gas turbines for low or medium MWI

(Modified Wobbe Index) gaseous fuels. The combustion tests on DLE and SAC gas turbines with natural gas containing C₂⁺ species have shown that the significant progress in fuel flexibility with the gaseous fuels. Further expansion of fuel flexibility to incorporate liquid fuels is underway.

In this research, apart from Jet-A fuel, Gas-to-liquids (GTL) fuel, a nonstandard liquid fuel, is also used to investigate the atomization and emission performance. GTL liquid fuels are derived from natural gas through Fischer-Tropsch process with partial oxidation of methane followed by water gas shift reaction. The synthetic gas produced through this process is passed over an iron or cobalt catalyst to produce higher hydrocarbon liquid fuels. These GTL fuels are clean, cheap, and easily transportable. It produces low emissions than conventional fuels and the price is cheaper than gasoline and diesel. In gas turbine applications, due to high prices and emissions levels of conventional fuels, the attention has been drawn towards GTL fuels not only for applications in ground transportation but also for gas turbine engines. The chemical and physical properties of GTL fuels are marginally different from conventional Jet fuel, therefore GTL fuels are blended with conventional fuels, such as gasoline, diesel, jet fuels and used in the existing engines.

Many potential fuels can be utilized in gas turbines. This has been demonstrated by engine manufacturers and proved that the gas turbines have the distinctive capabilities of using a wider range of fuels. However, Richard [30] pointed out that there are potential difficulties to achieve the same emission standards with various fuels. He investigated the combustion characteristics on a premixed combustor for gas turbine applications and demonstrated that the various species in the fuel composition can make huge effects on the flame position, auto ignition and flashback occurrence, although the premixed combustor performs very well in reducing NO_x emissions. Lieuwen et al. [31] investigated the influence of the fuel composition (with natural gas and syngas) on the operability of a premixed gas turbine combustor. He indicated that the different

compositions of the fuel mixture have an enormous impact on the combustion behavior. He indicated that the presence of hydrogen in the fuel can make huge effects on flame speed, ignition delay time. This is due to its unique transport and flame speed properties. He also highlighted that the impacts of future fuels on the combustion system operability cannot just be summarized by Wobbe Index, heating value and other traditional design parameters.

Khalil et al. [32] implemented measurements on Colorless Distributed Combustion (CDC) combustor not only on gaseous fuels but also on liquid fuels to test fuel flexibility. The emission performance on different fuels was discussed, NO_x emission can be controlled in ~ 10 ppm level for both gaseous and liquid fuels under relative low equivalence ratio conditions (~ 0.6), although CO emission produced by liquid fuel is much higher than gaseous fuels. This is due to the insufficient available time for mixing and evaporation. It is imperative that one should understand the formation of NO_x under various conditions when the gas turbines are required to operate on a wide range of fuels at the same time they should be efficient and produce low NO_x . The section below gives a brief outline on the formation of NO_x .

1.2.3 Emission Regulations

1.2.3.1 NO_x Formation Mechanism

NO_x formation is controlled by the several factors such as equivalence ratio, inlet air temperature, the type of the fuel employed, and the fuel composition. These factors determine the predominant elementary chemical reactions, which are also referred to as the mechanisms of NO_x formation. In gas turbine engines, there are two major sources of NO formation occurring during combustion process. The first source of NO is generated through nitrogen in air reacting with oxygen (O_2). The second source of NO is the nitrogen contained in the fuel reacting with oxygen in air. These lead to three main mechanisms for NO formation in gas turbine combustors, which

are thermal NO_x mechanism (known as Zeldovich's mechanism), prompt NO mechanism and fuel NO formation mechanism. NO₂, the other constituent of NO_x, is a secondary product. It is formed by oxidation of NO in the combustion zone. The details of each mechanism are discussed in the following sections.

(a) Thermal NO_x Formation (Zeldovich's Mechanism)

The thermal NO_x formation is affected crucially by the temperature of the flame and the concentration of the nitrogen and oxygen. The overall chemical reaction of thermal NO_x formation is an endothermic reaction that can be expressed as follow:

The concentration of NO approaches to a maximum at high flame temperature that occurs close to stoichiometric conditions. However, this one step overall reaction does not explain in detail the formation of NO. NO formation consists of several elementary reactions. First oxygen atoms are produced through the dissociation of oxygen molecules as shown below:

where M is a third body with adequate energy to break the covalent bonds in oxygen molecule [33].

The oxygen atoms react with nitrogen through reactions 1.6 and 1.7. These account for the thermal NO_x formation. Reaction 1.6 is a relatively slow reaction due to the high activation energy necessary to break the strong covalent bonds in nitrogen molecules. In reaction 1.7, NO is produced

through a fast reaction between nitrogen atoms and oxygen. This reaction mechanism was assumed by Zeldovich in 1947 [38].

In fuel rich combustion, reaction 1.7 becomes nondominant due to the low O₂ concentration. Therefore, there is an extended third reaction shown below, which is postulated by Bowman [39]. Reaction 1.8 plays an important role especially near stoichiometric or fuel rich conditions.

Reactions 1.6 to 1.8 are considered as thermal NO formation mechanism, also known as Zeldovich's mechanism. The rate constants for reactions through 1.6 to 1.8 are shown below [34]:

$$k_{+1} = 1.8 \times 10^8 \cdot \exp\left(-\frac{38370}{T}\right) \text{ m}^3 \cdot \text{mol}^{-1} \cdot \text{s}^{-1} \quad (1.9)$$

$$k_{-1} = 3.8 \times 10^7 \cdot \exp\left(-\frac{425}{T}\right) \text{ m}^3 \cdot \text{mol}^{-1} \cdot \text{s}^{-1} \quad (1.10)$$

$$k_{+2} = 1.8 \times 10^4 \cdot \exp\left(-\frac{4680}{T}\right) \text{ m}^3 \cdot \text{mol}^{-1} \cdot \text{s}^{-1} \quad (1.11)$$

$$k_{-2} = 3.8 \times 10^3 \cdot \exp\left(-\frac{20820}{T}\right) \text{ m}^3 \cdot \text{mol}^{-1} \cdot \text{s}^{-1} \quad (1.12)$$

$$k_{+3} = 7.1 \times 10^7 \cdot \exp\left(-\frac{450}{T}\right) \text{ m}^3 \cdot \text{mol}^{-1} \cdot \text{s}^{-1} \quad (1.13)$$

$$k_{-3} = 1.7 \times 10^8 \cdot \exp\left(-\frac{24560}{T}\right) \text{ m}^3 \cdot \text{mol}^{-1} \cdot \text{s}^{-1} \quad (1.14)$$

The forward rate constant of reaction 1.7 is much larger than reverse rate constant for reaction 1.6, thus, reaction 1.7 is a much more rapid when compared with reaction 1.6. Reaction 1.6 is considered as rate-limiting step of the Zeldovich mechanism due to its large activation energy. This results in a high sensitivity to flame temperature and reaction 1.6 can only proceed under high temperature conditions. To have a better understanding of the NO formation rate, the NO formation net rate is investigated. The NO formation rate can be presented as follow:

$$\begin{aligned} \frac{d[NO]}{dt} = & k_{+1}[N_2][O] - k_{-1}[N][NO] + k_{+2}[N][O_2] - k_{-2}[NO][O] + k_{+3}[N][OH] - \\ & k_{-3}[NO][H] \end{aligned} \quad (1.15)$$

Due to its large activation energy and highly sensitive to the temperature, the reaction rate of most reactions in this mechanism are slower than the fuel burning rate. In the other words, these reactions occur after the combustion process is completed, which indicates that these reactions can be decoupled from fuel combustion process. Therefore, the radicals of combustion products such as O, H and OH can be assumed to be reaching equilibrium. The concentration of these radicals can be replaced by the equilibrium values in Equation 1.15. A simplified version of NO formation rate is presented in Equation 1.16 [34][35][36].

$$\frac{d[NO]}{dt} = 2k_1[O]_{eq}[N_2]_{eq} \quad (1.16)$$

The concentration of O at equilibrium state can be calculated from reaction 1.5, as shown in equation 1.17.

$$[O]_{eq} = \frac{k_0}{(RT_{eq})^{1/2}} [O_2]_{eq}^{1/2} \quad (1.17)$$

where T_{eq} is the equilibrium temperature, k_0 is the reaction rate from reaction 1.5. The formation rate of NO can be expressed as follow [4]:

$$\frac{d[NO]}{dt} = 6 \times 10^{16} T_{eq}^{-1/2} \exp[-69090/T_{eq}] \cdot [O_2]_{eq}^{1/2} [N_2]_{eq} \text{ moles/cm}^3 \cdot \text{sec} \quad (1.18)$$

It is evident that NO formation is strongly dependent on the temperature of the combustion gases and the O₂ concentration in the products. When the temperature is higher than 1800K, the thermal NO formation mechanism is dominant in NO generation. Thus, in practical applications, reducing flame temperature and controlling the dilution air in the combustion products can decrease thermal NO formation [36][37].

(b) Prompt NO Mechanism (Fenimore Mechanism)

Apart from the thermal NO mechanism, the other route for NO formation is the reaction of nitrogen with hydrocarbon radicals that are formed in the flame. The hydrocarbon radicals are formed during combustion process. These radicals are typically CH, CH₂, C₂ and so on. This mechanism typically does not exist in flames without CH radicals. For instance, for pure CO or H₂ combustion, CH radicals are not present during combustion, hence prompt NO is not formed. This mechanism is proposed by C. P. Fenimore in 1971 and often termed as Fenimore mechanism. Fenimore [40] conducted the experiments by measuring NO concentration along the axial direction from the burner exit. He concluded that the thermal NO generated in post flame zone is further supplemented by NO formation that occurs at the exit of the burner. The NO at the exit of the burner is formed within an extremely thin flame zone in a very short time, thus termed as “prompt NO mechanism”. As NO appears at the exit of the burner, the conditions are fuel rich, short residence time for the species and low flame temperature. Fenimore extrapolated that the reason for prompt NO formation is that the nitrogen molecules are attacked by the hydrocarbon radicals in the flame. The reactions that are dominant for prompt NO are shown below:

Reaction 1.19 is the dominant reaction and the rate-limiting reaction in this mechanism. Under fuel lean condition, fuel oxidation rate is rapid therefore restrains these reactions to occur. On the other hand, as equivalence ratio increases, the concentrations of the hydrocarbon radicals can reach a critical value, which is sufficient to initiate the above reactions. Under this condition, hydrocarbon radicals mainly attack the nitrogen covalent bonds to generate cyano compounds and amines that ultimately are responsible for significant NO formation.

Turns [35] concluded that the reaction mechanism for cyano compounds and amines oxidation to form NO occur at equivalence ratio less than 1.4. The reactions for prompt NO are shown below:

When equivalence ratio is higher than 1.4, other reactions can be initiated, and the above reactions no longer play a dominant role. Miller and Bowman [41] also summarize that the reaction comes backward to HCN other than forming NO due to the slower reaction rates. The above chain is restrained from forming NO.

It is also assumed that it is necessary to couple the prompt NO mechanism to the actual fuel combustion mechanism due to the uncertainty in rate of the reaction of 1.19 to 1.22, and the significant impact of the fuel-air concentration on NO. However, the actual hydrocarbon fuel combustion reactions are very complicated and include hundreds of elementary reactions. The calculations are costly and have limits. De Soete [42] achieved a method to calculate prompt NO by using a global kinetic parameter in the NO model. He conducted a comparison of the NO formation rate between the experimental data and the values obtained from numerical integration. He gave an overall prompt NO formation rate prediction in fuel rich condition, which is shown in the following equation:

$$\frac{d[\text{NO}]}{dt} = k_{pr} [\text{O}_2]^a [\text{N}_2] [\text{Fuel}] e^{-\frac{E_a}{RT}} \quad (1.25)$$

$$k_{pr} = 1.2 \times 10^7 \left(\frac{RT}{p}\right)^{(a+1)} \quad (1.26)$$

where k_{pr} is a parameter and dependent on the hydrocarbon fuel, R is the gas constant, p is the pressure in Pa, and E_a is activation energy in J/mol and a is the oxygen reaction order, which is dependent on equivalence ratio and other experimental conditions.

Backmier et al. [43] disapproved his prediction by their experiments. They pointed out that the equation 1.24 only supports specific fuel types. Backmier claimed that the experimental results disagree with De Soete's prediction for higher hydrocarbon fuels as equivalence ratio is increased. He optimized Backmier's prediction by introducing a correction factor f , which comprises consideration of different fuel type. The modified equation is:

$$\frac{d[NO]}{dt} = f k'_{pr} [O_2]^a [N_2] [Fuel] e^{-E'_a/RT} \quad (1.27)$$

$$f = 4.75 + 0.0819n - 23.2\phi + 32\phi^2 - 12.2\phi^3 \quad (1.28)$$

where n is the number of carbon atoms in one hydrocarbon fuel molecule, ϕ is equivalence ratio. The experimental results agree with the predicted results based on equation 1.27~28. The reaction rate of NO is corrected for the fuel type, air-fuel ratio, suitable for alkane hydrocarbon fuels and valid for equivalence ratio range from 0.6 to 1.6. For larger range, values of k'_{pr} and E'_a should be reconsidered.

(c) Fuel NO

In the earlier sections, the source of NO is from the nitrogen in air. The other source of nitrogen is from nitric compound contained in the fuel, which is called fuel nitrogen. Fuel nitrogen is present in many type of fuels and is oxidized into NO during combustion process.

Most fuel nitrogen exists in liquid and solid fuels. Natural gas doesn't contain fuel nitrogen. On the other hand, crude oil contains 0.01 to 0.3 % wt. fuel nitrogen, coal contains bound nitrogen up to 2% wt. Most common nitric compounds of fuel nitrogen are pyridine, indoles, chinoline, etc [44]. In combustion process, the fuel nitrogen is mainly HCN, or ammonia (NH₃). The produced

HCN, NH radicals follow prompt NO mechanism to form NO as discussed earlier. Thus, fuel NO formation is also a complicated process involving hundreds of reversible reactions. Many uncertain factors influence the reaction rate coefficients for fuel NO. Figure 1.7 presents the production of NO associated with the Fenimore mechanism. The source of nitrogen consists of nitrogen from air and nitrogen from fuel.

Figure 1.7 The production of NO associated with the Fenimore prompt mechanism [35]

To inhibit the fuel NO formation, the combustion process needs to maintain fuel rich condition and initiate N_2 formation reactions. The most common strategy for minimizing NO from fuel nitrogen is staged combustion. In staged combustion, the fuel is burnt in stages at different fuel-lean and fuel-rich conditions. Various applications utilize staged combustion such as an overfire air, off-stoichiometric combustion, etc. [45] The advantage of the staged combustion is that the thermal NO_x formation is inhibited. Additionally, N_2 formation time is provided thereby giving a better control of fuel NO_x formation.

(d) NO_2 Mechanism

The emission standards require that NO_x be reduced to appreciable levels. NO_x is primarily NO and NO_2 . While NO is the major component of NO_x , NO_2 is still a non-negligible species. In the atmosphere, NO_2 is responsible for acid rain, photochemical smog and has effect on human health. NO_2 is formed in lower temperature regions of the flame. At atmosphere condition nitric

oxide readily oxidize to form nitrogen dioxide. The following reactions are the elementary reactions for NO₂ formation process [35].

where HO₂ radical formation is based on the third body reaction shown below:

Since the condition for formation of HO₂ radicals is at low-temperature, lower than 1000 K, hence NO₂ is formed in the lower temperature regions of the combustor. High temperature conditions (higher than 1500K) can initiate reaction 1.30 and 1.31. NO₂ reacts with H and O radicals to form NO, which indicates that NO₂ escapes from destruction in high-temperature regions of the flame [44]. Thus, NO₂ is formed in the regions with high HO₂ concentration, and where NO molecules are transported by diffusion or turbulence of the flow. While NO_x is the primary emissions that are regulated, CO emissions also needs to be reduced. The presence of CO in the exhaust is a clear indication of poor combustion efficiency and improper design of the fuel injector.

1.2.3.2 Carbon Monoxide

The hydrocarbon combustion process can be simply divided into two steps. The first step is the breakdown of hydrocarbon molecules. During this process carbon monoxide is formed. In the second step carbon monoxide oxidize into carbon dioxide. There are two sources of CO during combustion process. The first source of CO is formed from incomplete combustion due to insufficient oxygen and relatively lower temperature. CO fails to oxidize into CO₂ at these

conditions and remains in the products. The second source of CO is formed under high temperature conditions (higher than 1850K) due to the dissociation of CO₂.

In gas turbine combustion applications, most of the CO is formed during incomplete combustion. Even though under overall fuel lean condition, higher levels of CO are also formed in some regions of the combustor due to poor fuel-air mixing quality. There are three most common reasons responsible for higher CO levels in the combustor. First, insufficient residence time of the reaction causes the slow reaction rates in the primary zone. Second, the poor mixing quality of fuel and air leads to unbalanced fuel concentration regions. Lean regions with weak mixture strength result in unstable flames. On the contrary, fuel rich regions produce local high levels of CO due to insufficient oxygen. Third, interaction of cooling air with products leads to the quenching of the reactions to produce high concentration of CO [1].

The oxidization process of CO relies on the existence of species containing hydrogen such as H₂O and H₂. The presence of hydrogen containing molecules produce hydroxyl radicals (OH) during combustion process that has massive influence on CO oxidation step other than O₂ and O radicals. Thus, the presence of H₂O and H₂ in the flame will increase the oxidation rate and speed up the oxidation reactions of CO. For instance, if H₂O is the main hydrogen containing species, then the following reactions occur during the process of oxidation of CO [46]:

In Reaction 1.33, CO is oxidized by O₂ (not hydrogen containing species) reaction, which is a slow reaction and does not make a significant contribution to the CO oxidization. Reaction

1.35 is the leading step in the chain. Significant formation of CO_2 is produced in this step. Reaction 1.36 produces H radical that reacts with O_2 to form OH and O radicals. These radicals are the species to propagate CO oxidization in reaction 1.35 [35]. For reactions that involve Hydrogen, the whole H_2 air reaction mechanism needs to be taken into consideration. The following reactions need to be included:

Glassman [46] pointed out that the reaction mechanism with HO_2 exists for the oxidation of CO as shown below:

There are several CO reduction strategies that are applied nowadays. For example, increasing the residence time by increasing the volume in the primary zone of the combustor or reducing the amount of the cooling air in the primary zone reduces CO formation. In the case of liquid fuels, improving the fuel atomization helps in reducing CO [1].

1.2.3.3 Unburned Hydrocarbon

Unburned hydrocarbons (UHC) is another indicator of incomplete combustion. At normal engine operation conditions, the UHC concentration is low as the combustor is operated at relatively low equivalence ratio where the oxidation rates are relatively fast. However, like CO formation, hydrocarbon can undergo destruction even with poor mixing quality of the fuel and air as shown in Figure 1.8. Local fuel-rich spots arise due to poor fuel-air mixing and hydrocarbon molecules polymerize rather than oxidize due to the insufficient oxidants. UHC are toxic and contain many uncertain carcinogens and mutagens. The type of fuel burnt, and its oxidation limits determine the amount and composition of UHC. Non-combustible species in fuel and lubricants

Figure 1.8 Influence of fuel atomization on UHC emissions [1]

contribute to the UHC emission significantly. There are a lot of difficulties to investigate the hydrocarbon oxidation reaction mechanism of a fuel in detail due to the broad range and complexities of the hydrocarbons and the studies are still under development.

The previous section discussed about the fuel flexibility related to gas turbines, the emissions associated with burning of liquid and gaseous fuels and the fundamental mechanism of liquid fuels atomization. These sections are chosen as the present study is focused on development of a novel injector for application in aero-derivative gas turbines that run on both liquid and gaseous fuels. It is therefore also essential to understand the development of gas turbine combustor technology in such applications. The following section discusses some of the combustor technologies adopted in the aviation and the power industries, and the progress that is made towards LDI technology.

1.2.4 Combustor Technologies

Modern gas turbine combustor technologies incline towards Dry Low NO_x (DLN) combustion rather than reducing NO_x with water injection. DLN combustors have tremendous potential for NO_x reduction capabilities and have shown considerable development opportunities. When it comes to aviation gas turbines, the trend is towards lean direct injection technologies. The most popular modern gas turbine combustor technologies are Rich burn-Quick quench-Lean burn (RQL) combustor, Lean Premix Prevaporized (LPP) combustor, and Lean Direct Injection (LDI) combustor.

1.2.4.1 RQL (Rich burn-Quick quench- Lean burn) Combustor

Figure 1.9 illustrates the standard concept of RQL combustor. As shown in the figure, the RQL combustor consists of three main zones. The combustion in the primary zone is fuel rich as indicated by “rich burn” on the right side of the graph. The NO_x formation rates can be restrained in this zone due to the relative low flame temperature and insufficient oxygen content. As the combustion products flow downstream, they encounter the high-speed cooling air jets in the second zone, where the operating conditions are near stoichiometric condition as shown by “high NO_x route”. This gives rise to the increase in the gas temperature and the oxygen concentrations, an optimum condition for thermal NO_x formation producing high NO_x in this zone. The third zone is the dilution zone where the further gradual dilution air comes is fed into the combustor. In this zone, the equivalence ratio is decreased towards the leaner condition as indicated by “lean burn”. At lean burn conditions, the flame temperatures are below 1800 K thereby inhibiting the formation of NO_x .

In the primary zone of the practical combustors, the equivalence ratio is typically between 1.2 and 1.6 [1]. Although a higher equivalence ratio above 1.6 favors low NO_x formation, the

combustion products contain excessive CO, soot, and smoke emissions. RQL combustor also requires fine atomization of the fuel and a good quality of fuel-air mixing to ensure that the efficiency and performance of the combustors are good. Air is introduced into the primary zone through an injector with a high swirl to enhance the fuel-air mixing quality. A central recirculation zone is produced downstream of the injector for a stable flame. This design can cut down the NO_x formation [47]. The transition zone from fuel rich to fuel lean zone is a fast process and the main source of NO_x formation. The residence time in this zone where the stoichiometric combustion occurs is minimized by optimizing the aerodynamics of the combustor in this zone. The temperature in the dilution zone is managed carefully as it has effects on emissions formation. A high temperature in this zone can cause large amounts of NO_x formation, whereas if the temperature is low, CO, UHC and soot fail to completely oxidize and remain in the combustion products. Thus, the selection of the equivalence ratio in dilution zone is of decisive importance to meet emission regulations. The selected equivalence ratio in most lean burn combustion zone of RQL combustor is around 0.5~0.7 [48].

Figure 1.9 Principle of Rich burn-Quick quench-Lean burn (RQL) combustor

RQL combustor is being widely utilized in the current aircraft applications [49]. Nataka et al. [50][51] carried out emission analysis on a RQL combustor design with low heating value (LHV) gaseous fuels and achieved desirable NO_x emission results. The advantage of the RQL combustor is that it can generate a stable flame and achieve better LBO limits. However, the applications where liquid fuel is employed, it is still limited by the excessive formation of UHC and soot in the fuel rich zone. Improvements of quick quench mixing process are necessary. RQL technology has reached its limits in reducing NO_x emissions. The modern gas turbine combustors are shifting towards fuel lean combustor technologies.

1.2.4.2 Lean Premixed Prevaporized (LPP) Combustor

The deficiencies of RQL concept has led to the development of a series of lean combustion concepts. In the lean combustion concepts, the fuel is burnt lean downstream of the injector exit. One of the initial combustor designs employed is Lean Premixed Prevaporized (LPP) concept. In the LPP combustor, a homogeneous fuel-air mixture is burnt at low equivalence ratio conditions in the combustor to reduce NO_x formation. A typical LPP combustor comprises of three major regions. The first region consists of an LPP module that is designed specifically to evaporate the liquid fuel and premix it with air before it enters the combustor. In the second region, the premixed fuel-air mixture enters the combustor where it is burnt. The flame can be operated at low equivalence ratio slightly above the lean blowout limits. Recirculation zones are generated by the airflow pattern to ensure the flame stability. The temperature is distributed uniformly at the combustor exit. Typically, the gas temperature is lower than 1850 K which inhibits the thermal NO_x formation. If the temperature is lower than 1900 K, the NO_x formation does not depend on the residence time anymore [52][53]. By maintaining lower temperatures in the combustor, CO, UHC and soot be consumed completely by increasing the residence time in the combustion zone

[53]. The third region is the conventional dilution region where the cooling air enters the combustor and mix with the exhaust gases. This decreases the gas temperature and meet the turbine requirements. Posechl et al. [54] reported that even with the flame temperature higher than 2000 K, the NO_x emission can be reduced to lower than 10 ppmvd in LPP combustor. One of the main problems with LPP combustors is the autoignition of the fuel-air mixture in the LPP module. Further, the LLP combustors are prone to flashback. Since the LPP modules are designed for the fuel to fully evaporate in the module, the length of the module is long enough to provide enough residence time. The high inlet temperature at high power engine operation conditions and the long residence time in the module are often sufficient for autoignition to occur. The application of LPP concept is extremely restricted due to these drawbacks.

1.2.4.3 Lean Direct Injection (LDI) Combustor

To overcome the disadvantages in the LPP concept, Lean Direct Injection (LDI) technology has been developed for gas turbine applications. As it implies, in the LDI concept the air and fuel are directly injected into the combustion zone separately through an LDI injector. The fuel and air are mixed rapidly slightly downstream of the injector. The injectors are designed to achieve a high quality of fuel-air mixing performance in a short time. Similar to LPP concept, it operates at low equivalence ratio where the temperature is low for restraining NO_x formation. The short flame length in LDI combustor decreases the residence time of the species in the flame zone thereby reducing the NO_x formation.

The advantage of LDI concept is that no premixing process is involved like in LPP module. The mixing happens in the combustor. This mitigates the risk of the occurrence of the autoignition and flashback. The other advantage is that the concentration of CO, UHC and soot in the products can be diminished due to the low equivalence ratio operating condition. The LDI concept also

carries some disadvantages. As described earlier, LDI design needs to achieve the good fuel-air mixing quality. Even though the overall operating condition is fuel lean, poor fuel-air mixing performance from the injector can cause local fuel rich regions, non-uniform distributions of temperature and local hot spots, which ultimately lead to high NO_x formation. To eliminate these problems, a couple of solutions have been proposed for the injector design. For example, a highly swirling airflow is utilized for fuel atomization and one large injector is split into several smaller size injectors. This LDI technology with multiple small injectors is referred to as Multipoint-LDI (M-LDI) concept [69]. This design generates multiple swirling recirculation zones to enhance fuel atomization. The fuel droplets are rapidly dispersed into the air and evaporated. Also, the mixing time required for the fuel and air to achieve a good degree of mixedness is cut down extremely. This reduces the overall residence time of the species and restrains NO_x formation. The temperature profile at the combustor exit tends to be consistent and prevent the local hot spots occurring. The injector design is crucial in this technology as it should be able to handle small flow rates of fuel at partial loads.

NASA started the development of LDI concept in early 1990s. In United States, many researchers and research institutions have directed their interests towards the LDI technology due to its low NO_x emission capabilities under practical engine operating conditions [55]. There is some early LDI research employing both liquid and gaseous fuel. Anderson [56] designed a lean combustor with diesel fuel for automotive gas turbine applications. In this design neither an ignitor nor flame holder is used due to the high inlet air temperature (1100 to 1250 K). He investigated the effects of the inlet air temperature and the reference velocity on NO_x emissions. The reduction of NO_x is caused by the increase of inlet air temperature as well as the reference velocity due to the improved vaporization. Alkabie [57][58] et al. and Hussain [59] conducted NO_x emission

measurements on natural gas and propane with radial swirlers. Alkatie [57][58] investigated the effects of the swirler configuration on NO_x emissions, Hussain [59] utilized the radial and axial multi jet combustion design to achieve rapid fuel-air mixings and stable flames. They both demonstrated that the propane flame produced approximately the double NO_x than natural gas [57][59]. Tacina et al. [60] compared the data of LDI concept with the data of LPP concept of Anderson [61] and Semerjian [62], the results are shown in Figure 1.10. The comparison is based on gaseous propane. In this graph, the LDI data from Anderson [56] employed liquid diesel as fuel. Figure 1.10 illustrates that the Hussain's data agrees with Anderson's LPP data [61] very well under lower adiabatic combustion temperature conditions. Shaffar [63] and Karim et. al. [64] also confirmed that NO_x produced by LDI can approach to LPP level. However, Alkatie's design produced more NO_x emissions than Hussain's. From the LDI data of Anderson [56], liquid fuel produced more NO_x than gaseous fuel. The effects of the atomization and vaporization of liquid fuels on NO_x emissions is evident from these experiments.

As discussed earlier, multipoint LDI (M-LDI) concept can enhance the fuel-air mixing and mitigate NO_x emissions. Many researchers have been working on multipoint injection applications. Yoshimura et al. [65] indicated that multipoint technology not only has the capacity of NO_x reduction but also prevents flashbacks, hence hydrogen addition can be done in this concept. This is also confirmed from other research studies such as York et al. [66], Asai [67] and Marek et al. [68].

Figure 1.10 Direct injection Low NO_x combustion data and comparison with LPP data [60]

Figure 1.11 Comparison of NO_x emission of LDI with LPP [69]

Tacina et al. [69][70][71] implemented multipoint LDI measurements with various swirler configurations containing multiple fuel injectors. They tested JP8 and Jet-A fuels at inlet air temperature up to 810K and at combustor pressure up to 2760kPa. They obtained a correlation between Emission Index of NO_x (EINO_x) and parameters such as inlet air temperature, pressure, equivalence ratio and pressure drop. The results show that the NO_x emissions increase with inlet air temperature, pressure, equivalence ratio, and decrease with pressure drop. They also investigated the effect of swirler direction and the number of injectors on NO_x emissions. Figure 1.11 illustrates the comparison of the results of LDI EINO_x with LPP concept [69]. Various configurations of LDI combustors and LPP combustor models were investigated. The figure indicates that the NO_x emissions from LDI combustors are generally higher than LPP concept, nevertheless LDI results can approach to LPP results with specific configurations at flame temperature of 1900~2000 K. Furthermore, the injectors with stronger swirl intensities can form more NO_x emissions. This is because, at a given pressure drop, if the flow has a strong swirl, then the axial velocity is weaker leading to longer residence time and therefore more produces more NO_x. It is also concluded that LDI concept has a potential for NO_x emissions reduction by approximately 70% of 1996 ICAO standards [72].

NASA Glenn Research Center [73][74] also conducted multiple tests on LDI concept at pressure up to 5500 kPa in their ASCR facility. Two correlations for NO_x are obtained as shown in equation 1.40 and 1.41. These are obtained from experimental data from ASCR facility and from tests carried out in Advanced Subsonic Technology Program. The correlations are given below:

$$EINO_x = 0.104 \exp\left(\frac{T_3}{185}\right) FAR^{1.32} p_3^{0.68} (\Delta p/p_3)^{-0.36} \quad (1.40)$$

$$EINO_x = 1.359 \exp\left(\frac{T_3}{194}\right) FAR^{1.69} p_3^{0.595} (\Delta p/p_3)^{-0.565} \quad (1.41)$$

The correlation of $EINO_x$ is given as a function of inlet air temperature, equivalence ratio, inlet air pressure and pressure drop. These two correlations are almost identical except for the powers of the parameters [75].

Cai et al. [76] carried out tests to investigate the spray characteristics of non-reacting and reacting flows at atmospheric conditions on a single injector. The spray performance with strong swirl is analyzed. The effects of the swirl on the droplets size and velocity profiles are also investigated. It is concluded that the helicoidal vanes can improve the fuel atomization to generate a uniform droplet size. It is also demonstrated that the flow turbulent velocities increase with the heat release during the combustion process. Tedder et al. [77] investigated the effects of the swirler vane angle and the injector location on the flow recirculation zone formation and the spray behavior for a single point LDI concept. It is concluded that the swirler vane angle of 60° can generate recirculation zones, and the injector tip when located close to the venturi throat can improve the atomization quality and angular distribution.

There are also numerical studies on LDI concept. Liu et al. [78] examined the two-phase reacting flow for a LDI concept with Jet-A fuel by using National Combustion Code (NCC) that is developed by NASA Glenn Research Center. Several approaches are discussed, including steady Reynolds-Averaged Navier-Stokes (RANS), unsteady RANS and the dynamic flow structure simulation (DFS). The reaction model is selected as a five-species single step global reaction mechanism. Spray performances in the reacting flow from different models were validated and compared in their study.

Ajmani et al. [79] also carried out numerical studies on next generation LDI-2 design combustors with NCC code. Two pressure drop conditions of 4% and 5% at medium and high power operating conditions were selected for simulations. The $EINO_x$ and the combustor exit

temperature were computed and compared with the experimental results. The comparison is shown in the following table. Case A1 and A2 are at 4% and 5% pressure drop at medium power condition and Case B1 and B2 are at high power condition. T_4 represents the adiabatic temperature, the computed combustor exit temperature matches with the adiabatic temperature well within 5% error. The calculated $EINO_x$ from NCC are higher than the $EINO_x$ from experimental results except for Case A1, the accuracy of NCC is around 70% when compared with experimental data. The accuracy can be improved by using the optimized chemical reaction mechanism.

Case	T (inlet)	FAR	ΔP	T_4 (outlet)	$EINO_x$	$EINO_x$
	(K)	(Expt)	%	(NCC, K)	(Expt)	(NCC)
Case A1	811	0.0214	4	1535	2.86	2.0
Case A2	811	0.0214	5	1554	2.02	2.2
Case B1	905	0.0259	4	1716	3.65	4.2
Case B2	905	0.0259	5	1755	3.47	4.1

Table 1.2 Summary of NCC predictions for two power conditions with two different pressure drops at $P_3=896\text{kPa}$ [79]

In the present study, though the injector design is generic and has a wide range of applications, its main purpose is for the applications where the required pressure drop across the atomizer is low. In this study it is used for M-LDI concept. As discussed in the previous sections, the role of the injector is to provide a good atomization of the fuel and a good fuel-air mixing in the shortest length scales and residence time inside the combustor. Especially for M-LDI concept, the injector should be able to operate at low fuel flow rates at idle operating conditions. To achieve this, a novel method of fuel injection and atomization process is adopted in this study. Fuel is injected through a porous material and the atomization is predominantly carried out by shear

stripping phenomena as discussed in the previous sections. The following section provides a literature on the application of porous material for injector and burner developments in the allied fields.

1.2.5 Porous Material Applications

Porous material is widely utilized for filtration and lubrication applications. There are burners and injectors developed for injecting both liquid and gaseous fuels. The advantage of the porous material is that it can be used for simultaneous injection of both liquid and gaseous fuels. For liquid fuel applications, a liquid prefilming sheet is produced along the porous material surface for atomization whereas for gaseous fuel it behaves as a shear layer injection.

Typically for combustion applications, the porosity can be chosen from 30 μm to 100 μm . There are studies in direct injection diesel engines where a micro porous injector nozzle was developed. Dodge et al. [80] discussed the effects of the number of hole and holes size of a fuel injector on the combustion characteristics in a diesel engine. They indicated that the soot emission is reduced by injecting fuel through smaller injection holes. Reijnders et al. [81] investigated the spray performance by injecting diesel fuel through a porous injector. The injector configuration used is shown in Figure 1.12. A round shape fuel injector is developed for liquid fuel injection. The spray shapes of this injector and the conventional nozzle are showing in Figure 1.13 (a) and (b), respectively. The spray quality in Figure 1.13 (a) is fine, although the hemispherical distribution is not perfectly uniform. In Figure 1.13 (b), the amount of the entrained air is restricted effecting the fuel-air mixing. The mass of the fuel penetrating through the porous material is much higher than the conventional injector. Some issues of the porous material such as pressure resistance and permeability effects on atomization are still under research.

Figure 1.12 Close-up of porous injector tip

(a)

(b)

Figure 1.13 (a) Measured spray of the porous injector ; (b) Measured spray of a conventional injector

1.3 Scope and Objective

The objective of this dissertation is to design and develop a novel gas turbine injector for LDI concept and investigate the characteristics of this injector in terms of its aerodynamics, atomization, fuel-air mixing, combustion and emission performance. The porous material is utilized as the contact media between fuel and air to achieve good quality of the fuel-air mixing performance for both liquid and gaseous fuels. The characteristic of this injector is that liquid fuel atomization is realized by the impacts of the high-speed swirling flow on the prefilming surface.

As part of the injector development program, several injector designs are considered with different swirl configurations. Different fuel feed systems are developed to optimize the fuel distribution inside the fuel plenum. A comparison is made between the injector developed in this program with a conventional pressure swirl injector. The research work is primarily experimental in nature and focused on both non-reacting and reacting experiments at atmospheric conditions with preheated air temperature and at pressure drop across the injector that are similar to engine conditions. The studies are carried out for various fuel types. Natural gas is used as gaseous fuel and Jet-A fuel, 100% GTL fuel and blended fuel (50% Jet-A fuel and 50% GTL fuel) are used as liquid fuels in the present study. The effects of the inlet air temperature and the flame temperature on NO_x emission are quantified. Detailed comparisons are made between different configuration of swirlers, different fuel feeding system and various fuel types, respectively. An effort has been made for the numerical analysis of NO_x emissions for different fuels and compared with the experimental results.

Outline of the thesis

The present chapter highlights the ongoing research in the field of the liquid atomization, trends on fuel flexibility, emission regulations and gas turbine combustor technology developments. Chapter 2 describes the experimental facilities used and measurement techniques adopted in the present study. The results are presented in two chapters. Chapter 3 describes the results obtained in the non-reacting experiments that include atomization of liquid fuels (Jet-A fuel, GTL and blended fuels), aerodynamic measurements of the injector, and fuel-air mixing studies on gaseous fuels. Detailed analysis of two-phase flow and quantification of fuel-air mixing is presented in chapter 3. Chapter 4 presents the combustion characterization of the injector for the fuels tested. The emissions are quantified and a correlation for EINO_x is obtained based on the

experimental data at atmospheric conditions. Furthermore, an effort on numerical simulation on different fuels is presented. A detailed study of the reaction mechanisms to simulate the fuels is discussed. Chapter 5 finally summarizes the present study on the porous injector and outlines the recommendations for future work on this subject.

Chapter 2 Experimental Setup

An experimental approach is adopted in the present study to characterize the dual fuel injector with Porous Injection Technology (PIT). This chapter presents a brief description of the test facility and the experimental techniques used during the characterization. The chapter is broadly classified into two sections: injector design and experimental test setup. The section on injector design details the different designs used and the methodology adopted during the design phase. The experimental test setup section describes the test setup and the experimental techniques that are used during this study. Laser Doppler Velocimetry (LDV) was used in non-reacting aerodynamic measurements to measure the three velocity components and corresponding RMS velocities of the flow field. Particle Doppler Interferometer (PDI) was used to measure the droplet size, mean velocity of the spray and volume flux distribution of the spray in the combustor. Fuel-air mixing studies were carried out to investigate mixing performance of the injector. In reacting flow studies, flame visualization was carried out with a chemiluminescence filter and images were captured with a CCD camera and emissions were measured with an integrated emission analyzer system for both gaseous and liquid fuels.

2.1 Preliminary Injector Design

The preliminary injector design is shown in Figure 2.1. The porous tube is shown in red color. The preliminary design has straight flow through the porous tube for atomization. The fuel is injected into the fuel plenum from the right side. The preliminary design can cause an uneven distribution of the fuel around the porous tube. The finalized and modified injector design is introduced in the next section.

Figure 2.1 Cross section view of the preliminary injector design

The results of the atomization of the initial injector are shown in Figure 2.2. Patternation tests were carried out at $0.5D$ downstream of the injector at 4% pressure drop and equivalence ratio of 0.6. It is evident that the spray is not symmetric. The contour plot of D_{10} in Figure 2.2 (c) shows a non-uniform distribution of mean diameter with higher droplet diameters on the outer periphery of the spray. Further the mean diameter is higher on the right side. This is due to the higher volume flux on the right side of the spray, the distribution of the volume flux is shown in Figure 2.2(b). As can be observed the volume flux is higher on the right side of the spray. The reason for higher volume flux is uneven distribution of the fuel around the porous tube. Further a concentric ring of low volume flux around the higher volume flux region is observed which is not observed in the patternator data. Figure 2.2 (d) shows the distribution of SMD (Sauter Mean Diameter) D_{32} . Like D_{10} , the D_{32} shows a non-uniform distribution in the spray with marginally higher diameter on the right side of the spray. Also, higher droplet diameters are observed on the outer periphery of the spray which is expected. Based on these observations, it is concluded that the fuel distribution is not uniform inside the fuel plenum and the fuel plenum needs to be redesigned. Also, the hollow cone spray could not be achieved as there is no force acting on the liquid sheet that can make the liquid sheet adhere to the porous tube. The lack of force acting on the liquid

sheet allows the liquid to be sucked into the air stream that is flowing through the tube effectively creating a solid cone spray which is not desired in this application. It was felt that an axial swirler is needed at the entrance of the porous tube. Further a need for central blockage is felt so that there is no recirculation created inside the porous tube due to the axial swirl. Final modifications were carried out on the injector design. The design of the new set of injectors are described in the next section.

Figure 2.2 The results of the atomization in the preliminary injector

2.2 Injector Design

In the present study, three injectors were designed. Figure 2.3 shows the 3D model of the injector. Figure 2.4 shows the cross-sectional view of the three injectors designed and tested in this study. The first two injectors shown in Figure 2.4(a) and (b) are injectors with airblast atomizers. The fuel is fed through a porous wall with 30 μm porosity. The third injector in Figure 2.4(c) is an injector with a pressure swirl atomizer. The three injectors are made in two sections. As shown in the figure, all injectors consist of a bottom section with fuel feed system for the atomizer and a top section consists of a counter-rotating radial-radial swirler. Injector 1 and 2 have the same prefilming airblast atomizer with PIT, and Injector 2 and 3 have the same top counter-rotating radial-radial swirler. All injectors are 3D printed in cobalt chrome.

Figure 2.3 The 3D view of the injector

(a)

(b)

(c)

Figure 2.4 Cross section view of the three injectors(a) Injector 1 (b) Injector 2 (c) Injector 3

2.2.1 Fuel Plenum Design

The fuel plenum used in injector 1 and 2 is designed to ensure that the fuel distribution is symmetric. As shown in Figure 2.4(a), the porous tube shown in red color sits on an axial swirler with a central blockage. The axial swirler has 4 vanes, the vane angle is 60° , the vane thickness is 0.14". The diameter of the central blockage is 0.15", and the length is 0.5". The axial swirler and central blockage are used to impart swirl to the air inside the porous tube. It also ensures that the liquid sheet is attached to the porous tube. Further, the blockage ensures that there is no recirculation zone inside the atomizer.

Figure 2.5 Cross section view of the fuel channel

The cross-sectional view of the fuel plenum along a horizontal plane is shown in Figure 2.5. The arrows indicate the direction of the fuel flow. The orange color indicates the fuel flow, the middle black circle represents the outer surface of the porous tube. There is only one fuel injection port, as shown in this figure. The fuel enters the fuel port and splits into two channels. As it meets the barrier it again splits into four and again into eight channels that surround the porous tube. The objective of this design is to ensure that the fuel distribution around the porous tube is

symmetric. The depth of the fuel channel is chosen as 0.08". The depth of the channels is chosen based on the fuel flow rate requirements at the minimum operating condition. This ensures that the channels are always filled with fuel and are under positive pressure. The corners are also curved so that the flow turns smoothly without creating air pockets.

CFD simulations were carried out on the fuel plenum to investigate the velocity and pressure distribution inside the channels and around the porous tube. ANSYS FLUENT version 16 is used for the simulations. The channel walls are set at no slip boundary condition. The inlet conditions at the fuel port is set as constant mass flow rate of 2 pph, the direction of the inlet velocity of the fuel is set as normal to the inlet surface. To simulate the porous tube, the outlet is created with 70 equal size holes distributed uniformly around the outlet surface. The size of the holes is selected as 0.06 in^2 . The selected number and size of the holes are selected such that it is equal to the open area of the porous tube (~50% of the total surface area). PRESTO! discretization is used for pressure and the second order upwind is used for momentum and energy. The fluid flow is simulated with kerosene, and the material is set as steel. Heat losses are not considered. In the experiments, the fluid used is Jet fuel. As the physical properties of kerosene and Jet-A fuel are similar, it is assumed that it will not affect the simulation results.

Figure 2.6(a) shows the velocity distribution in the horizontal plane in the fuel plenum. In this figure, the inlet is at the bottom of the figure, and outlet of the flow is the circumference of the inner circle at the center of the geometry. As the fuel is flowing towards the porous material (center of the plenum), the mean velocity decreases. This is due to the total pressure losses in the channels. The velocity of the fuel increases after the flow splits into two branches, which is initially due to the reduction of the cross-sectional area of the channels. There are few stagnation regions observed in the inner circle of the fuel channel. This is due to the back pressure generated at the outlet. The

velocity distribution is uniform at the outlet which indicates that the fuel emanates uniformly from the porous tube along its circumference. To investigate whether the fuel emanates uniformly around the porous tube, the static pressure distribution of the flow at the outlet is also analyzed.

Figure 2.6 (b) shows the pressure distribution in the fuel channels. It is clear that the pressure decreases as the fuel flows towards the outlet of the geometry. The static pressure is near zero in the inner channels of the fuel plenum. As discussed above, there are pressure losses in the fuel channels. The pressure at the outlet reaches the atmospheric pressure. The porous tube causes least resistance. The pressure distribution is uniform at the outlet, same as the velocity profile of the flow. This shows the fuel channel inside the plenum can provide a uniform liquid sheet at the outlet of the porous tube which is essential to generate a uniform spray.

(a)

(b)

Figure 2.6 Velocity and pressure distribution profile inside fuel channel from simulation

(a) Velocity profile (b) Pressure profile

Although the simulation shows a uniform distribution, the asymmetric and non-uniform spray can also be caused by the uneven porosity of the porous material. The porosity of the porous tubes should be homogenous to achieve a uniform spray. The porous material used in the present studies is made of SS 316L, with porosity of 30 microns. This porosity is the average hole size of in the material. Figure 2.7 shows the concept of the permeability of the porous material [81].

(a)

(b)

(c)

Figure 2.7 (a) Not porous, not permeable; (b) Slightly porous, not permeable; (c) Highly porous, slightly permeable [81]

For injector 3, a conventional pressure swirl atomizer is used to generate the spray. This injector is used as a baseline for comparison of injectors with the PIT and the baseline injector. The flow number (FN) of the pressure swirl atomizer is chosen as 1 as the fuel flow rate are very low in the present case. The pressure swirl atomizer is inserted into the center of the radial-radial swirler. Figure 2.8 shows a schematic of the spray cone for these two depths. Based on the previous work [82] on injector 3, it was evident that when fuel was injected with deep insertion depth, the fuel impinges on the venturi causing a prefilming type of atomization. However, in the case of shallow insertion depth, only partial prefilming of fuel occurs. Figure 2.8 shows a schematic of the effect of insertion depth on the spray penetration. Therefore, in the present study, the deep insertion depth injector was chosen as a baseline injector. The depth from nozzle tip to venturi tip is 0.43".

Figure 2.8 Effect of nozzle insertion depth on spray penetration [82]

2.2.2 Counter-Rotating Radial-Radial Swirler

As discussed earlier, the injector is composed of two sections. The top section of the injector is a counter-rotating radial-radial swirler. It consists of a primary inner swirler and a secondary swirler as seen from the downstream towards the injector. The primary swirler has counter-clockwise rotating air passage and the secondary swirler has a clockwise rotating air passage, and they are separated by the venturi.

For Injector 1, the primary and the secondary swirler have 11 and 20 straight vanes, and vane angles of 58.2° and 70°, respectively. The diverging angle of flare is 71.8°. The thickness of the venturi tip is 0.023". The diameter of the exit of the injector is 1". The height of the flare is 0.28". The vane loading parameter (VLP) is also calculated to estimate the aerodynamic performance of the injectors. The equation is shown in Equation 2.1,

$$VLP = \frac{2B_p(\sin\phi_v)}{R_o - R_i} < 1 \quad (2.1)$$

where B_p is the vane to vane throttling gap, Φ_v is the swirl vane angle, R_o is the outer radius of the swirl vanes, and R_i is the inner radius of the swirl vanes. The calculated VLP for the primary and the secondary swirler are 0.67 and 0.69, respectively, which are smaller than 1, and indicates good aerodynamic properties. The details of the swirler of Injector 2 and 3 are listed in Table 2.1.

The effective area of the three injectors are calculated at 4% pressure drop across the three injectors at atmospheric condition. The calculation is based on the measured air flow rate. In this study, all measurements are carried out at 4% pressure drop across the injector. The effective area of Injector 2 is larger than Injector 3 by 0.0175 in², due to the axial swirler at the bottom of the fuel plenum. The effective area is estimated through the following Equation 2.2.

$$\dot{m}_a = A_{eff} \sqrt{2\rho\Delta P} \quad (2.2)$$

The pressure drop across the injector is measured by the differential pressure transducer, and the air mass flow rate is measured by CMF010 micro motion Coriolis flow meter. The calculated effective areas of these injectors are shown in Table 2.1.

Injector parameter	Injector 1	Injector 2	Injector 3
Primary swirl vane angle (P)	58.2°	58.2°	58.2°
Secondary swirl vane angle (S)	70°	70°	70°

Axial swirl vane angle (A)	60°	60°	-
Height of the secondary swirler (in)	0.08	0.16	0.16
Air flow rate split $M_{R-R}:M_A$	80:20	88:12	-
Air mass flow rate at 4% pressure drop (pph)	50	82	70
Porous tube edge angle	60°	60°	-
Venturi angle	10°	5°	5°
Effective area (in ²)	0.086	0.14	0.12
VLP (Secondary)	0.67	0.67	0.67
VLP (Secondary)	0.69	0.69	0.69

Table 2.1 Details of swirlers of Injector 1, 2, 3

2.2.3 Design Approach

Porous material is utilized in this study to optimize fuel-air mixing performance for liquid fuels and gaseous fuel applications. In the case of liquid fuel, as fuel is injected through the porous tube, a thin liquid sheet is produced on the inner surface of the tube. The air coming from bottom axial swirler generates swirling air stream to shear the liquid sheet and initialize the primary atomization. This design increases the fuel-air contact surface to achieve fine atomization. The prefilming surface area benefits the atomization especially under high pressure engine conditions. This design will not be affected by the liquid loading. It will work under all engine conditions. In the gaseous fuel application, the increased contact area between fuel-air at the injection point enhances fuel-air mixing. Furthermore, the gaseous and liquid dual phase fuel injection can be accomplished with this design.

2.3 LDV Setup

A schematic of the LDV setup is shown in Figure 2.9. The air supply system consists of an Atlas Copco GA90-series compressor, rated for a flow rate of 512 cfm at a pressure of 150 psig, with a built-in demoisturizer system, a 1040-gallon air settling tank, a pressure regulator and a series of 2" piping to connect it to the test setup. The temperature of air during the measurements is $\sim 60^\circ\text{F}$. The test setup consists of an air plenum, the injector assembly, combustor and the exhaust piping. The injectors are mounted on the air plenum with an adapter flange. A honeycomb straightener is placed inside the air plenum to obtain a uniform flow and to homogenies turbulence. In order to simulate the combustor, a confinement (rectangular section) made of plexiglass is placed on the air plenum. The LDV setup is mounted on a 3D traverse system. The traverse system has a resolution of 0.004" (0.1 mm). A liquid seeding generator is used to seed the flow. In the present study, diesel is used in the seeding generator to produce micron size particles. The particle size is < 5 microns. The particle seeder is connected to the upstream of the air plenum. This ensures that particles are uniformly dispersed in the air stream.

Figure 2.9 Schematic of LDV system

To ensure that the velocity of seeding particles represents the gas velocity, Stokes number (Stk) is calculated as shown in Equation 2.3. Stokes number is a dimensionless number that is used to define the behavior of particles suspended in a flow field. The Stokes number is the ratio of the characteristic time of a droplet to the characteristic time of the flow. For $Stk \gg 1$, droplets will not follow the flow due to their high inertia especially in high gradient flows. For $Stk \ll 1$, the particles follow the flow faithfully. The accuracy of droplets following the flow is $< 1\%$ if $Stk \ll 0.1$.

$$Stk = \frac{\rho_p d_p^2 u_0}{18\mu_g l_0} \quad (2.3)$$

In this study, the axial and radial velocity magnitudes of the CTRZ (Central T Recirculation Zone) are ~ 30 m/s and 20 m/s respectively. The calculated Stk are 0.047 and 0.04 . It shows that the seeding particles follow the flow faithfully. The velocity of particles measured by the PDI system can be considered as the gas velocity.

A Meriam 2100 series smart pressure transducer is connected to the air plenum to measure the pressure drop across the injector. Aerodynamic measurements are carried out at atmospheric conditions. The pressure drop across the injector is $4\% \pm 0.1\%$. The measured pressure drop is the differential pressure between upstream of the injector and the ambient pressure.

A PDI-200 series LDV system from Artium Technologies is used to measure the three velocity components in the flow field. The PDI system consists of a transmitter, a receiver, and two signal processors. The PDI system is set to acquire $30,000$ samples or wait for 60 seconds at each measurement point. The settings used for the LDV system are listed in Table 2.2. The exit diameter (D) of the injector is $1''$. The confinement size is chosen as $2D \times 2D \times 10D$ ($2'' \times 2'' \times 10''$). The confinement ratio used is 5.09 , which is the ratio of the cross-section area of the confinement to injector exit area.

During the aerodynamics measurements, the center of the injector exit is used as the origin for the 3D cartesian coordinate system with Z axis in the axial direction of the injector and X-Y horizontal plane representing the cross-section of the confinement. For injector 1, the measurements are carried out in X-Y plane at the $Z/D = 0.12$ from the injector exit. The measurements grid in the X-Y plane has a spatial resolution of $0.04D$. A total number of 1400 points are acquired in a square grid. Measurements near the wall of the confinement could not be acquired as the laser beams from the PDI system are blocked. Therefore, the measurement grid is $1.5D \times 1.5D$ even though the confinement is $2D \times 2D$. The PDI system is a two-component system. To obtain the third velocity component, the injector is rotated by 90° clockwise and measured in the same plane under the same operating conditions.

To investigate the shape and length of the Central Toroidal Recirculation Zone (CTRZ) of the full developed flow, the measurements are also carried out in the X-Z plane. The measurement plane is at $4D$ from the exit of the injector in the axial direction for both injector 1 and 3. The spatial resolution between the measurement points is $0.2D$ in the axial direction, and $0.04D$ in the horizontal direction.

Channel	Green	Blue
Wavelength (nm)	532	473
Focal length (mm)	300	300
Fringe spacing (μm)	2.66	2.365
Beam diameter (mm)	1.3	1.3
Beam separation (mm)	60	60

Beam waist (μm)	156.3 138.7
Velocity measurement range (m/s)	-100~100
Velocity accuracy	1%

Table 2.2 Setting details of LDV system

2.4 PDI Setup

PDI system is used to quantify the spray emanating from the injectors. A schematic of PDI system used in the present study is shown in Figure 2.10. PDI 200 from Aratum, which is used for LDV measurements is also used for spray measurements. The setting details for PDI system is shown in Table 2.3. The PDI measurements are used to measure the droplet size and velocity of the droplets. The axial and radial velocities of the droplets are measured in this test. During PDI droplet size measurements, the injector is mounted on a different test setup. This allowed the injector to face downwards and without confinement. This avoids the droplets sticking to the confinement. This is essential as the quality data is dependent on the signal from the droplets and data acquisition by the PDI system. If the droplets stick to the confinement, then the signal from the droplets is distorted and the PDI system rejects the particles assuming it is a stray signal. This leads to an overestimation of the droplet size. Since the atomization completes very close to the atomizer, the absence of the confinement will not affect the droplet size significantly though it can affect the aerodynamics. Jet-A fuel, GTL (gas to liquid) fuel and blended fuel (50-50 of Jet-A-GTL) are used in the present study. The spray is characterized for these fuels. Jet-A fuel is stored in a fuel tank and supplied by a fuel pump. The upstream pressure of Jet-A fuel is maintained at 40 psig. The GTL and the blended fuels are stored in separate fuel tanks and pressurized by the compressed air from the compressor. The fuel mass flow rate of fuel is monitored by a CMF10 micro motion Coriolis flow meter. At each measurement point, a total

15000 samples are acquired to calculate time-averaged droplet velocity and size. The maximum time is set at 60 s.

Figure 2.10 Schematic of PDI setup

Channel	Green	Blue
Wavelength (nm)	532	473
Focal length (mm)	500	500
Fringe spacing (μm)	4.4	3.9
Beam diameter (mm)	1.3	1.3
Beam separation (mm)	60	60
Beam waist (μm)	260.5	231.6
Velocity measurement range (m/s)	-100~100	
Velocity accuracy	1%	
Droplet size measurement range (μm)	-100~100	
Droplet size accuracy	1%~3%	

Table 2.3 Details of PDI settings

2.5 Patternator

Spray patternator is used to investigate the spray characteristics such as spray angle, fuel distribution in the cross-sectional plane downstream of the injector and a quantifiable symmetry of the spray in a plane. SetScan patternator, Model OP-600, is used in these measurements. The instrument has a measurement region of 7" diameter. The patternator is a laser extinction tomographic system that provides the drop surface area density (number of drops per unit volume at a specific location multiplied by the surface area of the drops). The local mass transfer, momentum, energy and species are directly proportional to the surface area density of droplets in the spray. The patternator has six lasers that are used to form laser sheets as shown in Figure 2.11. Linear photodiode array detectors are placed on the other end of the sheet. As the spray is illuminated by the laser sheet, the light extinction of the sheet is measured by the detectors. Initially, calibration of the local extinction coefficients is carried out. The path integrated extinction of laser sheets is linearly proportional to drop surface area density for liquid sprays. The data obtained from the detectors is used to reconstruct the distribution of the spray in the cross-section plane. These measurements are carried out on the injectors at the design phase.

Figure 2.11 SETScan patternator [83]

2.6 Fuel-Air Mixing Test Setup

A schematic of fuel-air mixing test setup is shown in Figure 2.12. The injector is mounted on the air plenum. A Meriam 2100 differential pressure transducer is used to monitor the pressure drop across the injector. A fuel feed system is connected to the injector. CO₂ is used as a gaseous fuel to simulate natural gas as the momentum flux ratio of CO₂ to air at room temperature is similar with the momentum flux ratio of natural gas to air at engine operating conditions. The momentum flux ratio is shown in Equation 2.4. Usually, while characterizing the fuel-air mixing momentum flux ratio needs to be kept constant, which ensures that the CO₂-air mixing profile obtained downstream of the injector represents the natural gas.

$$J = \left(\frac{\dot{m}_f}{\dot{m}_a} \right)^2 \cdot \frac{\rho_a}{\rho_f} \cdot \left(\frac{A_a}{A_f} \right)^2 \quad (2.4)$$

Figure 2.12 Schematic of fuel-air mixing test setup

As mentioned before, CO₂ is used to simulate gaseous fuel. CO₂ is supplied to the injector by a cryogenic dewar which contains liquid CO₂ stored at -160°C. A Mokon oil heat exchanger is used to heat the CO₂ to bring it to the room temperature. The oil heater is a counter flow heat

exchanger that maintains the CO₂ temperature at 300 K. CMF010 and CMF025 Micro Motion mass flow meters are used to measure the mass flow rate of CO₂ and air respectively.

In the present experiments, the CO₂ concentrations are measured at the exit of the injector by collecting samples with small micron sized probes. A four-probe rake is mounted at the exit plane of the injector. A vacuum pump at the end of each probe is used to suck the sample. The collected sample is passed to CO₂ sensors, COZIR make, through a series of flexible tubes. The output of the sensors is linear to CO₂ molar fraction. The mole fraction is converted into the fuel mass fraction. The measurements are carried out at different locations downstream of the injector with and without a confinement. The confinement is a square test section made of plexi-glass with dimensions of 3D x 3D x 10D (3'' x 3'' x 10''). The configuration of the confinement is shown in Figure 2.13. The four-probe rake is mounted on a 3D traverse system. The data of each sensor reading, and the corresponding location of the probe are recorded and controlled by LABVIEW. For the confinement case, the rake is moved in a square grid of size of 2.36'' x 2.36'' (60 mm x 60 mm), with an increment of 0.04'' (1 mm). The scanned area is smaller than confinement size because the size of rake hinders the rake to move closer to the wall. For the unconfinement case, a circle measurement grid is used with an outer diameter equal to the flare diameter. This is due to the dilution of the mixing zone with the ambient air. The radial increment is 0.04'' and the circumferential increment is 5°. The sampling duration at each measurement point is set at 50 s to ensure that the sensor readings reach a steady state.

Figure 2.13 Confinement for mixing studies

Sensor Calibration

The COZIR sensors need to be calibrated before and after the measurements. In the calibration procedure, CO₂ is injected upstream of the air plenum to obtain fully premixed CO₂ and air. During the calibration, CO₂ mass flow rates are increased from zero to twice the mean mass flow rate of CO₂ in the actual measurements. At each CO₂ flow rate, the sample is collected by the probes at the exit of the injector. The sensor readings are noted down along with the measured CO₂ and air mass flow rates. The pressure drop across the injector is maintained constant during the whole procedure, which is the same condition as in the actual tests. The sample collection time at each point is set at 50 seconds to ensure that the sensor readings reach steady state. CO₂ mass fraction is calculated based on measured CO₂ and air mass flow rate as shown in Equation 2.5. A linear calibration curve is obtained from calculated mass fraction and corresponding sensor reading.

$$MF = \frac{\dot{m}_f}{\dot{m}_f + \dot{m}_a} \quad (2.5)$$

A typical calibration curve for one sensor is shown in Figure 2.14. The curve represents a linear relation between sensor reading and measured mean mass fraction from the flow meters. Each sensor in the four-rake probe has its own calibration curve.

In the actual measurements, CO₂ is injected through the fuel inlet to the injector. The four-probe rake scans the area at the exit plane of the injector. Based on the calibration curve, the measured sensor readings in actual tests are converted to the local mass fraction. From the several tests carried out in the past it is concluded that there is approximately 5% error in the mass fraction measurements. The error is due to several factors, which include the variation of humidity in air during the experiments, different response times between the sensors due to the sample tubing and vibrations of the probes.

Figure 2.14 Calibration curve for the sensor

2.7 Combustor Configuration

In the present study, gaseous and liquid fuels are used during the combustion tests. Natural gas is used as a gaseous fuel. Jet-A fuel, GTL fuel and blended fuel (50-50 of Jet-A and GTL) are tested separately to investigate the fuel flexibility of the injectors.

The combustion tests are carried out at various inlet air temperatures and equivalence ratio conditions. The pressure drop across the injector is maintained at 4% during the measurements. The size of the combustor used is 3" x 3" x 10", which is the same size as the confinement that is used in mixing studies. The combustor is made of stainless steel 316L. It has two side optical access with quartz glass. This is used for flame visualization studies. Flame visualization is carried out with a CH* Chemiluminescence filter mounted on a CCD camera (Flame image model of LaVision make). An exposure time of 10 ms is used during the flame imaging.

2.8 Emission Analysis

Emission analysis is carried out during these studies. To ascertain the quality of the combustion, the emission data of the local combustion products is acquired inside the combustor. The sampling probe is shown in Figure 2.15, is made of 316 L stainless steel tubes. It has an inner tube with diameter 1/8" for sample collection. To avoid chemical reactions occurring inside the tube and the tube from overheating, the 1/8" sampling tube is surrounded by two secondary cooling tubes with diameters of 1/4" and a 3/8". An oil convection cooling system is used to keep the sampling tube at a constant temperature. To increase the accuracy of the collection of species in the combustion products, the temperature of the oil is kept at 230°F by a Mokon's HTF 350 series heat transfer oil system. This heat transfer system can be used to keep the oil temperature up to 350°F. This condition ensures that there is no condensation of water vapor in the combustion products inside the sampling tube. Further the chemical reactions are quenched at this temperature

and the sample species are preserved. The sampling probe is connected to the analyzers through a heated line, a heated filter and a chiller. The heated line is connected to the other end of the sampling probe. The temperature of the heated line is maintained at 380°F. As the UHC (unburned hydro carbon) condenses in water, a part of the sample is directed from the heated line to the Hydrocarbon (HC) analyzer. The rest of the sample is passed through a chiller to remove water vapor in the sample. The dry sample from the chiller is passed on to NO_x analyzer (CAI make and Model 600 CLD), CO and O₂ analyzers (CAI make and model 600 NDIR emission analyzer), to measure NO_x and O₂ concentration in the combustion products. The CLD emission analyzer is used to measure NO and NO₂ based on dry sample. The operating principle is based on the chemiluminescence concept. NO is measured by converting NO to NO₂ and O₂. The concentration of NO is obtained by the converted NO₂. The CAI Model 600 NDIR emission analyzer measures CO and O₂ concentrations based on dry sample. These instruments are calibrated before each test.

Figure 2.15 Details of emission probe design

Figure 2.16 Measurement grid for emissions

The measurements are carried out at 6'' downstream plane of the injector exit. Data is acquired in a square grid of size 2.6'' x 2.6'', as shown in Figure 2.16. Total 25 sample points are acquired. The measurement grid is smaller than the combustor size as the sampling probe traverse system has a few physical obstructions that prevents the sampling probe to reach closer to the wall. At each measurement point, the data is acquired for 60 seconds to ensure that a steady state is reached.

The measured NO_x is the sum of nitrous oxide (NO) and nitrogen dioxide (NO_2). The unit of measurement is ppmvd (based on dry sample). The NO_x concentration is corrected to 15% O_2 as a reference for comparison of different sets of operating conditions. The calculation of NO_x corrected to 15% O_2 is shown in Equation 2.6.

$$X_{\text{NO}_x, \text{dry@15\%O}_2} = (X_{\text{NO}, \text{dry}} + X_{\text{NO}_2, \text{dry}}) \cdot \frac{20.9-15}{20.9-X_{\text{O}_2, \% \text{dry}}} \quad (2.6)$$

Chapter 3 Non-Reacting Flow Measurements

In this chapter, non-reacting flow measurements are carried out on the injectors. The non-reaction flow measurements are broadly classified into aerodynamics studies on the injector, spray characterization and fuel-air mixing studies. All these measurements are carried out at atmospheric pressure and at ambient temperature.

3.1 Aerodynamic Studies on Injectors

The aerodynamic studies on the injectors are discussed in this section. The aerodynamics studies of injector 1 and 3 are studied in the present study. The radial-radial swirler used for injector 2 and 3 are the same therefore it is expected that the aerodynamics performance of injector 2 and 3 will be the same. The effect of the confinement on aerodynamics is discussed as well.

3.1.1 Aerodynamic Studies on Injector 1

Figure 3.1 (a) (b) (c) show the axial, radial and tangential velocity contours at $Z/D=0.12$ from the exit of injector 1. In Figure 3.1(a), the vectors represent the resultant velocity of the radial and tangential velocity components in the horizontal plane. The vector length represents the velocity magnitude of the resultant in the horizontal plane. The magnitude of the axial velocity component is represented by the colormap. A closed shape Central Toroidal Recirculation Zone (CTRZ) can be observed from the contour plot. The axial velocity shows a symmetric distribution at the exit of the injector. As shown in Figure 3.1(a), the swirling flow emerges from the exit of the injector and expands in the confinement. However, due to the sudden expansion of the flow in the confinement, an adverse pressure gradient is created at the center of the injector. The flow turns inwards in the center leading to the formation of the CTRZ. The presence of the CTRZ increases the turbulent intensity which in turn improves the fuel-air mixing quality. Besides the CTRZ,

Corner Recirculation Zones (CRZ) are also observed in the corner regions of the confinement.

This assures that the fuel remains in the recirculation zones to achieve the flame stabilities.

(a)

(b)

(c)

Figure 3.1 Velocity contours at $Z/D=0.12$ Injector 1

(a) Axial velocity contour (b) Radial velocity contour (c) Tangential velocity contour

Figure 3.1 (b) shows the radial velocity contour at $Z/D=0.12$ for injector 1. The radial velocity contour is asymmetric with higher velocities in the upper region. The swirling flow from the inner venturi has three velocity components. The positive radial velocity implies that the flow is expanding outwards from the center of the injector to the peripheral region. The radial component increases until the flare edge. This is the region of the maximum mass flux of air. Beyond this region the radial component decreases due to the confinement. The radial velocity must reach zero near the wall due to the no-slip condition. The radial velocity in the center of the injector is near zero. The negative radial velocity at the four corner regions is also observed as shown in the figure. This indicates that a reverse flow is formed due to the confinement which is also the reason for the formation of CRZs.

Figure 3.1 (c) shows the tangential velocity contour at $Z/D=0.12$. The negative value shows that the flow is rotating in the clockwise direction when seen from the downstream of the injector.

Similar to the radial velocity contour, the tangential velocity contour also shows higher velocity on one side of the injector. The variation is less than 15%. In the injector design, the primary swirler generates the counter-clockwise rotating flow while the secondary swirler generates the clockwise rotating flow. The clockwise rotating flow shown in the figure indicates that the secondary swirler controls the flow structure. However, counter clockwise rotating flow near the exit of the injector can be observed as a result from the inner primary swirler.

The axial and radial velocity are also measured along X-Z plane. Figure 3.2 (a) and (b) show the axial and radial velocity contours from 0.1D to 4D downstream of the injector. The vectors represent the velocity component of axial and radial velocities in X-Z vertical plane. The direction of the vectors represents the direction of the flow. As can be observed, the air is flowing towards upstream in the center of the injector and flows downstream in the periphery region of the injector exit. This demonstrates that there is a reverse flow in the center due to CTRZ. The structure of the CTRZ can be clearly observed.

In Figure 3.2 (a), the shape and structure of the central toroidal recirculation zone (CTRZ) are shown. The axial velocity distribution is symmetric along the center of the injector. The length of the CTRZ can be determined by locating the axial velocity stagnation distance. As shown in the figure, axial velocity reaches a stagnation at $\sim 3.15D$ downstream of the injector. This is the length of the CTRZ. The size of the confinement also influences the shape of the CTRZ. In the present scenario, the width of the confinement restricts the expansion of CTRZ as is evident from the flow turning near the wall at $\sim Z/D=0.6$.

Figure 3.2(b) shows the radial velocity contour. As can be observed, the radial velocity is zero in the center of the injector. Near the injector exit, the radial velocity increase to a maximum of ~ 20 m/s as represented by blue and red regions and decreases towards the wall. This is also

shown in Figure 3.1 (b). This demonstrates a strong CTRZ which is needed for flame stabilization [84]. Beyond $Z/D = 0.6$, the radial velocity is near zero along the axial direction. This shows that the swirl effectiveness is reduced beyond $Z/D = 0.6$ where the flow impinges on the confinement walls and then turns in the axial direction. It also represents that the flow expands outwards from the injector exit up to $Z/D=1.2$, the flow then turns inwards in the center beyond $Z/D=1.2$. This is due to the confinement effects.

Figure 3.2 Velocity contours at X-Z plane; (a) Axial velocity contour (b) Radial velocity contour

3.1.2 Aerodynamic Studies on Injector 3

Figure 3.3 (a) and (b) shows the aerodynamics studies on injector 3 and the corresponding axial and radial velocity contours in X-Z plane [85]. In these studies, the standard size $2D \times 2D \times 10D$ could not be used. Therefore, a $2D \times 2D \times 5D$ confinement is used for these measurements. It was concluded from previous studies [85] that the length of the confinement has minimal effects

on the size of the recirculation zone [85]. In Figure 3.3 (a), a similar shape and size CTRZ is observed for injector 3, although injector 3 has a relatively higher positive velocity at the edge of the injector and a higher reverse velocity than injector 1. The higher axial velocity confirms that injector 3 has a lower swirl number compared to injector 1. The length of the CTRZ which is decided by the stagnation point is $\sim 3D$, which is about the same as the length of the CTRZ in injector 1. The width of the CTRZ for injector 3 is also similar to injector 1.

Figure 3.3 Velocity contours in X-Z plane for injector 3 [85]

Figure 3.3 (b) represents the radial velocity contour in injector 3. The shape of the CTRZ is depicted by a solid black line. Similar to injector 1, air flows outwards near the injector exit as represented by blue and red regions. The magnitude of the radial velocity in injector 3 is same as injector 1, nevertheless the radial velocity reaches zero at $Z/D=0.7$, which is higher than Injector 1. Due to the difference in the vane heights in the radial-radial swirler for injector 1 and 3, the

effective area is larger for injector 3 when compared to injector 1. Therefore injector 3 has higher air flow rate than injector 1 with a stronger turbulent swirling jet.

3.1.3 Comparison between Injector 1 and 3

A comparison of axial, radial and tangential velocity profiles along the centerline is made between injector 1 and 3 at the exit of the injector, as shown in Figure 3.4. Figure 3.4 (a) shows the axial velocity profiles for the injectors. The axial flow velocity for injector 3 is much higher than injector 1. The velocity peak occurs at $\sim 0.5 R/D$, where the flare is located. However, the reverse flow velocity is in the same range. Injector 1 has a central axial swirler in the porous tube. As the central axial swirling flow interacts with the secondary swirling flow from the venturi, it weakens the central reverse flow. Similar effect is observed for injector 3 due to the presence of the pressure swirl atomizer in the center. Therefore, the injectors have same reverse flow velocity. It should be noted that the velocity profiles are acquired at the exit of the injector. The velocity stagnation point inside the injectors can be different for both the injectors.

The radial velocity for both injectors are almost the same as shown in Figure 3.4 (b) which shows that the flow expansion rate is the same for both the injectors. The tangential velocity which indicates the rotational component of the flow is shown in Figure 3.4 (c). It is apparent that injector 1 has a smaller maximum velocity peak which indicates that it has a weak swirl. The velocity peak occurs at $\sim 0.5D$. Further, it can be observed that the velocity gradients between $\pm 0.5 D$ is different for both the injectors. The velocity gradients give an indication of how fast the flow is accelerating from the center of the injector towards the periphery. As can be observed for injector 1, the velocity gradients are less compared to the injector 3. Since there is flow through the center for the injector 1, the interaction of the inner axial flow through the porous tube and the primary swirl effectively reduces the velocity gradients. However, in the case of injector 3, due to the presence of the

pressure swirl atomizer, there is a stagnation region in the center. The velocity gradients observed is due to the primary swirling air filling the void. It is apparent that the velocity from the venturi surface to the center should tend to zero velocity. Therefore, the gradients are steeper for injector 3.

One of the parameters that determine the performance of the injector is the swirl number (SN). Swirl number is calculated to quantify the swirl intensity of the injectors. Swirl number is characterized by the ratio of the axial flux of angular momentum to the axial flux of axial momentum, and defined by the following equation:

$$SN = \frac{J_{tg}}{J_{ax}} = \frac{\int_0^R w u r dr}{\int_0^R u u r dr} \quad (3.1)$$

where J_{tg} is momentum of tangential velocity, J_{ax} is momentum of axial velocity. w is the local tangential velocity, u is the local axial velocity and, r is the radius of the injector.

(a) Axial velocity

(b) Radial velocity

(c) Tangential velocity

Figure 3.4 Comparison of velocity components between injector 1 and 3 at $Z/D=0.12$

The SN is calculated from the measured velocity data at $Z/D= 0.12$. The calculated SN of injector 1 is 0.75 and for injector 3 the SN is 0.6, which indicates that these are low strength swirler. The advantage of lower SN is that it has higher axial velocity. The higher axial velocity reduces the residence time inside the combustor and will reduce NO_x emissions. However, there is a risk of flame extinction. On the contrary, for higher strength swirler (with swirl number greater than one) has better flame stabilization performance over a wide range of load conditions.

Figure 3.5 shows the comparison of turbulent kinetic energy between injector 1 and 3. In fluid dynamics, TKE is defined as the kinetic energy of the turbulent part of the flow, and characterized by the following equation:

$$TKE = \frac{1}{2}(U'^2 + V'^2 + W'^2) \quad (3.2)$$

where TKE is turbulent kinetic energy, U' is root mean square (RMS) of axial velocity, V' is RMS of radial velocity and W' is RMS of tangential velocity. The measurements are carried out at $Z/D=0.12$ plane. This is the region where fuel-air mixing takes place. TKE has a peak value around $R/D = 0.56$ which is also the location where the axial velocity has a maximum value. This is due to the strong turbulence generated at the periphery of the flare due to the formation of shear layer. The minimum TKE is observed at the center of the recirculation zone for both the injectors. TKE is higher for injector 3 than injector 1. This is due to the higher mass flow rates of injector 3.

Figure 3.5 Comparison of TKE at $Z/D = 0.12$ plane between injector 1 and 3

The comparison of Turbulent Intensity (T.I.) between injector 1 and 3 is shown in Figure 3.6. T.I. is an important parameter that is used to calculate the intensity of turbulence in the flow.

The effectiveness of the fuel-air mixing is governed by turbulent intensity. The calculation is shown in Equation 3.3,

$$T.I. = \frac{\sqrt{U'^2 + V'^2 + W'^2}}{U_o} \quad (3.3)$$

where U_o is the upstream mean velocity and calculated from Equation 3.4.

$$U_o = \dot{m} / \rho A_{upstream} \quad (3.4)$$

As shown in Figure 3.6, injector 1 has a stronger T.I. This is due to the smaller axial velocity of injector 1. The radial velocities for both injectors are in the same range. This leads to an increased T.I. The peak of T.I. for injector 1 is at $R/D=0.5$, whereas the peak for injector 3 is at $R/D=0.6$. This indicates that injector 3 has a wider swirling flow than injector 1.

Figure 3.6 Comparison of T.I. at $Z/D=0.12$ plane between injector 1 and 3

3.2 Spray Characterization

In this section, the non-reacting cold spray measurements are carried out by using patternator and a 2D PDI system. In the spray characterization process, several key parameters relating to the spray are measured. Some of the parameters are droplet size distribution, mean

parameters such as D_{10} , D_{32} (also called Sauter mean diameter (SMD)), spray symmetry, and representative volume-based diameters such as $D_{0.5}$ and $D_{0.9}$. These parameters are essential to understanding the fuel flux distribution inside the combustor.

3.2.1 Patternator Studies

Patternation tests are carried out on injector 1 to investigate the spray structure and distribution in the downstream of the injector. These tests are essential as the properties of the porous material is not known and there is no literature available on how the porous material will generate a spray. One of the key factors in this study is to understand the spray emanating from the injector 1 and evaluate the symmetry of the spray produced by the porous material. This is essential for the injector design. Patternation tests give the droplets surface area density distribution. The droplets surface area density is proportional to the fuel concentration and it is the product of the droplets surface area and the counts of droplets per unit volume. The distribution of droplet surface area density plays a significant role in combustion applications as the surface area determines the rate of evaporation of the fuel. The evaporation rate is critical for fuel-air mixing and emissions.

Figure 3.7 shows the droplets surface area density distribution in a plane downstream of injector 1. For all cases, a hollow cone spray is observed with low fuel concentration in the center and a doughnut shaped region with high fuel concentration. The fuel emanates from the porous tube along its surface. The fuel is sandwiched between the air flowing through the porous tube and the primary swirling air. This leads to the formation of doughnut shape fuel distribution. The expanding flow from the injector leads to the hollow cone spray at the exit of the injector. Figure 3.7 (a) (b) shows the surface density contours at a constant pressure drop of 4% and at equivalence ratio of 0.6 and 0.8. The contour plot is at the horizontal plane of $Z/D=0.5$. As it can be observed,

at a constant pressure drop, the increase in equivalence ratio leads to increase in the distribution of droplet surface area density. The increase in droplet surface area is due to increase in volume flux of the fuel that leads to increase in droplet number density. The increase in equivalence ratio also makes the spray more uniform and symmetric. As the equivalence ratio is increased, the fuel manifold pressure increases. This leads to more even distribution of the fuel around the porous tube leading to a more symmetric spray.

Figure 3.7 Droplets surface area density distribution

Figure 3.7 (a) and (c) shows the droplets surface area density at two pressure drop conditions at a constant fuel flow rate. It is observed that the droplets surface area density increases with increasing pressure drop. As the pressure drop across the injector increases, the velocity of the air through the porous tube increases. The higher air velocity enhances the surface stripping phenomena on liquid sheet that is spread on the porous tube. The surface stripping is a process by which the liquid is stripped off from the liquid by the high momentum air stream. The surface stripping tends to produce fine atomization [87][88]. The fine atomization increases the counts of droplets per unit volume leading to a higher droplets surface area density (smaller droplets have higher surface to volume ratio).

Figure 3.7 (a), (d) show the droplets surface area density distribution at different downstream locations at constant pressure drop and equivalence ratio conditions. As shown in the figures, the droplets surface area density is more uniform downstream of the injector. Also, the mean droplet surface area density increases downstream of the injector although the maximum value decreases. The decrease in the maximum value of droplets surface area density is due to the expansion of the spray in the combustor along with the flow. The atomization is completed very close to the atomizer. However, the droplet concentrating is very high. As the spray expands the droplet density per unit volume decreases leading to decrease in local droplets surface area density. For the same reason, the droplets surface area density also becomes uniform. From patternation tests, the symmetry and fuel distribution in a plane downstream of the injector is examined. However, this is not sufficient for understanding the spray combustion. Therefore, PDI measurements are carried out to investigate the spray droplet size distribution and spray dynamics.

3.2.2 Droplet Size Measurements

The Particle Doppler Interferometry (PDI) measurements are carried out on Jet-A fuel and blended fuels without a confinement. This is to avoid the droplets impinging on the confinement which will cause the distortion of the doppler signal. PDI system cannot validate the distorted signal thereby rejects the droplets and generates very low data rates. Further, the data will be biased towards bigger droplets. The spray characteristics for all three injectors are investigated.

Figure 3.8 Droplet size frequency distribution curves based on number and volume

Figure 3.8 shows a typical droplet size-frequency distribution at a selected measurement point in the measurement grid. Though this graph is not directly used in the data analysis, it is shown here to indicate the pertinent droplet weighted diameters that are used in the spray analysis. Various representative diameters are marked on the plot. To characterize the spray, representative diameters are necessary to completely define the spray. This is done so that the experimental data can be condensed to carry out the meaningful data analysis. Even for CFD analysis where PDF's are required, the experimental data is condensed to these representative diameters and a curve fitting algorithm is used to generate the spray distribution. A typical size-frequency distribution follows a log-normal distribution. In the spray analysis, the distribution is more closely associated

to Rosin-Rammler distribution. The blue curve shows the droplet size versus number distribution. As can be observed, it has a typical log-normal distribution which is skewed towards the smaller droplets. The volume distribution on the other hand has a much broader peak and skewed towards right as expected as the volume of the fluid is mostly in the larger droplets. The weighted effect of the large droplets skews the distribution. For combustion applications, it is the volume distribution of the fuel that is essential. At every measurement point, the particle distribution data obtained is condensed to representative diameters such as D_{10} , D_{32} , $D_{0.1}$, $D_{0.5}$ and $D_{0.9}$. D_{10} is the mean diameter of the entire spray, D_{32} is the droplet diameter based on the ratio of volume to surface area in the entire spray.

Figure 3.9 shows the comparison of various diameters of injectors 1, 2 and 3. The measurements are carried out along X axis. It is apparent that D_{10} , the linear average parameter of all the droplets is the smallest diameter at each measurement location for all three injectors. The difference between D_{10} and D_{32} increases from the center to the periphery of the injector. This is because the effects of the larger droplets on the D_{32} is much more significant than on D_{10} . This indicates that the larger droplets are present in the outer zones of the spray. Furthermore, D_{32} is smaller than $D_{0.5}$ as expected, which explained in Figure 1.5. The distribution of $D_{0.5}$ is not as symmetric as that of D_{32} and D_{10} in all three injectors as $D_{0.5}$ is sensitive to volume flux variations. In hollow cone sprays, the spray volume is less in the center and peripheral regions, which significantly effects $D_{0.5}$ distribution. In addition, injector 3 has larger $D_{0.5}$ in the center region when compared to injector 1 and 2 due to the difference in the design of atomizers. Moreover, as can be observed, the width of the spray of injector 1 is narrower than injector 2 and 3 due to the presence of larger droplets at same radial location of injector 1. In practical combustion applications, characteristic mean diameter D_{32} is considered as an important parameter since it

determines the surface area available for the evaporation of the fuel for a given volume of the droplets. D_{32} is used in this present study for comparing different injectors. For all injectors, D_{32} increases from the center of the injector, and stays constant from the injector exit ($R/D=0.5$) to $R/D=0.8$. The following sections discuss the comparison of different injectors.

(a) Injector 1

(b) Injector 2

(c) Injector 3

Figure 3.9 Comparison between various diameters, $\Delta p/p = 4\%$, $\phi = 0.6$, $Z/D = 0.4$

To investigate the symmetry of the distribution of the spray, the measurements are carried out along four lines for injector 1 as shown in Figure 3.10. The central orange circle represents the flare edge of the injector, which is the exit of the injector. The distance between the measurement points is $0.04D$. The radial distance measured is from $-1.2D$ to $1.2D$. The D_{32} distribution along the four lines is shown in Figure 3.11. The measurements are carried out with Jet-A fuel, at $\Delta p/p = 4\%$ and equivalence ratio $\phi = 0.8$. It is apparent that the D_{32} distribution is symmetric along the centerline. There are minor variations in the outer periphery of the injector. However, the volume flux on the outer periphery is relatively low for any spray.

The D_{32} is $\sim 20\mu\text{m}$ in the center of the injector. D_{32} is smallest in the center of the injector. It increases steeply from center to $R/D = 0.24$, which is the radius of the porous tube. The smaller droplets are produced due to the surface stripping on the liquid sheet. This also indicates that the surface stripping is dominant on the porous tube. Moreover, due to the presence of low velocities in CTRZ, the smaller droplets tend to follow the flow whereas bigger droplets because of their inertia tend to disperse outwards. The D_{32} is almost constant $\sim R/D = 0.5$, which is the radius of the

flare of the injector. This is also the region with strongest swirl. D_{32} gradually increases along the periphery of the injector.

Figure 3.10 Measurements locations (number represent the line scans)

Figure 3.11 D_{32} distribution of injector 1 along the four center lines, $\Delta p/p = 4\%$, $\phi=0.8$, $Z/D=0.4$

Figure 3.12 shows the fuel volume flux and D_{32} as a function of radial distance, and Figure 3.13 shows the fuel volume flux with droplets mean axial velocity as a function of radial distance. Fuel volume flux distribution is a significant parameter to understand the spray structure. The volume flux distribution indicates the location of fuel in the combustor, also termed as fuel

placement. Often the volume flux distribution should align with injector aerodynamics for efficient combustion. The volume flux is used to determine the spray characteristics such as representative diameter and weighted mean diameters.

Figure 3.12 D_{32} and fuel volume flux distribution for injector 1, $\Delta p/p=4\%$, $\phi=0.8$, $Z/D=0.4$

Figure 3.13 Fuel volume flux and droplets mean axial velocity distribution for injector 1
 $\Delta p/p=4\%$, $\phi=0.8$, $Z/D=0.4$

In Figure 3.12, the volume flux is calculated with a code developed in-house, the accuracy is within 10% at atmospheric conditions. Although the distribution of D_{32} is symmetric, the

distribution of the volume flux shows asymmetric. This is due to the unequal distribution of fuel in the fuel manifold surrounding the porous tube. Moreover, the porosity of the material is not homogenous and there could be variations in the porosity around the circumference. The penetration rate can vary along the surface. The manufacturing defects can also cause some deviations resulting in unequal volume flux distribution. This is also confirmed in the patternation tests. It is also evident that the volume flux associated with the larger droplets is lower in the periphery region of the spray even though D_{32} is increasing in that region. It should be noted that these measurements are carried out at atmospheric conditions. At engine conditions, these large droplets will undergo secondary atomization and produce a finer spray. The maximum volume flux appears around $X/D=0.5$. This is the location of the edge of the injector flare where the mean axial velocity is maximum, as shown in Figure 3.13. It indicates that the maximum fuel volume flux and mass flux of air are in the same spatial location. This will lead to a good fuel-air mixing quality. A good volume flux distribution improves the liquid fuel atomization quality, fuel-air mixing performance and facilitates evaporation process during combustion. In the present case, the D_{32} is $\sim 10 \mu\text{m}$ in the center of the injector which is good for rapid evaporation and combustion. The low D_{32} in the center, where CTRZ exits, assists in stabilizing the flame. D_{32} gradually increases along the periphery of the injector.

For comparative studies between injectors and different fuels a single parameter is essential. The weighted average D_{32} , which also refers to as Global D_{32} , is used for this purpose. It is calculated based on the local volume flux and the local D_{32} , as shown in the following equation:

$$\overline{D_{32}} = \frac{\sum V_i D_{32i}}{\sum V_i} \quad (3.5)$$

where V_i is the local volume flux, D_{32i} is the local D_{32} . i is the local point along X axis. The smaller the Global D_{32} the better the spray quality will be.

The effects of axial distance on the D_{32} is also discussed. The D_{32} distribution at different downstream locations are shown in Figure 3.14 (a). The measurements are carried out along line 1 as shown in Figure 3.10. As mentioned earlier in the patterning study, the width of the spray increases and the surface area density distribution tends to be more uniform downstream, which is also confirmed in Figure 3.14 (a). The D_{32} distribution of the spray increase as Z/D is increased. However, the D_{32} in the center of the injector is not affected. This can be attributed to the spray density and atomization process. The atomization, liquid sheet breakup and formation of droplets, is complete close to the atomizer. But, the droplets still experience acceleration as can be observed from the velocity gradients in Figure 3.14 (b). This makes the droplets elliptical. The PDI instrument can only measure spherical droplets. Mostly, large droplets are rejected by the PDI instruments at measurement locations close to the injector. Moreover, the spray density is high close to the injector. This also leads to rejection of particles by PDI instrument. The smaller droplets experience less deformation and therefore they are recognized by the PDI instrument. In the center of the spray the spray density is less, and the droplets are small. Therefore, the PDI instruments measures the D_{32} accurately. As Z/D increases there is no effect on D_{32} . However, in the regions of high spray density like at flare edge, the measurements are biased towards smaller droplets when measurements are carried out close to the injector. This is the reason as Z/D increases the D_{32} increases since more large droplets are recognized by the PDI instrument. The droplets have either undergone secondary atomization or turned spherical as the momentum decreases due to expansion of the flow.

Figure 3.14 (b) and (c) show the mean droplet velocity profiles at different downstream locations. Figure 3.14 (b) shows the axial velocity distribution. As can be seen there is a negative velocity region from -0.4-0.4 X/D indicating that it is a hollow cone spray. As expected, the

maximum axial velocity and the CTRZ decrease downstream of the injector. The spray spreads outwards due to the swirling flow. The maximum axial velocity of the droplets still maintains the same at $Z/D=0.8$. In Figure 3.14 (c), the maximum radial velocity decreases at downstream locations of the injector. At $Z/D = 0.2$, the spray in the center contains both large droplets and small droplets. Since the radial velocity shown in the Figure 3.14 (c) is a weighted average velocity of the droplets, it is biased towards large droplets. The large droplets however have inertia which is directed radially outwards. This leads to a more “shallow” curve with less velocity gradients. At $Z/D = 0.8$, the spray is already spread across and the larger droplets are shifted radially. However, the smaller droplets remain in the center due to their low stokes number. This is the reason for steep velocity gradients in the center as measurements are carried downstream of the injector. The effect of large droplets moving outwards can also be noted beyond X/D at ± 0.4 . The radial velocity fluctuations near the proximity of the injectors exit ($Z/D=0.2$) is due to the flare affects.

(a) D_{32}

(b) Droplets mean axial velocity

(c) Droplets mean radial velocity

Figure 3.14 Droplets characteristics at different downstream locations of injector 1

$$\Delta p/p = 4\%, \phi = 0.8$$

The comparison of mean velocity profiles between the air and the spray at $Z/D=0.2$ is shown in Figure 3.15. The measurements are carried out along X-axis centerline without confinement for both aerodynamic and spray measurements. The comparison is made to highlight that the peak velocity regions of spray and air coincide as can be observed in Figure 3.15 (a). The width on the droplets from peak-to-peak ($-0.6 < X/D < 0.6$) is narrower when compared with air.

This is expected as the larger droplets have higher inertia and tend to spread much slower than the air. The peak velocity of the droplets is smaller than the air. This is also expected as the energy required for atomization is drawn from air in the form of momentum exchange between air and liquid phase. This is reflected in the velocity profiles. Further, the droplets mean axial velocity is biased towards larger droplets which has less velocity than the air. In Figure 3.15 (b), the radial velocity of the droplets shows the same trend as air flow, although the magnitude of the droplets velocity is much smaller than the air velocity, which indicates that during the momentum exchange the swirl strength is decreased and the spray has a weaker swirling strength and therefore the rate of dispersion of the spray in the combustor is slower. Moreover, the magnitude of the reverse velocity of the droplets in the CTRZ is almost the same as the velocity of the air flow. This suggests that the droplets size in the center is sufficiently small due to surface stripping compared to the seeding particles used during aerodynamic measurements.

(a) Mean axial velocity

(b) Mean radial velocity

Figure 3.15 Comparison of the velocity between the air and droplets for injector 1, unconfined case, $Z/D=0.2$

The effects of the pressure drop on the spray characteristics is also carried out. As shown in Figure 3.16 (a), the local D_{32} decreases as the pressure drop increases. This is because the atomization (D_{32}) is inversely proportional to the velocity of the air for airblast injectors. With higher air velocity, there is an increased momentum exchange between air and liquid leading to formation of smaller droplets. The increase in surface stripping of the liquid sheet at higher pressure drop also leads to smaller droplets. It is also observed that the width of the spray increases with the pressure drop. As the droplets are smaller, they follow the air more faithfully. This is evident from the aerodynamic data presented in Figure 3.15.

Figure 3.16 (b) and (c) shows the droplets mean velocity profile at different pressure drops. The velocity profiles follow the same trend for all the cases, although the magnitude of both axial and radial velocities increase with pressure drop. As the pressure drop increases the change in velocity increases marginally. This is because the velocity changes with the square root of pressure drop. Also, it should be noted that the reverse flow velocity in the CTRZ increases as the pressure

drop increases. As explained before, with the increase in pressure drop the particles in the CTRZ are smaller which have higher velocities. Figure 3.16 (c) shows that the location of the maximum radial velocity shifts outwards when the pressure drop is increased. This is again due to the formation of finer spray at higher pressure drop as shown in the D_{32} plot in Figure 3.16 (a).

(a) D_{32}

(b) Mean axial velocity of spray

(c) Mean radial velocity of spray

Figure 3.16 The effect of pressure drop on the spray aerodynamics for injector 1,
 $\phi=0.8$, $Z/D=0.4$

Another significant requirement for this injector is to use variety of the fuels so that it can be operated not only on Jet-A fuel but also on gas-to-liquid (GTL) fuel and blended fuels. The GTL fuel used in this study is S8 fuel which is derived from natural gas. The details of the physical and chemical properties of the GTL fuel are given in references. [96][97] To investigate the consistency of the atomization for GTL fuels, the PDI measurements are carried out with 100% GTL fuel and blended fuel (50% GTL & 50 % Jet-A by volume) as well. The results are shown in Figure 3.17, the fuel flow rate and operating conditions for GTL and blended fuels are kept the same as for Jet-A fuel. Figure 3.17(a) shows that the D_{32} distribution between GTL and blended fuels are identical. The D_{32} is marginally larger than the Jet-A fuel although the differences are not significant. D_{32} is a function of air velocity, fuel velocity, liquid sheet thickness and liquid surface tension. Since the other parameters and operating conditions are the same, the smaller D_{32} indicates that GTL fuel has a smaller surface tension than Jet-A fuel. The difference is less than 8% due to

the similar physical properties. The identical D_{32} distribution for GTL and blended fuels suggest that GTL fuel dominates the spray characteristics in the blended fuel.

In Figure 3.17(b) and (c), the axial and radial velocity profiles are presented. The magnitudes of both axial and radial velocities for Jet-A fuels are higher than GTL fuel and blended fuels. The variation among Jet-A fuel, GTL and, blended fuels is $\sim 10\%$. The higher mean axial velocities of Jet-A fuel also confirm that smaller droplets are produced for Jet-A fuel. GTL fuel is a combination of compounds consisting of acyclic hydrocarbons. The molar weight is almost same as Jet fuel. However, due to the significantly different composition, the combustion characteristics varies. The differences observed in spray characteristics in Figure 3.17 shows that atomization is not significantly affected. Therefore, any difference in the emissions of all these fuels can be directly attributed to their composition.

(a) D_{32}

(b) Mean axial velocity

(c) Mean radial velocity

Figure 3.17 Comparison of GTL, blended and Jet-A fuels for injector 1

$$\Delta p/p = 4\%, \phi = 0.8, Z/D = 0.4$$

3.2.3 Comparison between the Injectors

The comparison of the spray characteristics between the injectors is analyzed in this section. Injector 1 and 2 have the same fuel feeding system, i.e. the porous material (airblast atomizer) and fuel manifold but with different swirler geometries (top section); injector 2 and 3 have the same counter radial-radial swirler but different fuel feeding systems (injector 1 achieves atomization by

the surface stripping on the porous tube; injector 3 has a pressure swirl atomizer). Figure 3.18 (a) shows the comparison of the spray characteristics between injector 1 and 2. The D_{10} distributions are almost same between injector 1 and 2, although injector 1 has larger D_{10} in the outer regions. D_{32} distribution indicates that injector 2 has smaller D_{32} than injector 1. D_{32} increases steeply for injector 1 beyond radial distance of $0.8D$. This is because injector 1 has less mass flow rate of air going through when compared with injector 2 even though both have same equivalence ratio.

(a) Comparison of diameters

(b) Comparison of volume flux

(c) Comparison of mean axial velocity of spray

(d) Comparison of mean radial velocity of spray

Figure 3.18 Comparison between injector 1 and 2, $\Delta p/p=4\%$, $\phi=0.6$, $Z/D=0.4$

The higher mass flow rate of injector 2 has more energy and therefore the swirl strength lasts much farther downstream of the injector. It should be noted that these measurements are carried out without confinement. Since the amount of the air for the same pressure drop is different between these two injectors, the maximum volume flux is higher for injector 2 as shown in Figure 3.18 (b). Figure 3.18 (c) shows the mean axial velocity of spray for both injector 1 and 2. The magnitude of mean axial velocity of spray for injector 1 is smaller than injector 2 due to the

weakening of the swirl strength at $Z/D=0.4$, as explained before. Figure 3.18 (d) shows the variation of mean radial velocity of spray. As can be observed the radial velocity for injector 2 is higher than injector 1 indicating that the swirl strength is retained for the injector 2 at $Z/D=0.4$ whereas for injector 1 the strength is diminished. The variation is $\sim 20\%$. For both injectors it is observed that the radial velocity is fluctuating in the center of the injector.

The comparison between injector 2 and 3 is shown in Figure 3.19. As shown in Figure 3.19 (a), although the D_{32} is smaller in the outer periphery region for injector 3 when compared with injector 2, it has relatively larger D_{32} in the center of the injector. Both injectors have smaller droplets in the center due to the presence of the CTRZ. Injector 2 has an axial swirler at the base of the injector. The smaller droplets produced due to the surface stripping of the liquid surface on the porous tube is contained and imparted by the axial swirler. In the case of injector 3, the spray formed is by pressure swirl atomizer which tend produce larger droplets. This is confirmed by the axial velocity profiles shown in Figure 3.19 (c). D_{32} increases more rapidly from the center for injector 3 than for injector 2. This is attributed to the nature of atomization process. In the case of injector 3, the fuel jet from the pressure swirl atomizer undergoes a jet breakup which expands in a conical fashion whereas for injector 2, the atomization occurs through airblast atomization over the periphery of the porous tube which provides more annular spray. This is reflected in more broad distribution of D_{32} for injector 2. However, the volume flux distribution for both injectors does not seem to vary as shown in Figure 3.19 (b). The volume flux represents symmetric for both injectors along the center line. The maximum volume flux of both injectors appears at the same location, which is the outer edge of the injector exit. Although the fuel flow rates for two injectors are different at same operating location, the variation is less than 15% therefore the differences are not large enough to be observed from the volume flux distribution. The maximum location is also

the location where D_{32} distribution is stable for both injector. This distribution overlaps with the swirling air flowfield and facilitate the fuel-air mixing performance. Figure 3.19 (c) shows the mean axial velocity of the liquid phase. The positive and reverse flow velocities for injector 2 are higher than injector 3. The difference of the reverse flow velocity magnitude is $\sim 30\%$, which is due to the presence of the axial swirler at the base and the presence of smaller droplets in the center. The radial velocity distribution is identical for both injector 2 and 3, which suggests that the axial swirler and spray formation process does not affect the radial velocity which is mostly controlled by the primary and secondary swirler. The radial velocity variations in the center is due to the measurement being carried out very close to the exit of the injector. These variations fade as the spray moves downstream.

(a) Comparison of D_{10} and D_{32}

(b) Comparison of volume flux of fuel

(c) Comparison of mean axial velocity of spray

(d) Comparison of mean radial velocity of spray

Figure 3.19 Comparison between injector 2 and 3, $\Delta p/p=4\%$, $\phi=0.6$, $Z/D=0.4$

The weighted average D_{32} is calculated for the three injectors. The results are shown in Table 3.1. The operating condition is at 4% pressure drop, equivalence ratio of 0.6, and at $Z/D=0.4$.

	Injector 1	Injector 2	Injector 3
Weighted D_{32} (μm)	45.9	43.6	41.1
Weighted D_{10} (μm)	21.5	16.7	14.7

Table 3.1 Weighted average diameters for the injectors

Injector 1 has the largest D_{32} due to the weaker swirling flow. Injector 3 has the smallest droplets size due to the flow number used in this case. Although the injector 1 and 2 with porous have larger D_{32} , the variation is within 10%. Moreover, the measurements are carried out at atmospheric conditions. At higher static pressure near engine conditions, due to the surface stripping phenomena it is expected that injector 1 and 2 will produce much finer spray with lower D_{32} than injector 3. Further as noticed from D_{32} distribution and spray aerodynamics, the spray from injector 1 and 2 disperses much quicker than injector 3. The emission characteristics of these injectors will be discussed in the next chapter.

3.3 Fuel and Air Mixing Studies

Another important characteristic of the injector 1 and 2 is that they are capable of working with gaseous fuel. To validate this concept, fuel and air mixing studies are carried out on injector 1. For these studies, CO_2 is used as fuel simulant.

Fuel-Air mixing measurements are performed with and without confinement downstream of the injector, respectively. Figure 3.20 (a) shows the fuel mass fraction distribution without confinement at 4% pressure drop and equivalence ratio of 0.7. The equivalence ratio for mixing studies is calculated based on natural gas stoichiometric ratio. The measurements are carried out at $Z/D=0.08$ downstream of the injector. In Figure 3.20 (a), the inner dashed line represents the outer diameter of porous tube, the middle-dashed line represents the tip of venturi, the outer dashed line is the edge of the injector exit. The fuel is injected through the porous tube into the primary swirling air. The fuel rapidly mixes with the primary swirling air. Low fuel concentration is observed in the center region, due to the presence of the CTRZ. Higher fuel concentrations are observed between the annular space of venturi and the flare inner edge. It is also the region with higher mass flux. The primary air interacts with the secondary swirling air downstream of the venturi. From the fuel concentration measurements, it is quite evident that the mixing is rapid as depicted by high concentration between the venturi and the inner flare edge. This is the region where secondary air is interacting with the primary air. The radial distribution of the fuel has a symmetric profile, however, minor circumferential variations in the mass fraction are still observed. This may be attributed to either the quality of the porosity of the porous tube or manufacturing defects in the fuel channel inside the plenum as mentioned in section 3.2.2. As the combustion experiments are carried out with the confinement, the fuel-air mixing studies are also investigated with the confinement. Figure 3.20 (b) represents the fuel concentration distribution with the confinement. As can be observed, the mass fraction distribution is similar to the unconfined case, although higher concentrations are observed in the CTRZ. The higher concentration is caused by the confinement effects as more fuel mass is recirculated into the CTRZ. Low fuel concentration points are also observed at the outer edge of the injector exit, which is due to the cooling air around the

injector flare. The cooling air passes between the injector and the flange to which it is mounted. From the exit of the injector to the wall of the confinement, the fuel mass fraction is uniform due to thorough mixing in the outer recirculation zone (ORZ).

(a) without confinement

(b) with confinement

Figure 3.20 Fuel mass fraction at $Z/D = 0.08$, $\Delta p/p = 4\%$, $\phi = 0.7$

To investigate the mass fraction distribution downstream of the injector, measurements are carried out at two horizontal planes that $Z/D = 0.43$ and 0.75 from the injector exit. The

measurements are carried out at the same operating conditions and equivalence ratio. Figure 3.21 shows the mass fraction profiles along X axis at $Z/D = 0.08, 0.43$ and 0.75 planes. Based on the flame imaging, $Z/D = 0.75$ is the location of maximum heat release (discussed in the next chapter). As can be observed at $Z/D = 0.08$, the fuel concentrations are lower in the center where CTRZ is observed. The concentrations are higher around $X/D = 0.4$ and concentrations taper towards the wall of the confinement. At $Z/D = 0.43$ plane, the maximum fuel mass fraction decreases significantly and move towards outer region. At $Z/D = 0.75$ plane, the fuel mass fraction profile shows a flat profile indicating that the fuel and air are fully mixed. As the flow expands downstream of the injector due to the swirling flow, the axial and radial velocity decreases. Further, the RMS velocities are higher downstream of the injector [84]. The increased turbulence levels therefore enhance the fuel-air mixing.

Figure 3.21 Variation of Y_F at different axial location (with confinement $\Delta p/p=4\%$, $\phi=0.7$)

To study the effect of fuel pressure drop across the porous tube on the fuel-air mixing profile, experiments are carried out at selected equivalence ratios ($\phi=0.6$ and $\phi=0.8$) at a constant pressure drop of 4% across the injector. These experiments are carried out without confinement (without the test section). Figure 3.22 shows the mass fraction profiles at different operating

conditions. As can be observed, as the equivalence ratio increases, the local mass fraction values also increase. However, the profiles look similar with the maximum concentrations around $Z/D = 0.4$ plane for all three cases. To evaluate whether there are any variations in the spatial gradients in the mass fraction, the profiles are normalized with the average mass fraction.

Figure 3.22 Effect of equivalence ratio on Y_F , without confinement, $\Delta p/p=4\%$

Figure 3.23 Effect of equivalence ratio on normalized Y_F , without confinement, $\Delta p/p=4\%$

Figure 3.23 shows the normalized mass fraction at different equivalence ratios. Normalized mass fraction is defined in Equation 3.6,

$$Y_{normalized} = \frac{Y_{local}}{Y_{avg}} \quad (3.6)$$

where Y_{local} is local fuel mass fraction, Y_{avg} is the average of fuel mass fraction at each condition. This shows that the injector performance at various fuel flow rates provide same fuel-air distribution and that the change in fuel flow rates do not affect the mass fraction distribution. The peak of normalized fuel mass fraction is around 1.55~1.62 for all cases, and the uncertainty between the cases is approximately 2%.

To quantify the quality of fuel-air mixing, Degree of Unmixedness (D.U.) is calculated. Typically, D.U. is used to compare the mixing performance of the different injectors or design variations of a particular type of injector at nominal operating conditions. The lower the value of D.U. the better is the mixing. In the present study, though only one injector design configuration is used, it is pertinent to calculate D.U. so that the performance can be compared with future design variations of the injector. D. U is calculated for confined and unconfined cases. It is defined as:

$$D. U. = \frac{Y_{rms}^2}{Y_{mean}(1-Y_{mean})} \quad (3.7)$$

where Y_{mean} is the mean mass fraction in the plane of measurements, and Y_{rms} is defined in following equation:

$$Y_{rms} = \sqrt{\frac{\sum_{i=1}^n (Y_i - Y_{mean})^2}{n-1}} \quad (3.8)$$

where Y_i is the local mass fraction, and n is the total number of the measuring points in the plane. D.U. value of zero implies that the fuel and air are fully premixed [86]. D. U is calculated at an equivalence ratio of 0.7. The calculated D.U. for unconfined case is 0.25 at the exit of the injector.

The aerodynamics measurements, the spray characterization and the fuel-air mixing studies show that the injectors with porous tube perform on par with conventional injectors if not better. The next chapter discusses the combustion characteristics of these injectors.

Chapter 4 Reacting Flow Measurements

In this chapter, reacting flow measurements are carried out on the all three injectors at various equivalence ratio and inlet air temperature conditions. All cases are at atmospheric pressure and the pressure drop across the injector is maintained at 4%. The flame behavior and emission characteristics are investigated in this chapter. The main purpose of combustion studies is to compare the emission performance between airblast atomization (injector 2) and pressure swirl atomization (injector 3) on similar sized injectors. Injector 1 is used for testing the proof of the concept of using the same injector for gaseous and liquid fuels. Jet-A, 100% GTL, 50%-50% GTL Jet-A blend and natural gas are used as fuels during the testing program. Chemkin simulations are carried out to compare the predictions with experimental results.

4.1 Flame Characterization

Flame characterization is carried out to estimate the flame length and maximum heat release zone. The difference in the general shape of the flame and width of the flame is almost similar for injectors 1, 2 and 3. Though there are minor variations in flame shape-for 1D analysis is Chemkin where only the length of the flame is needed for residence time estimations-the variations among the injectors is not significant. However, as the mass flow rates are higher for injector 2 and 3, the mean velocity is higher than injector 1 and hence the residence time is shorter.

Figure 4.1 shows flame structure of injector 2 from downstream of the injector at different inlet air temperature conditions without filters. A hollow cone flame can be observed from both conditions, with light blue flame around the injector exit. This flame structure confirms the hollow cone spray distribution as explained in Chapter 3. The front side of the flame is brighter than other sides in both conditions. This is due to the unequal fuel distribution caused by the lack of the homogeneous porosity of the porous material surface and the manufacturing defects. The effects

of inlet air temperature on the flame structure can also be exemplified in Figure 4.1. At 477 K (400 °F), the flame extended readily to the downstream; at 588K (600°F), the flame is brighter, shorter and narrower. This is because of the insufficient temperature for the fuel vaporization, and insufficient fuel-air mixing in the flame zone. As inlet air temperature is increased, the fuel-air mixing can be improved leading to reduction in NO_x emissions.

Figure 4.1 Flame visualization for injector 2 from downstream of the injector, $\phi=0.7$

Figure 4.2 shows flame images taken with CH* radical filter for injector 1. The heat release and cumulative distribution curves are shown at various operating conditions for natural gas flame. In the present experiments, the pixel intensity in these images from the camera is proportional to the heat release [89]. An in-house algorithm is written in Matlab to track the flame zone. The flame images are integrated across the flame width to quantify the variation of heat release along the axial distance. In the present studies, the heat release and the cumulative heat release along the axial distance are quantified. The left edge of the image is the location of the exit of the injector. The black solid line shows the heat release rate distribution downstream of the injector, and red solid line represents cumulative heat release distribution. A symmetric flame is observed in all

four cases. The maximum heat release occurs at $\sim Z/D=0.75$ from the injector exit. As can be observed, from Figure 4.2 (a) and (b), the flame shape is identical with minor variations in the flame length at two different preheat temperatures. Figure 4.2 (c) and (d) shows the flame liftoff is higher at lower equivalence ratios. Though the flame liftoff is not quantified in the present study, it is observed that flame liftoff is proportional to the equivalence ratio. Further it is observed that the width and length of the flame decrease with the increase in the inlet air temperature and equivalence ratio.

Figure 4.2 Chemiluminescence Imaging at different operating conditions

Figure 4.3 shows the effect of equivalence ratio on the normalized heat release along the axial distance for the flames shown in Figure 4.2. The equivalence ratio is varied from 0.6 to 0.9 at constant inlet air temperature of 533K (500°F). As equivalence ratio is increased, the location of maximum heat release shifts towards the injector exit. However, the location of maximum heat release is not affected significantly at higher equivalence ratios. The maximum heat release is $\sim Z/D=0.75$ downstream of the injector exit.

Figure 4.3 Effect of equivalence ratio on normalized heat release at $T_3 = 533$ K (500°F)

Figure 4.4 shows the effect of inlet air temperature on the heat release distribution along the length of the combustor at a constant equivalence ratio of 0.7. As the inlet temperature is increased, the maximum heat release curve shifts toward the injector exit indicating that the reaction zones are shorter. Though the location of the maximum heat release is not affected significantly, the gradients in the heat release vary along the length of the combustor. As the inlet air temperature is increased, the heat release distribution tapers quickly beyond the location of the maximum heat release. This indicates that the reaction rates are much faster with increase in inlet

air temperature. The effect of equivalence ratio on the heat release distribution is higher than the inlet air temperature. This is evident in the flame length measurements.

Figure 4.4 Effect of inlet air temperature on normalized heat release distribution at constant equivalence ratio

Figure 4.5 shows the effect of equivalence ratio on the flame length for various inlet air temperatures. The flame length is defined as 90% of the cumulative heat release. As the inlet air temperature is increased, the flame length is shorter at any equivalence ratio. The effect of inlet air temperature on the flame length decreases as the equivalence ratio is increased. The flame length is shorter as the equivalence ratio is increased. The variations in the flame length tapers as the equivalence ratio is increased. This indicates that the residence time of the species in the flame zone is relatively constant at higher equivalence ratio and inlet air temperature but at lower equivalence ratio the variations in the residence time needs to be considered during simulations.

Figure 4.5 Effect of equivalence ratio on flame length at different inlet air temperatures

4.2 Emission Characterization

One of the most considerable emissions under discussion is NO_x , which includes NO and NO_2 . The effects of the inlet air temperature, flame temperature and fuel type on the NO_x emissions are discussed. The NO_x concentration is corrected to 15% O_2 (dry sample), which is convenient for comparison. Emission index is also investigated and used for comparison with the previous results of LPP combustors.

To ensure the accuracy of the experiments and measurements operating at expected equivalence ratio, the O_2 concentrations in the products is compared between theoretical and experimental O_2 concentrations. In the first method, the O_2 concentration is calculated based on the measured equivalence ratio, which is obtained from the mass flow rates of fuel and air. Then the equivalence ratio is used to calculate O_2 concentration in the products by assuming that the combustion process is complete. The calculation is shown below. The overall equilibrium equation is written in terms of equivalence ratio as shown below:

where x and y represent the number of the atom of C and H in the hydrocarbon fuel, respectively. ϕ represents reaction initial equivalence ratio, the reaction is considered as fuel lean condition. The mole fraction of O_2 from this equation should match the measured O_2 from the exhaust sample. Since the sample obtained from the emission analyzer is a dry sample, the water content is removed in the equation and O_2 concentration is obtained in terms of ϕ as follow:

$$X_{O_2} = \frac{(1-\phi) \cdot (x + \frac{y}{4})}{x \cdot \phi + (4.76 - \phi) \cdot (x + \frac{y}{4})} \quad (4.1)$$

where X_{O_2} is the O_2 concentration in the products (theoretical calculations). For gaseous fuel combustion, as natural gas is used, the fuel composition used for theoretical O_2 calculation is 92% methane and 8% ethane. For Jet-A fuel combustion, dodecane is used as surrogate for X_{O_2} calculation. This is valid as the molar weight of Jet-A fuel is around 170, which is close to dodecane ($C_{12}H_{26}$). The other way to acquire the O_2 concentration in the dry products is from the NDIR gas analyzer.

Figure 4.6 represents the comparison of O_2 concentration between these two methods in a natural gas combustion experiment. The black line represents the X_{O_2} calculated from the complete combustion reaction as explained above. The markers represent the X_{O_2} obtained from the gas analyzer. Both results depend on dry sample where the water content is omitted. The markers fall on the line although the values are slightly higher at higher equivalence ratio conditions. This is because in theoretical calculations it is assumed that the mean mass flow rate of fuel and air are constant. However, they change during data acquisition of exhaust gas sampling. The variation between theoretical and experimental O_2 concentrations is less than 10%.

Figure 4.6 Comparison between measured and calculated X_{O_2} for natural gas

4.2.1 Localized Emission Measurements

Localized NO_x measurements are carried out based on selected sampling locations inside the combustor. This is carried out to analyze the symmetry of the flame and the emission distribution. Global NO_x results are calculated to study the effect of inlet air temperature and flame temperature.

Localized emissions are obtained by moving the sampling probe in the combustor in a horizontal plane. As mentioned in the earlier chapter, the fuel distribution of porous material is slightly asymmetric due to the uneven fuel distribution on the inner surface of the porous material. To investigate the emission distribution and analyze the relation between the localized and global emissions, 25 measurement points are selected in a horizontal plane. NO_x emission distribution during combustion of Jet-A fuel in injector 1 is shown in Figure 4.7. The operating conditions are equivalence ratio of 0.7, inlet air temperature of 400°F and pressure drop of 4% across the injector. NO_x distribution is asymmetric with higher concentration on the left top side as expected, and the variation is ~25%. There are several factors that affect the symmetry of the NO_x distribution, such as non-uniform fuel distribution, inconsistent fuel-air mixing qualities, and variable amount of the

cooling air around the combustor. Besides the uneven fuel distribution as mentioned earlier, the cooling air is assigned unequally on all four walls which can cause the nonuniform NO_x distribution. The global emissions are calculated based on the weighted area of the 25 measured points grid and is referred to as the weighted average emissions. Equation 4.2 represents the global emissions calculation.

$$X_{global} = \frac{\sum X_i \cdot A_i}{\sum A_i} \quad (4.2)$$

where X_{global} is the global species concentration, X_i is the localized species concentration and A_i is the local area. It is found out that the calculated global emission concentration based on 25 points is nearly the same as the global emission concentration on the 5 points in the center, which are marked out with circle. The remaining emission measurements are carried out at the five center points to simplify the emission measurements.

Figure 4.7 NO_x distribution of injector 1 at $Z/D = 6$ horizontal plane, $\Delta p/p = 4\%$, $\phi = 0.7$, $T_3 = 400^\circ\text{F}$

Figure 4.8 NO_x distribution of injector 2 at Z/D =6 horizontal plane, $\Delta p/p=4\%$, $\phi=0.6$, $T_3=400^\circ\text{F}$

The localized NO_x measurement is also carried out on injector 2, the distribution of NO_x corrected to 15% O₂ during combustion of Jet-A fuel is shown in Figure 4.8. Due to the same fuel feed system and same experimental setup used, higher concentrations are observed at the top-left side of the plot, lower concentrations are observed at the bottom of the plot. The contour plot is similar to the plot in Figure 4.7. The distribution is uniform in the center since injector 2 has a strong CTRZ. The global NO_x, which is weighted average NO_x is also calculated based on the 25-points measurement grid. A five-point measurement grid is also used to represent the global NO_x at the combustor exit. The variation is ~25%. It is also evident that injector 1 has a lower NO_x distribution compared to injector 2 even though the equivalence ratio of injector 2 is 0.6. This phenomenon is discussed in the following section.

4.2.2 Effects of Inlet Air Temperature on NO_x Emissions

Calculated global NO_x emissions are used to investigate the effects of inlet air temperature on NO_x emissions. Figure 4.9 shows the NO_x emissions at various operating conditions for injector 1 with Jet-A fuel. The flame temperature referred here is the adiabatic flame temperature, which

is determined by the equivalence ratio and inlet air temperature at each condition. As can be observed from the plot, NO_x increases with increase in the inlet air temperature and the adiabatic flame temperature. Increasing the inlet air temperature leads to increase in the rate of reactions, which is one the most important factors to accelerate the NO formation. The reactions will achieve the equilibrium faster. As the adiabatic temperature increases, the formation of NO_x tends from prompt NO_x towards thermal NO_x . Prompt NO_x is produced under lower temperature, which is normally lower than 1900K [90], and at shorter residence time. Increasing the equivalence ratio leads to the increase in the flame temperature and restrain the prompt NO_x formation. However, higher temperatures are conducive for thermal NO_x formation. Thermal NO_x is very significant compared to prompt NO_x , and the rate of increase in NO_x is higher for thermal NO_x formation than prompt NO_x formation. Therefore, NO_x increases faster at higher adiabatic temperature conditions.

Figure 4.9 NO_x corrected to 15% O_2 at various operation conditions, injector 1

The effect of inlet air temperature on NO_x are also investigated for injector 2 and 3, and an interesting and unexpected phenomenon is observed, which is not observed for injector 1. Figure 4.10 and Figure 4.11 show the NO_x for Jet-A fuel for injector 2 and 3, respectively. Since injector 2 and 3 have higher air flow rate at 4% pressure drop than injector 1, the fuel flow rates at

corresponding equivalence ratio conditions are higher. The inlet air temperature varies from 477K (400°F) to 644K (700 °F).

Figure 4.10 NO_x corrected to 15% O₂ at various operation conditions, injector 2

Figure 4.11 NO_x corrected to 15% O₂ at various operation conditions, injector 3

In comparison to injector 1, the NO_x produced by injector 2 and 3 decreases as inlet air temperature increases from 477K (400°F) to 533K (500°F). Furthermore, NO_x for injector 3 keeps decreasing as inlet air temperature increasing from 533K (500°F) to 644K (700°F). Anderson [56] also noticed the same phenomena on a tested LDI combustor. It was reported in their measurements

that the NO_x decreases as the inlet air temperature varies from 1100K to 1250K at 2.5 atm. It was summarized that slow and incomplete evaporation at lower inlet air temperature condition accounts for this phenomenon. As the inlet air temperature increases from 400°F to ~600°F, the rate of evaporation increases. This in turn improves the fuel-air mixing quality. Ahmed [91] also observed the phenomena at 2 atm. The inlet air temperature was varied in their study from 477K (400°F) to 533K (500°F). The effect of static pressure on the NO_x formation was also investigated in their study. It was reported that the higher static pressure in the combustor will improve the atomization, increases the rate of evaporation and reaction rates. This leads to increase in NO_x formation with increase in inlet air temperature. These studies strongly agree with the results presented here on injector 2 and 3, although their results don't agree with injector 1 data. This is due to the lower fuel flow rates in injector 1. For the same pressure drop across the injector, the liquid film thickness is thinner on the porous material for injector 1. This leads to a better atomization. Moreover, the fuel temperature is higher as the porous temperature will be higher which in turn reduces the viscosity. The decrease in viscosity will also improve atomization and evaporation rate of the fuel.

4.2.3 Effects of Fuel Type on NO_x

One of the significant requirements for the concept of this injector is the fuel flexibility. Various fuels can be used in this injector. To study the effect of fuel flexibility, both liquid and gaseous fuels are tested in injector 1. Also, multiple liquid fuels are tested in injector 2 and 3.

Figure 4.12 shows NO_x emission in ppmvd at various operating conditions for both gaseous and liquid fuels for injector 1. Natural gas and Jet-A are used. The inlet air temperature is varied from 433K (400°F) to 560K (550°F), and the equivalence ratio from 0.6 to 0.8. As can be observed the NO_x emissions increase with flame temperature for natural gas as expected. It is also noted that

NO_x emissions are higher for Jet-A fuel when compared to natural gas combustion. The overall NO_x is ~ 2 ppm for natural gas and ~ 5 ppm for Jet-A fuel combustion at ~1900K adiabatic flame temperature. The trend line for both fuels are also plotted. It is evident that the rate of increase in NO_x with flame temperature is higher for Jet-A fuel than for natural gas. This is due to the fuel physical and chemical properties. Gaseous fuel normally produces less NO_x when compared with liquid fuels. This is because gaseous fuels readily mix whereas in case of liquid fuels the fuel should undergo atomization and evaporation. This leads to pockets of rich and stoichiometric zones leading to higher NO_x and CO. Moreover, Jet-A fuel has higher C/H ratio and aromatic compounds that produces more NO_x.

Figure 4.12 Comparison of NO_x between different fuels, injector 1, $\Delta p/p=4\%$

One of the most noteworthy features is the effect of the various liquid fuels on NO_x formation. Blended fuel used in the measurements has 50 % GTL fuel mixed with 50% Jet-A fuel by volume. Since the GTL fuel and Jet-A fuel have similar C/H ratio, it is assumed that the stoichiometric air-fuel ratios of GTL and blended fuel are the same with Jet-A fuel by ignoring small variations. Although the C/H ratio is almost identical between these two fuels, the compounds are significantly different. GTL fuel is derived from natural gas and mainly consists

of n-alkanes and iso-alkanes. The percentage of total alkanes in GTL fuel is up to 99.5%, on the contrary, the percentage of total alkanes in Jet-A fuel is around 80%. Unlike GTL fuel, Jet-A fuel mainly contains cycloalkanes and around 28% of total alkanes.

Figure 4.13 and Figure 4.14 show the NO_x emission produced by various fuels in injector 2 at inlet air temperature of 477K (400°F) and 644K (700°F), respectively. The equivalence ratio is varied from 0.55 to 0.8. Jet-A fuel is the baseline, the differences among GTL fuel, blended fuel and the baseline are evident. As can be observed in Figure 4.13, at 477K inlet air temperature condition, GTL fuel produces less NO_x emissions than Jet-A fuel by 10~15%; and blended fuel produces 5~10% less NO_x than Jet-A fuel. At $T_3 = 644\text{K}$ (700°F), shown in Figure 4.14, the difference in the NO_x values among the fuels is reduced. Although blended fuel produces less NO_x than GTL and Jet-A fuel for all conditions, the difference in the NO_x values falls within the realm of experimental error and can be ignored.

Figure 4.13 Comparison of NO_x among different liquid fuels, injector 2, $\Delta p/p=4\%$, $T_3=477\text{K}$

Figure 4.14 Comparison of NO_x among different liquid fuels, injector 2, $\Delta p/p=4\%$, $T_3=644\text{K}$

To further investigate the fuel effects, the correlation between the adiabatic flame temperature and NO_x emissions is discussed in equation 4.3,

$$NO_x = A \cdot \exp\left(\frac{B}{T_{flame}}\right) \quad (4.3)$$

where T_{flame} is the adiabatic flame temperature, the coefficients of A and B are for each fuel for an injector. Regression analysis is carried out to obtain the constants. Table 4.1 shows the constants obtained for injector 2 and 3 based on the experimental data points.

As shown in Table 4.1, at $T_3=644\text{K}$ (700°F) condition, the constant B of the blended fuel is the smallest for injectors. This indicates that blended fuel has higher rate of NO_x formation with adiabatic flame temperature. However, the constant A is smallest for GTL fuel both injectors at $T_3=644\text{K}$ (700°F). This indicates that GTL produces less NO_x at lower adiabatic flame temperature conditions. The differences of NO_x formation among all injectors are not significant and within 15%.

T_3	Injector	Fuel type	A	B (K^{-1})	Temperature range (K)
533K (500°F)	#2	Jet-A	441529	-22358	1850~2250
		GTL	80679	-19194	1850~2250
		Blended	137201	-20156	1850~2250
	#3	Jet-A	13792	-14515	1850~2250
		GTL	11015	-14094	1850~2250
		Blended	19916	-15362	1850~2250
644K (700°F)	#2	Jet-A	42993	-16557	1850~2250
		GTL	41880	-16518	1850~2250
		Blended	44624	-16734	1850~2250
	#3	Jet-A	20979	-15034	1850~2250
		GTL	10084	-13560	1850~2250
		Blended	26926	-15623	1850~2250

Table 4.1 NO_x concentration curve fit coefficients (ppm, dry sample corrected to 15% O₂)

4.2.4 Effect of Injector Configuration on NO_x formation

Due to the different fuel feeding system and swirler used in the studies, it is pertinent to discuss the effects of the injector configuration on NO_x formation.

Figure 4.15 shows the comparison of NO_x between injector 1 and 2 for Jet-A fuel. The inlet air temperature is varied from 400 to 600°F, and equivalence ratio from 0.55 to 0.6. For injector 1, NO_x emissions at 477K (400°F) inlet air temperature is 50% less than injector 2. The reason could be due to the fuel preheating in the fuel plenum leading to faster evaporation of the fuel. The size of the fuel plenum is same for both injector 1 and 2. Even though the cold flow spray measurements show that the SMD for injector 1 is higher than injector 2, the fuel flow rate for the injector 1 is much lower than injector 2. This leads to higher fuel temperature for injector 1 at the time of atomization. The increase in fuel temperature decreases the viscosity of the fuel leading to better atomization, faster evaporation of the fuel and quick fuel-air mixing.

However, as the inlet air temperature increases to 533K (500°F), the NO_x emissions produced by injector 1 exceeds injector 2. As the air temperature increases, the effect of fuel temperature decreases. As stated earlier, the SMD of injector 2 is smaller than injector 1. The smaller SMD improves rate of evaporation and increases better fuel-air mixing. However, the major factor that could be affecting the decrease in NO_x for injector 2 is the residence time and turbulent kinetic energy. Injector 2 has higher flow rates therefore the residence time is shorter. Moreover, the aerodynamics measurements show that injector 2 has higher TKE. The higher TKE enhances better fuel-air mixing.

Figure 4.15 Comparison of NO_x between injector 1 and 2, $\Delta p/p=4\%$, Jet-A fuel

Figure 4.16 and Figure 4.17 represent the comparison of NO_x between injector 2 (porous tube) and 3 (pressure swirl atomizer), Jet-A and GTL fuels are used respectively. The inlet air temperature varies from 400°F to 700°F, equivalence ratio from 0.55 to 0.6. In Figure 4.16, it is apparent that injector 3 produces more NO_x than injector 2 in the range of inlet temperatures tested. This is because that injector 2 has much better evaporation than injector 3, as explained before due to the fuel plenum effect, even though it has higher SMD than injector 3 at room temperature (as discussed in Chapter 3). The effect of evaporation of droplets on NO_x emissions is far more

significant than atomization at the inlet temperatures tested. However, the difference in the NO_x formation decreases at higher adiabatic flame temperatures indicating that the atomization mechanism is not relevant at these operating conditions. The difference in NO_x levels falls within the experimental error. However, it is well known that airblast atomization has a much better spray dispersion and fuel-air mixing. The emission performance of GTL fuel is similar to Jet-A fuel as shown in Figure 4.17.

Figure 4.16 Comparison of NO_x between injector 2 and 3, $\Delta p/p=4\%$, Jet-A fuel

Figure 4.17 Comparison of NO_x between injector 2 and 3, $\Delta p/p=4\%$, GTL fuel

4.2.5 Comparison with Other Combustor Concepts

To characterize the performance of the porous injector, emission index of NO_x is calculated. Emission index is a parameter that is used to evaluate and compare the behavior of combustors. As mentioned earlier, since Jet-A is a combination of range of hydrocarbons and the molar weight is similar to dodecane (C₁₂H₂₆). Therefore, dodecane is used for the emission index calculations. Emission index calculation is shown in equation 4.4.

$$EINO_x = \frac{m_{NO_x}}{m_{fuel}} = \left(\frac{X_{NO_x}}{X_{CO} + X_{CO_2}} \right) \left(\frac{x MW_{NO_x}}{MW_F} \right) \cdot 1000 \quad (4.4)$$

where x is the number of moles of carbon in a mole of fuel (C₁₂H₂₆), MW is the molar weight, X_{NO_x}, X_{CO}, X_{CO₂} represent the concentration of individual species in the products. In this case, all species concentrations are calculated based on dry sample. The unit of calculated EINO_x is g/kg which indicates the gram of the NO_x produced per kilogram of fuel burnt. Since NO and NO₂ is measured separately in the measurements, the molar weight of NO_x can be calculated from the following equation:

$$MW_{NO_x} = \frac{30 \cdot X_{NO} + 44 \cdot X_{NO_2}}{X_{NO} + X_{NO_2}} \quad (4.5)$$

X_{NO_x} and X_{CO} can be obtained from gas analyzers. The molar weight of the fuel is 170. By ignoring the concentration of CO and unburned hydrocarbons, the CO₂ concentration (X_{CO₂}) in the products can be obtained from the equation discussed in section 4.1, the calculation is expressed in equation 4.6,

$$X_{CO_2} = \frac{x \cdot \phi}{x \cdot \phi + (4.76 - \phi) \cdot (x + \frac{y}{4})} \quad (4.6)$$

where x and y are the numbers of moles of carbon and hydrogen in a mole of fuel (C₁₂H₂₆), respectively, and φ is the equivalence ratio obtained from O₂ concentration (X_{O₂}). The calculation for O₂ concentration (X_{O₂}) is shown in equation 4.7.

$$\varphi = \frac{(1-4.76 \cdot X_{O_2}) \cdot (x + \frac{y}{4})}{x \cdot X_{O_2} + (1-X_{O_2}) \cdot (x + \frac{y}{4})} \quad (4.7)$$

The equivalence ratio is obtained from O₂ concentration in the products rather than the readings from fuel and air flow meter.

The emission index of NO_x for injector 1, 2 and 3 are compared with LPPs combustors. Tacina [60] summarized the emission results for multiple LPP concepts with various fuels and plotted the data of EINO_x with adiabatic flame temperature. The LPP data was acquired from research work of Anderson[56], Semerjian[62], Cooper and Tang. The EINO_x data of the injectors in the current study are plotted along with the data from Tacina [60] in Figure 4.18. In the data presented in the graph, Anderson used propane, Semerjian and Cooper used Jet-A, and Tang used Hexane. The NO_x results of Jet-A for injector 2 and 3 are compared with LPP data. NO_x results of natural gas for injector 1 are plotted in Figure 4.18 even though natural gas combustion is not LPP combustion. The effect of static pressure and the fuel used are significant and cannot be overlooked. Therefore, the result from the present investigation are compared with Semerjian's results which was carried out with Jet-A fuel at atmospheric conditions. It is apparent that EINO_x of injector 2 and 3 at 644 K (700°F) inlet air temperature is close to Semerjian's results, although Semerjian had higher inlet air temperature. Semerjian also investigated the residence time effects on the NO_x emission, which is discussed in detail in the next section. EINO_x at 588K is slightly lower than the EINO_x at higher inlet air temperature as expected. As can be observed, EINO_x acquired from natural gas in injector 1 are the lowest values in this chart as it is a gaseous fuel.

Figure 4.18 NO_x comparison for the injector 1, 2, 3 and Tacina's [60] complied results of LPPs combustors

4.3 Numerical Simulations of NG, Jet-A and GTL Fuel Combustion

4.3.1 Reaction Mechanism

So far experimental results were presented. In this section a brief discussion on numerical predictions of NO_x emissions are discussed. This is carried out so that the cold flow mixing measurements can be used to predict the emissions. If this is successful, then numerical simulations can be used to predict the emissions at engine conditions. Chemkin Pro was used to investigate the emission performance for each injector. GRI mech 3.0 [95] was used to simulate natural gas combustion. NASA code[92] was used to simulate Jet-A fuel and CNRS code[94] was used to simulate GTL and blended fuels for the estimation of NO_x emissions. The simulations are used to

analyze the dominant chemical reactions and their reaction rates and compare simulations with experimental results.

The NASA reaction mechanism [92] used in this study for Jet-A fuel was developed based on three-component surrogate mixtures for Jet-A. The surrogate mixture contains decane (72.7% wt.), benzene (18.2 % wt.) and hexane (9.1% wt.) and are represented by a single compound $C_{11}H_{21}$. This mechanism involves 51 species and 242 reactions. For the studies on simulations of oxidation of a 100% GTL and a 50-50 mixture of GTL and Jet-A, CNRS [94] mechanism was used under the same initial experimental conditions (temperature, 477~644K (400-700°F); pressure, 1 atm; equivalence ratio, 0.5-0.8). The chemical kinetic model consists of 6228 reactions involving 2006 species was used. It is based on previous chemical mechanisms for the oxidation of n-decane, gasoline, and kerosene in similar conditions. This reaction mechanism was developed for synthetic kerosene (SK) surrogate model fuels [94]. The composition of model fuels utilized in this study is different from GTL and blended fuels as shown in Table 4.2. SK represents GTL surrogate fuel, and SK-Jet-A represents blended surrogate fuel in the following section.

Fuel	n-decane	propylbenzene	iso-octane	propylcyclohexane
SK	48.79%	0.47%	50.74%	0.00%
SK-Jet-A	57.66%	9.6%	26.43%	6.3%

Table 4.2 Composition of model fuels in the study for GTL and blended fuels (mole fraction)[94]

The NO formation reactions are added to CNRS reaction mechanism to estimate NO_x emissions. The NO_x formation reactions are obtained from GRI Mech 3.0[95] represented by Reaction 1 to 27 in Table 4.3. These 27 NO formation reaction mechanism are obtained from GRI mech 3.0. The 27 unique reactions include all the required NO_x formation mechanism. The pre-exponential factor A, reaction order and activation energy E_A are shown in the table. There are four

pathways for NO formation in the table and two of them are the primary pathways. Reactions 1-3 represent thermal NO reaction mechanism, in this mechanism, the parameters have a significant dependence on the temperature, which contributes most of the NO emissions above 1900 K of the flame temperature. The other important NO formation pathway is prompt NO mechanism represented in Reactions 2-3 and 18-27. In this mechanism, NO is formed rapidly under fuel lean conditions. Two less important pathways are N₂O pathway, explained in Reactions 4-9 and NNH pathway manifested in Reactions 2-3 and 10-20.

#	Reactants		Direction	Products			A	b	E _A
1	N	NO	←	N ₂	O		2.70E+13	0	355
2	N	O ₂	→	NO	O		9.00E+09	1	6500
3	N	OH	→	NO	H		3.36E+13	0	385
4	N ₂ O	O	→	N ₂	O ₂		1.40E+12	0	10810
5	N ₂ O	O	→	NO	NO		2.90E+13	0	23150
6	N ₂ O	H	→	N ₂	OH		3.87E+14	0	18880
7	N ₂ O	OH	→	N ₂	HO ₂		2.00E+12	0	21060
8	N ₂ O	M	↔	N ₂	O	M	7.91E+10	0	56020
9	NH	NO	←	N ₂ O	H		3.65E+14	-0.45	0
10	NNH		←	N ₂	H		3.30E+08	0	0
11	NNH	M	←	N ₂	H	M	1.30E+14	-0.11	4980
12	NNH	O ₂	←	HO ₂	N ₂		5.00E+12	0	0
13	NNH	O	←	OH	N ₂		2.50E+13	0	0
14	NNH	H	←	H ₂	N ₂		5.00E+13	0	0
15	NNH	OH	←	H ₂ O	N ₂		2.00+E13	0	0
16	NNH	CH ₃	←	CH ₄	N ₂		2.50E+13	0	0
17	NNH	O	→	NH	NO		7.00E+13	0	0
18	NH	OH	→	N	H ₂ O		2.00E+09	1.2	0
19	NH	O	→	NO	H		4.00E+13	0	0
20	NH	O ₂	→	NO	OH		1.28E+06	1.5	100
21	CH	N ₂	→	HCN	N		3.12E+09	0.88	20130
22	HCN	O	→	NCO	H	CO	2.03E+04	2.64	4980
23	NCO	O	→	NO	CO		2.35E+13	0	0

24	NCO	OH	→	NO	H		2.50E+12	0	0
25	NCO	O ₂	→	NO	CO ₂		2.00E+12	0	20000
26	HCN	O	→	NH	CO		5.07E+03	2.64	4980
27	NCO	H	→	NH	CO		5.40E+13	0	0

Table 4.3 NO formation pathway [93] (27 reactions from GRI mech 3.0[95])

In this study, the experimental results are compared with numerical results by measuring the total amount of NO_x. Since it is difficult to define the NO_x pathways in the experiments, numerical study is used to determine and identify the NO_x formation route and the contribution of each route under different conditions.

4.3.2 Premixed Combustion Model

The network model used in Chemkin Pro is a perfectly stirred reactor (PSR) followed by a plug flow reactor (PFR), the network model is shown in Figure 4.19. In the current Chemkin simulations, a fully premixed combustion model is adopted with a perfectly stirred reactor. In the present studies, the fuel-air mixing is not fully premixed at the exit of the injector. However, at Z/D=0.4 downstream of the injector the fuel-air mixing is fully premixed. This is the region where maximum heat release occurs. By assuming a perfectly premixed model in the current simulations, it is expected that the error is marginal. Moreover, the simulations are used for comparing different injectors and fuels.

PSR is a 0-D reactor and used to simulate the flame zone. PSR is solved from the energy equation based on conservation of energy. By assuming it as an energy equation problem, rather than a fixed gas temperature problem, the reactor can undergo heat loss and temperature change. The probe location for measuring emissions in the products is in the post flame zone. A PFR is used to represent post flame zone. PFR is a 1-D reactor and solved with energy conservation equation as well. Heat loss and temperature change are taken into account in PFR.

Figure 4.19 Chemkin Pro diagram view for premixed combustion model

4.3.2.1 Residence time in the model

The inputs for this model depend on the operating conditions in the experiments. The required inputs are the inlet reactant temperature, mass flow rate and equivalence ratio. These parameters are variables. The pressure is maintained at 1 atm. Since the volume of the PSR is indeterminable in the practical experiments due to distributed heat release, the residence time in the PSR must be provided. The total residence time is determined by the residence time in PSR and PFR, which is represented in equation 4.8.

$$t_{residence} = t_{PSR} + t_{PFR} \quad (4.8)$$

The residence time in PSR is estimated by the aerodynamics and flame characterization. The residence time in PFR is determined by Chemkin based on the output from the PSR and the length of the combustor. In this study, the length of the PFR is obtained from the measurement location of the emission probe.

The residence time in the PSR is the total time required for fuel-air mixing (t_{mixing}) and the time required for reactions ($t_{reacting}$) to complete. For liquid applications, the residence time in PSR also includes evaporation of the liquid droplets ($t_{evaporation}$). This is given as

$$t_{combustion} = t_{evaporation} + t_{mixing} + t_{reacting} \quad (4.9)$$

Under low inlet air temperature conditions (lower than 600°F), the evaporation time cannot be ignored as discussed in 4.2.2.

4.3.2.2 Heat loss

In the practical experiments cooling air is provided around the combustor to keep the combustor walls at a nominal working temperature. The combustion occurs under non-adiabatic conditions due to heat losses from the combustor walls. Therefore, heat losses are considered in the Chemkin model as shown in Figure 4.20.

Figure 4.20 Heat loss model description

The heat loss represented in the figure includes radiation losses from the gas, convection and conduction losses from inner and outer walls. The amount of heat loss depends on the operating conditions and the amount of the cooling air used. In the PSR model, the heat loss is calculated based on the measured temperature of the cooling air and the temperature of the combustor wall, the inner and outer wall surface area of the combustor, and the heat transfer coefficient at the wall. In the PFR model also the heat loss is quantified along the length of the combustor and heat transfer coefficient is calculated.

4.3.3 Chemkin Prediction for Gaseous and Liquid Fuel

4.3.3.1 Gaseous Fuel Simulations

Natural gas combustion was carried out for injector 1 only. This is carried out to check the feasibility of dual fuel combustion in a porous injector. Chemkin simulations with GRI Mech 3.0 was carried out for injector 1 only and the results were compared with experimental results of injector 1. Figure 4.21 shows the comparison of NO_x emissions between experiments and simulations by Chemkin Pro. The data points on the graph are the experimental results, the dashed line represents the simulation with GRI Mech 3.0. Both simulations and experimental show that the NO_x emissions increase with adiabatic flame temperature which is expected. As can be observed, the simulations by GRI Mech 3.0 closely matches with the experimental data at temperature below 1900 K. This illustrates that for this application, the GRI Mech 3.0 NO_x reaction mechanism is modeled very well for natural gas application under 1900 K flame temperature. The rate of increase of NO_x emissions is nearly constant until 1900 K in both simulations and experiments. At temperature below 1900 K, the NO_x formation is usually attributed to the prompt NO_x mechanism. Above 1900 K, the effect of adiabatic flame temperature on the NO_x formation is significant as can be noted in Chemkin simulations which is attributed to the thermal NO_x . However, Chemkin over predicts the NO_x emissions at flame temperature over than 1900 K, it is apparent that the increased rate of NO_x emissions in simulations is higher than the experiments. This could be due to a couple of reasons. The residence time used in the PSR is based on mean velocity in the flame zone, which is a crude estimate. The other reason could the heat losses from the combustor walls are under estimated. Similar deviations between simulations and experimental data was observed by Elkady [90].

Figure 4.21 Comparison of NO_x between simulation and experimental results (Natural gas)

To find the sources of NO_x, a detailed study was carried out to understand the reaction paths for NO_x formation. As explained earlier and in chapter 1, NO_x is formed mainly by two mechanisms viz, thermal NO_x and Prompt NO_x. It is important to know the amount of NO_x formed by each mechanism. The contribution of thermal NO to the total NO formation at different equivalence ratio conditions are shown in Figure 4.22. These values are obtained from Chemkin Pro calculation. As shown in the figure, the contribution of thermal NO exceeds 50% at equivalence ratio of 0.73 and achieves nearly 100 % approximately at equivalence ratio of 1 where flame approaches the highest temperature. This indicates that the contribution of the thermal NO to the total NO production increases with equivalence ratio at fuel lean condition due to the increase in the flame temperature. The details of NO production rate at different equivalence ratio conditions is discussed in Figure 4.23.

Figure 4.22 The effect of equivalence ratio on thermal NO contribution to total NO production

As mentioned, based on Figure 4.22, the reaction rate depends on flame temperature significantly. The effect of equivalence ratio on the rate of production of NO reaction mechanism is shown in Figure 4.23. Upper four reactions (reaction 21, 22, 27, 19) show the prompt NO_x formation that are also shown in Table 4.3. Lower three reactions (reaction 1, 3, 2) show the thermal NO_x formation that also shown in Table 4.3. As discussed in Chapter 1, prompt NO is affected by the equivalence ratio significantly since prompt NO is produced under fuel rich conditions. As equivalence ratio increases, the fuel oxidation rate slows down due to presence of less oxygen. This allows the remaining hydrocarbon radicals to react with the nitrogen to form prompt NO. At lower equivalence ratio of 0.6, the rate of production of all four reactions are not significant which indicates that prompt NO is restrained and can be ignored at fuel lean conditions. For thermal NO_x formation, as can be observed at equivalence ratio of 0.6 reaction (2) is dominant. As the equivalence ratio is increased to 0.7, both reaction (3) and (2) become competitive. At equivalence ratio of 0.8 reaction (3) is more dominant. Reaction (2) is dependent on O₂ concentration. At lean operating conditions, there is enough O₂ which aids in production rate of NO through reaction (2). As the equivalence ratio is increased, the O₂ concentrations decrease and

the contribution of NO through reaction (2) decreases. In Chemkin Pro simulation, the NO formed from reaction (3) and (2) are considered as thermal NO formation. Although prompt NO is also formed through these two reactions.

Figure 4.23 Equivalence ratio effect on ROP of NO reaction mechanism

4.3.3.2 Jet-A Fuel Simulation

Chemkin simulation is carried out and compared with experimental results for Jet-A fuels on injector 1, 2, separately. NASA code is used. Figure 4.24 shows the comparison of NO_x between experimental results and simulations for injector 1. Heat loss is considered in the simulations. As can be observed, simulations match with experimental results well for flame temperature less than 1900 K. NASA's reaction mechanism predicts ~25% higher NO_x than experimental results at higher temperature conditions. This may due to the under prediction of the heat losses at higher equivalence ratio conditions. The comparison of NO_x for injector 2 is shown in Figure 4.25. Similar to injector 1, the simulation predicts well at lower adiabatic flame temperature conditions (lower than 2000K). At 2000K adiabatic flame temperature, NO_x from simulation is over 35% higher

than the experimental data. It indicates that the increase in the NASA's rate of reaction mechanism is faster than experimental results.

Figure 4.24 Comparison of NO_x between experimental results and simulations on injector 1
T₃=560K (550°F)

Figure 4.25 Comparison of NO_x between experimental results and simulations on injector 2
T₃=644K (700°F)

The effect of the residence time on NO_x are also discussed for Jet-A fuel. The effects are shown in Figure 4.26. The results presented are for injector 2. The PSR residence time is shown in the figure. The PFR residence time is not included in this figure. At longer residence time of $t=0.8$ ms, the simulation results match with the experimental results at lower flame temperature, and overpredicts at higher flame temperature. At the shorter residence time of $t=0.5$ ms, the simulations match the experimental results more closely. It appears from these simulations that the $t=0.5$ ms is the approximately the time required for evaporation, mixing of air-fuel and combustion. The detailed NO_x formation reaction pathways are shown in Figure 4.27.

Figure 4.26 The effects of residence time on NO_x emissions on injector 2, Jet-A fuel,
 $T_3=477\text{K}$ (400°F)

Figure 4.27 shows ROP of NO_x from NASA reaction mechanism for injector 2. The effect of the equivalence ratio is also explained in the figure. As mentioned earlier, reaction (22), (27), (19) represents prompt NO_x formation. Reaction (1) (2) (3) represent thermal NO_x formation. The ROP in thermal NO_x formation at $\phi=0.65$ is more than twice at $\phi=0.55$. This indicates that the thermal NO_x formation is sensitive to flame temperature. As the fuel becomes rich, the ROP of

prompt NO_x formation also increases. This is due to remaining CH radicals reacting with nitrogen to form HCN and NCO and then into NO.

Figure 4.27 ROP of NO_x from NASA code, T₃= 644K (700°F), injector 2

4.3.3.3 GTL Fuel and Blended Fuel Simulation

CNRS is used to simulate NO_x formation for both GTL and blended fuels. The difference is the fuel composition. The input of the composition is based on Table 4.2. Figure 4.28 shows the comparison of NO_x between experimental data and simulations for GTL and blended fuels. The experimental data is obtained for injector 2. Injector 2 has higher mass flow rates than injector 1. In the figure 4.28, GTL(CNRS) represents NO_x prediction by using CNRS mechanism, simulating synthetic kerosene fuel. Blend(CNRS) represents synthetic kerosene blended with Jet-A fuel. As expected, the GTL(CNRS) predicts lower NO than Blend(CNRS) due to the low-content of the aromatic components and the high-content of paraffins. Similar to other mechanisms, the CNRS mechanism predicts well at lower adiabatic flame temperature conditions and it overpredicts at higher temperature. The detail rate of production of NO formation routes is explained in Figure 4.29.

Figure 4.28 Fuel type effects on NO emissions, injector 2, $T_3 = 644\text{K}$ (700°F)

The rate of production of NO formation for GTL and blended fuels is shown in Figure 4.29. CNRS represents fully synthetic kerosene (SK) fuel model, CNRS-B represents synthetic kerosene fuel blended with Jet-A fuel model. Blended fuel model mechanism has a faster production rate in every reaction as expected. Due to the higher concentration of propyl benzene in SK-Jet-A fuel, the prompt NO formation is facilitated. This is because the propyl benzene and propyl cyclohexane can decompose to form CH_3 , CH_2 , CH and other hydrocarbon radicals directly. More CH radicals produce more N radical in prompt NO formation reaction (21), as shown in Figure 4.29, which can also improve thermal NO formation reaction (2) and (3). However, the difference of production rate of thermal NO formation reaction (1) is not significant, and the process is slow due to the requirement of high activation energy.

Figure 4.29 Fuel type effect on ROP of NO reaction mechanism

$T_3 = 644\text{K}$ (700°F), $\phi = 0.55$

Chapter 5 Conclusions

In this work, an airblast injector is designed by using the Porous Injection Technology (PIT) to increase the contact surface area between the fuel and air. The formation of the spray is achieved by the surface stripping of liquid sheet on the porous material surface. Three different injectors are discussed in this work. Injector 1 has a fuel feed system with PIT. Injector 2 has the same fuel feeding system as injector 1 with a different counter rotating radial-radial swirler. Injector 3 has the same top swirler as injector 2 but with a conventional pressure swirl atomizer.

LDV measurements are carried out with the confinement to investigate the aerodynamic of the injectors. A CTRZ and several CRZ are observed from the contour plot. The length of the CTRZ is around 3" for all injectors. Injector 1 has a lower axial velocity than injector 3. The injector 1 has a higher Swirl Number of 0.75 and injector 3 has a swirl number of 0.6.

Spray measurements are carried out to investigate the droplet surface area density, droplets size and droplets velocities. A hollow cone spray is observed from the patternator measurements for the injectors. Droplet surface area density increases with increase in pressure drop and equivalence ratio. D_{10} , D_{32} , $D_{0.5}$ and $D_{0.9}$ are investigated from PDI measurements. For all the three injectors, the distributions of D_{10} and D_{32} are symmetric along the centerline at the downstream of the injectors. The weighted D_{10} is $\sim 20\mu\text{m}$, the weighted D_{32} is $\sim 45\mu\text{m}$. Injector 3 has the smallest weighted D_{10} and D_{32} . The comparison of spray characteristics of GTL and Jet-A fuels show that D_{32} is not significantly affected as the physical properties are the same between the fuels.

The gaseous fuel-air mixing measurements are also carried out to prove the concept for gaseous fuel applications. A doughnut shape fuel-air distribution is observed at the exit of the injector. The fuel-air distribution is uniform at 0.43" downstream of the injector with confinement.

The emission tests are carried out at various inlet air temperature and equivalence ratio conditions. For injector 1, both gaseous fuel and liquid fuel are used. NO_x emissions for injector 1 are ~ 2 ppm for natural gas and ~ 5 ppm (corrected to 15% O_2 respectively) for Jet-A combustion at adiabatic flame temperature of 1900 K. NO_x emissions increase with increase in inlet air temperature and equivalence ratio due to the increase in the rate of reactions and the higher flame temperature. Jet-A fuel, GTL fuel and blended fuel are tested on injector 2 and 3. For injector 2 and 3, NO_x emissions decrease as the inlet air temperature is increased from 400°F to 600°F. Nevertheless, NO_x emissions increase beyond this inlet air temperature. The differences in the NO_x vs inlet air temperature for injector 1 and 2 are due to the lesser fuel flow rate in injector 1, which leads to a better evaporation of fuel at 400°F inlet air temperature.

Chemkin simulations are carried out with GRI Mech 3.0 for natural gas combustion. NASA reaction mechanism is used to simulate Jet-A fuel and the prediction matches with experimental data of Jet-A fuel below 1900 K adiabatic flame temperature. The CNRS reaction mechanism is used to simulate both GTL and blended fuel. The predictions also match with experimental results for GTL and blended fuel below 1900 K flame temperature. However, both the mechanisms overpredict NO_x emissions beyond 1900 K flame temperature, since thermal NO_x dominates NO_x formation over 1900 K adiabatic flame temperature. This indicates that the reaction mechanism overpredicts thermal NO_x formation for all 3 injectors.

The results discussed in this thesis show that the injectors 1 and 2 with porous injection technology (PIT) performs on par with the injector 3, which is a conventional design with pressure swirl atomizer. The spray characteristics, gaseous fuel-air mixing data, and emissions data acquired at atmospheric conditions show that the differences observed are insignificant and fall within experimental error. Since the measurements are acquired at atmospheric conditions, it can

be concluded that the injector with PIT will perform better at higher static pressures in the combustor as PIT ability to disperse the fuel is not affected by static pressure in the combustor.

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