I, Abdelallah Ahmed, hereby submit this original work as part of the requirements for the degree of Master of Science in Aerospace Engineering.

It is entitled:
Investigation of High Pressure Combustion and Emissions Characteristics of a Lean Direct Injection Combustor Concept

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Investigation of High Pressure Combustion and Emissions Characteristics of a Lean Direct Injection Combustor Concept

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Abstract

The present work investigates emission characteristics and flame behavior of a Lean Direct Injection (LDI) combustor at elevated inlet air temperatures and pressures. The LDI consisted of a 9-point fuel injection system setup in a 3 by 3 array, where each point is made of a fuel nozzle fitted into a counter-rotating radial-radial swirler. To optimize flame anchoring and low NOx potential, two swirlers with varying intensities were used. Swirler #1 has a swirl number of 1.03 and is considered the high strength swirler. The larger recirculation zone created by this swirler is desirable for the increased turbulence and residence time, which will allow for more complete combustion and flame anchoring. Swirler #2 has a swirl number of 0.6, and is considered the low strength swirler. The higher axial velocities of this swirler allowed for a decreased residence time, which will lessen NOx production. To balance flame anchoring with lower NOx potential, 3 high strength swirlers were used in the central row of the array and 6 low strength swirlers were placed in the first and third row.

To allow for a wide range of operating conditions, three fuel stages were employed with this combustor. The three stages consist of the pilot flame, which is the central cup operating solely, the 5 cup-staged flame, which is the central circuit operating with the 4 side circuit, and the 9 cup-staged flame, which is all active injection points.

Three emission probes collected localized combustion byproducts, which were used to measure the molar fractions of nitric oxide, nitrogen dioxide, carbon monoxide, oxygen, and unburned hydrocarbons. Tests were undertaken with inlet air temperatures and pressures varying from 400°F (478-K) to 515°F (541-K) and 1-atm to 7-atm, respectively.

Test results indicate that NO$_x$ formation is highly dependent on the fuel staging. The emission index of NO$_x$ (g-NO$_x$/Kg-Fuel) were similar for just the central circuit lit (pilot) to all
circuits lit (9-cup). For example, with 400°F inlet air temperature at 4atm, the EINO\textsubscript{x} for the pilot ($\phi=0.16$), 5-cup ($\phi=0.35$), and the 9-cup ($\phi=0.60$) flames were 1.4 g/Kg, 1.35 g/Kg, and 1.56 g/Kg, respectively. However, the EINO\textsubscript{x} becomes exponentially proportional to the equivalence ratio when considering the three injection circuits independently. Correlations of the EINO\textsubscript{x} were developed for the three circuits with independent variables being equivalence ratio, inlet pressure and inlet air temperature. The experimental results indicate that this combustor generates NO\textsubscript{x} at similar rates as lean premixed combustors. However, the benefits of the LDI is its inherent ability of avoid unwarranted flashback and auto-ignition.
Acknowledgement

First and foremost, I would like to thank God for all his blessing and guidance. All that I have accomplished is due to Him and I strive to better myself for His love and mercy.

I cannot express enough gratitude to Dr. Jeng for all that he has done for me. He took me in although I had no knowledge of combustion and an inability to hold a wrench correctly. He trusted me with fixing and operating the high pressure rig, as well as conducting various projects for GE, which I had absolutely no background understanding of. To be blunt, I wouldn’t have trusted myself with many of these tasks, but his trust and determination in me pushed me to my limits both mentally and physically, leading me to accomplish and learn so much in my short time at the lab. Thank you Dr. Jeng!

I would like to thank Samir, Wessam, and Curt for being my go to guys when I cannot figure something out. You three have always had your doors and time opened for me and have guided my development at the lab.

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I had two highly memorable and happy years working with everyone at the lab. I grew up being told not to play with fire, but instead my job was to do just that and the joy of it was only made possible by the incredible patch of graduate students and co-ops I have had the pleasure of working with.
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1. Introduction

In regards to growing concerns about the environmental damage caused from the combustion of fossil fuels, aviation gas turbine engines are pressured to produce fewer pollutants as to abide to ever-more stringent civil aviation regulations. Although carbon dioxide (CO$_2$) is heavily regarded as the main contributor to the greenhouse effect, it is a natural product of hydrocarbon combustion and can only be limited by burning less fuel [1]. Due to the nature of its production, CO$_2$ regulations of jet engines were not typically enforced by the International Civil Aviation Organization (ICAO). However, as effects of climate change on weather patterns and environmental irregularities began to emanate, ICAO sought fit to propose regulations on CO$_2$ in 2016 at the 10$^{th}$ meeting of the Committee on Aviation Environmental Protection (CAEP/10). The regulation essentially forces the aviation industry to improve the specific fuel consumption (SFC) of their engines, leading to a more efficient thermal cycle [2]. Overall engine efficiency is mainly dictated by the compressor and turbine design, as the combustor tends to operate at near 100% efficiency. Alongside the projected transformation of the compressor and turbine designs due to the new CO$_2$ regulation, additional combustor technologies, such as modulated cooling, are expected to emanate as to further decrease the pollutant.

Instead, modern combustors are typically designed to limit unburned hydrocarbons (UHC), carbon monoxide (CO), and oxides of nitrogen (NO$_x$), which currently make up the ICAO gaseous emission standards. UHC are toxic and combine with NO$_x$ to create smog; UHC are a result of inefficient combustion. Carbon monoxide is a highly toxic gas, which disrupts the blood’s ability to carry oxygen. In very short residence time combustion, which exists in aviation gas turbine engines, emissions of CO tend to be minimal near stoichiometric combustion, in which increased burning rates are available to convert the gas to CO$_2$. The major concern of gas turbine combustion,
however, is NO\textsubscript{x} formation. Smog, depletion of the ozone, as well as being toxic, makes NO\textsubscript{x} of special interest and trepidation. NO\textsubscript{x} formation is exponentially proportional to temperature, which tends to arise near stoichiometric combustion [1][3].

Aviation engines are regulated based on the masses of emitted pollutants produced per unit of thrust (Dp/Foo) generated during the Landing Take-off cycle (LTO). Dp/Foo is calculated by:

\[
\frac{Dp}{Foo} = Emission\ Index \left(\frac{\text{g pollutant}}{\text{kg fuel}}\right) \times SFC \left(\frac{\text{kg fuel}}{\text{min} \times \text{KN}}\right) \times Time\ in\ Mode\ (\text{min})
\]

According to ICAO guidelines, the LTO cycle consists of 26 minutes operating at 7% of available thrust (idle), 4 minutes operating at 30% of available thrust (approach), 2.2 minutes operating at 85% of available thrust (climb), and 0.7 minutes operating at 100% available thrust (take-off)[4]. Due to the standardized LTO cycle, the \textit{time in mode} of equation 1.1 is maintained constant. So to decrease the Dp/Foo value, aviation engineers can either develop engine cycles with lower SFC or modify the combustion process to produce fewer pollutants and decrease their emission indices (EI) [1].

Table 1-1 shows regulatory limits on UHC, CO, and NO\textsubscript{x} based on the CAEP/6 for both subsonic and supersonic engines with rated output thrust greater than 26.7 KN. UHC and CO limits have remained constant since the introduction of ICAO emission standards in 1996. On the other hand, due to its harsher impact on the environment, NO\textsubscript{x} limitations have continually been made more stringent. Figure 1-1 shows the gradual decline of allowable NO\textsubscript{x} and the ICAO’s expected values for the next decade [4]. The goal of more than halving NO\textsubscript{x} regulatory standards require aviation industries to drastically modify their engine design. This report will focus on the combustor, discussing how pollutants are formed during the combustion process, methods of
mitigating them, and detailing emission results of a Lean Direct Injection (LDI) combustor concept.

ICAO Gaseous Emissions Standards

<table>
<thead>
<tr>
<th>Emission (g/kN)</th>
<th>Subsonic Turbojet/Turbofan Engines</th>
<th>Supersonic Turbojet/Turbofan Engines</th>
</tr>
</thead>
<tbody>
<tr>
<td>HC</td>
<td>19.6</td>
<td>140 (0.92)(\pi_{00})</td>
</tr>
<tr>
<td>CO</td>
<td>118.0</td>
<td>4550 ((\pi_{00})^{-1.03})</td>
</tr>
<tr>
<td>NO(_x)</td>
<td>32 + 1.6(\pi_{00})</td>
<td>36 + 2.4(\pi_{00})</td>
</tr>
<tr>
<td></td>
<td>−1.04 + 2(\pi_{00}) (2007 + engines)</td>
<td></td>
</tr>
</tbody>
</table>

Table 1-1: Aviation emission standards ([www.icao.com](http://www.icao.com)) [1]

![Figure 1-1: Regulatory standards of NO\(_x\) for various CAEP meetings [4]](image)

**1.1 Carbon Monoxide and Unburned Hydrocarbons**

CO and UHC are both indicative of combustion inefficiency. Along with limiting these pollutants during the LTO cycle to abide by civil regulation, further limitations are needed during
cruise conditions as it is customary for commercial jet engine to achieve >99% combustion efficiency [5]. UHC include fuel within the flue gases in the form of liquid or vapor, which include the fuel supplied and any lighter HC created by its breakdown within the combustor. Improper fuel vaporization and burning rates are the main reasons for UHC as well as any combustion freezing phenomena such as excessive diluting or cooling air [1]. The chemical kinetics associated with UHC varies with the type of fuel used and tends to become more and more complex with heavier fuels such kerosene, but the manners in which they arise are highly similar to CO, which will further be investigated.

In rich combustion, the lack of oxygen becomes the primary reason for significant amounts of CO. As the richness of the flame increases, the amounts of the pollutant is expected to reach its maximum value due to conditions rendering it ever more difficult for the gas to oxidize to CO$_2$. In lean combustion, high levels of CO are formed due to insufficient burning rates, low flame residence time, and low flame temperatures, which all contribute to incomplete combustion. CO tends to be relatively difficult to oxidize as compared to hydrocarbons, so it is typically considered the limiting factor in achieving complete combustion [1]. Removal of CO is mainly achieved by the following two chemical reactions:

\[ CO + OH \leftrightarrow CO_2 + H \]  
\[ CO + H_2O \leftrightarrow CO_2 + H_2 \]

Reaction 1.2 is accomplished at high temperature and is much more rapid than reaction 1.3, which requires much less activation energy. In general, CO can be reduced by increasing the operating pressure, inlet air temperature, flame temperature, and enhancing vaporization and fuel/air mixing. These parameters all increase burning rates, which will ensure more CO oxidation.
occur within a short amount of residence time. Rink et al. [6] measured CO of a flame tube at variable pressures and equivalence ratios, but maintained constant inlet air temperatures and Sauter Mean Diameter (SMD) of the diesel fuel droplets. Their results, shown in figure 1-2, indicate CO is greatly influenced by pressure. However, the minimum amount of the pollutant does not exist at stoichiometric conditions, where the flame temperature is greatest, but rather between equivalence ratios of 0.8 and 0.9. They attributed this phenomenon to fewer available oxygen at stoichiometric, as well as disassociation of CO$_2$ to CO being more and more prevalent at temperatures greater than 1850-K.

![Figure 1-2: Emission indices of CO of a flame tube operating at variable pressures and equivalence ratios [6]](image)

The higher operating pressures and temperatures of modern engine have highly decreased the amount of CO and UHC. Lefebvre [1] mentions CO decreased by a factor of two in a high-pressure combustion tests with low equivalence ratio when the pressure is increased from 9 to 14 bars. He
continues to say that the same increase in pressure eradicated CO by not allowing the dissociation of CO₂. Since the ICAO standards for CO and UHC have maintained constant since their introduction in 1996, there has not been much struggle for the aviation industry to keep up with demand in this regard. This is due to newly designed engines capabilities to sustain higher operating pressures and flame temperatures.

1.2 Oxides of Nitrogen

Modern jet engines have immensely improved combustion efficiencies, which have led to major reduction in CO and UHC. Thus, the next generation of development has mainly focused on the reduction of NOₓ [1]. It is then very important to understand the mechanism that contribute to NOₓ formation and the methods to avoid them. This section is intended as a review of previous work that will allow for better recognition of how NOₓ is produced and ways to mitigate it.

In gas turbine combustion there exists four major chemical mechanisms that contribute to NOₓ formation. These mechanisms are the Zeldovich mechanism (also known as thermal NOₓ mechanism), N₂O-intermediate mechanism, NNH mechanism, and the Fenimore mechanism (prompt NOₓ)[1,3]. It is also well documented that hydrocarbon fuel containing nitrogen will lead to further production of NOₓ as the fuel breaks down in the chemical reactions. The nitrogen in the fuel is called “fuel bound nitrogen” and the NOₓ produced from this method is called “fuel NOₓ.” Fuel NOₓ contributes insignificant amount of all NOₓ pollutants at engine operating conditions. Smith noted that at engine idle conditions, fuel NOₓ contribute to approximately 1.5% of total NOₓ, and this percentage considerably decreases at takeoff and cruise conditions [7]. Furthermore,
not much can be done to prevent fuel NO\textsubscript{x} production other than removing the fuel bound nitrogen, so further investigation of this mechanism is futile.

### 1.2.1 Zeldovich Mechanism

The Zeldovich mechanism can be used alone to accurately predict nitric oxide (NO) formation in situations where negligible NO\textsubscript{x} is formed due to the combustion process [3]. It can also be used to estimate post flame NO production, as the reactions from this mechanism is decoupled from the combustion process with equilibrium assumptions [8].

The formation of NO\textsubscript{x} due to the Zeldovich mechanism becomes appreciable at flame temperatures greater than 1800-K. This can be shown by examining the mechanism which is as follows:

\[
O + N_2 \xrightleftharpoons[k_1]{k_1} NO + N \\
N + O_2 \xrightleftharpoons[k_2]{k_2} NO + O
\]

1.4

1.5

This is the mechanism introduced by Zeldovich in 1947. Bowman introduced a third reaction in 1971, which when coupled with the previous two is considered the extended Zeldovich mechanism [9]. The third reaction, which considers radical OH is as follows:

\[
N + OH \xrightleftharpoons[k_3]{k_3} NO + H
\]

1.6

The formation rate coefficients are [3]:

\[
k_{1f} = 1.8 * 10^{11}\exp\left(-\frac{38,370}{T}\right) \quad 1.7
\]

\[
k_{1r} = 3.8 * 10^{10}\exp\left(-\frac{425}{T}\right) \quad 1.8
\]

\[
k_{2f} = 1.8 * 10^{7}\exp\left(-\frac{4,680}{T}\right) \quad 1.9
\]
\[ k_{2r} = 3.8 \times 10^{11} \exp\left(-\frac{38,370}{T}\right) \]  
\[ k_{3f} = 7.1 \times 10^{10} \exp\left(-\frac{450}{T}\right) \]  
\[ k_{3r} = 1.7 \times 10^{11} \exp\left(-\frac{24,560}{T}\right) \]  

Where \( T \) is the gas temperature in Kelvin and the reaction coefficients are in units \( \text{m}^3/\text{kmol} \cdot \text{s} \).

The reaction rate of 1.5 is much more rapid than 1.4, and due to this, reaction 1.4 is considered the rate limiting reaction [3]. That is, NO formation due to the Zeldovich mechanism is dependent only on the reaction rate of 1.4, which in turn is dependent on the available atomic oxygen within the gas mixture. Creation of atomic oxygen is formed by the disassociation of dioxygen (\( \text{O}_2 \)) as shown:

\[ \text{O}_2 \xrightarrow{k_4} 2\text{O} \]  

By neglecting the extended reaction 1.6, the Zeldovich mechanism can be simplified using reactions 1.4, 1.5, and 1.13 to formulate a global reaction:

\[ \text{O}_2 + \text{N}_2 \xrightarrow{k_g} 2\text{NO} \]  

Turns suggests that in low residence time combustion, where equilibrium NO far exceeds actual values, the reverse reaction rates can be neglected [3]. Also with minimal combustion residence time, the Zeldovich mechanism becomes uncoupled to the combustion process and \( \text{N}_2, \text{O}_2, \text{O}, \) and \( \text{H} \) can be assumed to be in equilibrium while atomic nitrogen (\( \text{N} \)) is in steady state. With these assumptions a highly simplified rate of formation for NO concentration is equated by the following formula:

\[ \frac{d[\text{NO}]}{dt} = k_g [\text{N}_2] [\text{O}_2]^{\frac{1}{2}} \]
With the units being kmol/m$^3$.s and $k_g$ can be calculated by:

$$k_g = k_{1f} \left( \frac{k_4 p_{atm}}{R_u T} \right)^{\frac{1}{2}}$$  \hspace{1cm} 1.16

Equations 1.15 and 1.16 show that NO formation is highly dependent on the forward rate of formation of reaction 1.4, which is exponentially proportional to the temperature as shown in equation 1.7. At temperatures below 1800K, the reaction coefficient is too small to allow for significant NO production. However, above this value, the Zeldovich mechanism has shown to begin dominating NO production. It is also worth noting that N$_2$ and O$_2$ concentration of equation 1.15 will increase with higher pressure, which will subsequently increase NO formation.

### 1.2.2 N$_2$O-Intermediate Mechanism

The nitrous oxide intermediate mechanism is considered one of the main contributor of NO$_x$ at fuel lean conditions ($\phi < 0.8$) [10]. The mechanism involves four reactions, which are highly dependent on radicals’ O and H created from the combustion process. The reactions are:

$$O + N_2 \leftrightarrow N_2O$$  \hspace{1cm} 1.17

$$H + N_2O \leftrightarrow NO + NH$$  \hspace{1cm} 1.18

$$O + N_2O \leftrightarrow NO + NO$$  \hspace{1cm} 1.19

$$CO + N_2O \leftrightarrow NO + NCO$$  \hspace{1cm} 1.20

Very lean combustion has become highly popular in gas turbine combustion due to the avoidance of high flame temperatures, which lead to rapid NO$_x$ generation from the Zeldovich mechanism. As the transition to lean combustion becomes dominant, the nitrous oxide intermediate mechanism becomes highly significant in NO$_x$ control strategies.
1.2.3 NNH Mechanism

The NNH mechanism also plays an important role in lean combustion. NO is also produced from this mechanism due to radicals’ O and H. The two step mechanism is as follows:

\[ N_2 + H \leftrightarrow NNH \]  
\[ O + NNH \leftrightarrow NO + NH \]

Rutar et. al showed NO produced from the NNH mechanism becomes significant at ultra-lean conditions [11]. Furthermore, fuels with high carbon-to-hydrogen ratios showed to produce more NO through this route, which indicates the importance of this mechanism with the use of Jet-fuel.

1.2.4 Fenimore Mechanism (Prompt NO\(_x\))

In 1970, Charles Paine Fenimore measured NO\(_x\) concentrations from premixed combustion of H\(_2\), CO, and various hydrocarbon fuels with air [8]. He measured emissions at variable axial locations both within the flame zone and after it. For H\(_2\) and CO flames, the NO concentrations would go to zero as the measurement location approach the injection point. This was expected, as the only known method at the time to formulate NO was through the Zeldovich mechanism. However, Fenimore realized that for hydrocarbon fuel-air mixtures, the NO concentration had a positive intercept with the axial flame distance. This positive value is seen for both fuel lean and rich conditions, but is significantly more prominent on the rich side with the major influence occurring at an equivalence ratio of 1.3-1.4. Since these nitric oxides occur far more rapidly than
can be produced through the Zeldovich route, it became known as “prompt NO\textsubscript{x}.” Similar to Fenimore’s work, Elkady measured spatial distribution of NO\textsubscript{x} at atmospheric pressure showing prompt NO\textsubscript{x} existing prior the recirculation zone as shown in figure 1-3.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure1-3.png}
\caption{Spatial distribution of NO\textsubscript{x}, $\phi$, and Temperature for atmospheric combustion of a counter-rotating swirler using Jet-A liquid fuel [12]}
\end{figure}

Fenimore attributed this phenomenon to hydrocarbon radicals, which would react with N\textsubscript{2} to form atomic nitrogen leading to the nitric oxide formation. Turns [3] summarized the reaction routes for equivalence ratios less than 1.2, but considers that at higher values, many more reaction routes become prevalent, making the mechanism highly complex. The reactions are as follows:

\[ N_2 + CH \leftrightarrow HCN + N \quad 1.23 \]
\[ N_2 + C \leftrightarrow CN + N \quad 1.24 \]
\[
\begin{align*}
HCN + O & \leftrightarrow NCO + H \quad 1.25 \\
NCO + H & \leftrightarrow NH + CO \quad 1.26 \\
NH + H & \leftrightarrow N + H_2 \quad 1.27 \\
N + OH & \leftrightarrow NO + H \quad 1.28
\end{align*}
\]

Fenimore [8] showed that in certain cases, NO concentrations reached above equilibrium levels within the flame zone. It is inferred that the Zeldovich mechanism destroys the excess concentrations through the reverse reaction of 1.4.

To summarize the four pathways, figure 1-4 displays the contribution of nitric oxide formation generated by each mechanism based on a simulated model of premixed combustion conducted by Rutar et al. [11]. His results indicate that the NNH mechanism generates the majority of NO\textsubscript{x} in very lean (\(\phi=0.61\)) atmospheric conditions. Furthermore, higher levels of prompt NO\textsubscript{x} contribution are seen with lower carbon-to-hydrogen fuels such as methane. On the other hand, heavier fuels, such as kerosene or Jet-A, have shown to have an increased level of NO formation through the NNH, N\textsubscript{2}O, and the Zeldovich mechanisms.

![Figure 1-4: NO formation pathways in premixed combustion with \(\phi=0.61\), \(P=1\)-atm, and \(T_{ad}=1790K\). Based on a simulated model by Rutar et al. [11]]
1.2.5 Super-Equilibrium of Radicals

It’s clear that atomic oxygen (O) and hydroxide (OH) are highly involved in all four NO\textsubscript{x} producing mechanisms. It was assumed that the reaction rate for NO formation due to the Zeldovich mechanism is much slower than the combustion process, and the concentration of NO due to this mechanism can be calculated by equating the radicals (O and OH) to their equilibrium values. This allows the uncoupling of thermal NO\textsubscript{x} to the combustion process. However, within the flame, these radicals are known to far exceed their equilibrium levels by up to 1000 times [3]. This super-equilibrium levels tend to be more significant in lean flames, where modern aviation combustors operate. It is then immensely important to consider these radicals as they are highly responsible for nitric oxide formation within the flame-zone through the Zeldovich mechanism as well as the other pathways mentioned. NO\textsubscript{x} formation due to super-equilibrium levels of radicals can be decreased by reducing the residence time of the combustion chemical reactions. Table 1-2 shows that thermal NO\textsubscript{x} formed within the flame zone due to super-equilibrium of radicals contribute to the majority of total nitric oxide formed in lean combustion of a well-stirred reactor using methane and air. Note that HC-N\textsubscript{2} refers to prompt NO\textsubscript{x} formation.

<table>
<thead>
<tr>
<th>Flame</th>
<th>$\Phi$</th>
<th>$P$ (atm)</th>
<th>Total NO\textsubscript{x} (ppm)</th>
<th>Fraction of Total NO Formation</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Equilibrium</td>
</tr>
<tr>
<td>Premixed, laminar, CH\textsubscript{4}–air [22]</td>
<td>1</td>
<td>0.1</td>
<td>9 at 3 ms</td>
<td>0.04</td>
</tr>
<tr>
<td></td>
<td>1</td>
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<td></td>
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<td>1</td>
<td>29 at 3 ms</td>
<td>0.53</td>
</tr>
<tr>
<td></td>
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<td>1</td>
<td>20</td>
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<tr>
<td></td>
<td>1.27</td>
<td>1</td>
<td>20</td>
<td>0.03</td>
</tr>
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<td>1.32</td>
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<td>0.02</td>
</tr>
<tr>
<td>Well-stirred reactor, CH\textsubscript{4}–air [23, 24]</td>
<td>0.7</td>
<td>1</td>
<td>12 at 3 ms</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>0.8</td>
<td>1</td>
<td>20</td>
<td>—</td>
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<tr>
<td></td>
<td>1.0</td>
<td>1</td>
<td>70</td>
<td>—</td>
</tr>
<tr>
<td></td>
<td>1.2</td>
<td>1</td>
<td>110</td>
<td>—</td>
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<tr>
<td></td>
<td>1.4</td>
<td>1</td>
<td>55</td>
<td>—</td>
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</table>

*Table 1-2: Formation pathways of NO of premixed and well-stirred reactors utilizing methane and air [2]*
1.3 Modern Gas Turbine Combustor Technologies

Gas turbine combustion has made deliberate advancement throughout its existence. The major advancements were due to achieving military advantages in the air during the world wars. This was the point at which gas turbine engines were at its infancy stage and research of the mechanics involved in the combustor led to rapid understanding of the how to improve it. Initially there were various types of combustor, which varied from can, can-annular, and annular arrangements. Within each of these arrangements, the fuel was allowed to cross, counter, and parallel flow with the air to allow for stable combustion. The combustors initially varied drastically from one manufacturer to another. This later changed as a common ground was found to allow for a compact, efficient, environmentally friendly design of the rich burn, quick quench, lean burn combustor (RQL).

![Diagram of NOx formation relative to equivalence ratio and the path for an RQL combustor](image)

*Figure 1-5: Rate of NOx relative to the equivalence ratio and the path for an RQL combustor*
The RQL, as it implies, has three major zones in which the first zone is burning fuel rich in a centrally recirculating flow region that is being created by an axial or radial swirler; this is meant to anchor the flame and ensure a blowout does not occur. The second zone is meant to quickly drop the fuel-air ratio to achieve lean combustion through the addition of air from an outer and inner annulus. From the nitric oxide producing pathways, the most concerning one is the Zeldovich mechanism. This is because of the exponential proportionality to temperature, which can lead to tremendous amount of NOx if not handled properly. So to avoid major NOx formation, combustion needs to produce a low flame temperature, which can be achieved by either burning highly fuel rich or lean. The benefit of rich combustion is high flame stability, causing blowouts to be improbable. However, burning fuel rich leads to inefficient combustion and high levels of soot and UHC production. So solely burning rich is not a viable option. By burning rich and quickly diluting to a lean burn, a highly stable flame is developed as well as avoidance of high formations of nitric oxides. The final zone is called the dilution zone; in this region, the goal is to reduce the temperature of the air as it enters the turbine by adding more air from the outer and inner annulus. The RQL is the most popular gas turbine combustor design to date, as it is used in over 80% of commercial engines [1]. Image of an RQL flame taken by Abd El-Nabi [13] is shown in figure 1-6 with the operating zones outlined. A counter rotating swirler was used with Propane to generate this flame.
The major problem with the RQL is the transition from fuel rich to lean, where stoichiometric combustion could be reached causing maximum flame temperature and high NO\textsubscript{x} formation. Although the quenching process is designed to be quick, it is evident that the RQL emits much more NO\textsubscript{x} than fully lean burning combustors due to the local hot spots caused by this transition from rich to lean combustion\cite{5}.

The ICAO’s goal to halve the NO\textsubscript{x} emission standard by 2026 may be beyond the capabilities of traditional RQL combustor \cite{4}. The stringent regulations have forced rapid development of stable, fully lean combustors. The two prominent fully lean combustor types are the Lean, Premixed, Prevaporized (LPP) and the Lean Direct Injection (LDI). The LPP operates by vaporizing the fuel and fully mixing it with the air within the air mixer before allowing it to reach the dome for combustion. By premixing the air and fuel, LPPs can stably operate at ultra-
lean conditions and have shown to “reduce NOx emissions 90% relative to (1990s) engine technology [14].”

![Diagram showing combustion concepts](image)

**Figure 1-7: The three prominent combustor concept’s methodology of avoiding NOx**

However, by premixing the air and fuel at high inlet temperatures and pressures, LPP combustors are susceptible to auto-ignition and flashback within the mixer, which can lead to mechanical damage and engine failure. To remedy this, the LDI concept is designed to inject fuel directly into the flame front. However, rapid mixing of the fuel and air is required in LDI combustors to ensure the flame is stabilized and that the fuel does not escape the central recirculation zone unburned. To achieve this, it is required that finer fuel droplets exit the nozzle and an increased shear layer, which aids in further fuel atomization, is generated in radial-radial swirlers. To increase the size of the shear layer relative to fuel injection, the swirler size is significantly decreased, as compared to RQL, to allow multiple swirlers to operate in the conventional area of a single cup. With each miniature swirler generating its own shear layer, the fuel/air mixing is enhanced to allow entrainment of the flame in the central recirculation zone when operating at its intended lean conditions [6].
Extensive work on LDI at elevated temperatures and pressures done by Tacina et al. [14–18] have shown that by increasing the number of swirlers in a cup (i.e. decreasing the swirler size), the NO\textsubscript{x} emission index subsequently decreases. Tacina explains that as smaller swirlers create shorter recirculation zones, the residence time of the chemical reaction is decreased; the chemical kinetics required to form NO\textsubscript{x} is a slow process, so the decrease in residence time greatly impacts NO\textsubscript{x} production. At the same time, it can be noted that since direct fuel injection can have unmixed fuel zones, the residence times can be expected to be greater than that for LPP, which have very short residence times due to premixing of fuel and air. However, the smaller swirlers have shown to be unstable at ultra-lean conditions, indicative of engine idle levels. Furthermore, practical complications arise in the implementation of an increased number of swirlers and fuel circuits. Tacina suggested that fuel staging and fewer swirlers be used to ensure optimal operations at low power conditions.

Figure 1-8 shows EINO\textsubscript{x} for various LDIs tested by Tacina [16], as well as typically ranges achieved by LPPs operating at 2760 KPa. Note that the lean premixed case is indicative of combustors using gaseous fuels. The data indicates that LDIs utilizing swirlers with weaker swirl intensities produce fewer NO\textsubscript{x}. This is attributed to the higher axial velocities created by the decrease in swirl, which would reduce the flame residence time and recirculation zones where rapid NO\textsubscript{x} exists. Furthermore, the data indicates that LDIs typically produce more NO\textsubscript{x} than LPPs, but can achieve similar levels through varying the design and injection strategy.
2. Review of the Lean Direct Injection Combustor Concept

The combustor presented in this paper has been deliberately designed and tested under atmospheric conditions by Endicott and Haseman with the overall goal to develop an aircraft based combustion, which is fully lean, stable, and resistant to flashback through direct injection of the fuel into the combustion chamber. The rich burn, quick quench, lean burn concept has shown that the temperature gradient created by the transition from fuel rich to lean combustion is prone to produce sizable amount of NO$_x$ pollutants. That is, any minute residence time at near stoichiometric conditions will cause a significant increase of NO$_x$ and development in speeding up
the quenching process can only mitigate this phenomenon to a certain degree before being futile. As the International Civil Aviation Organization introduce ever more restrictive regulations of NO\textsubscript{x} in commercial jet-engines, the aviation industry has been struggling to keep up with demands through the use of the RQL concept. Instead, the industry has been adjusting to these regulations through the development of fully lean combustors, which would lower NO\textsubscript{x} by negating any flame hot spots caused by the transition from fuel rich to lean.

Although lean combustion is favorable from an emissions’ standpoint, it is known to be less stable than burning fuel-rich. The lean premixed prevaporized concept have shown to remedy this problem by premixing the air and fuel within the swirler prior to reaching the flame zone. Premixing allows for uniform distribution of the air and fuel as it reaches the combustor dome, which will not only create a highly stabilized lean flame, but will also mitigate the temperature gradient and lead to lower NO\textsubscript{x} pollutants. GE ventured with NASA in 1995 to develop a suitable LPP for use in commercial jet-engine [5]. They produced the Twin Annular Premixed Swirler (TAPS), which was first marketed in the GEnx in 2010 and due to its successful introduction is now being implemented in the CFM56 as part of LEAP. However, TAPS has challenges with auto-ignition margins and combustion dynamics. For example, Stickles et. al [5] mentioned that although TAPS was designed to avoid unwarranted auto-ignition, which LPPs tend to be susceptible to because of the premixing, there exists a margin where it occurs and this margin is expected to grow with higher pressures. Also, he noted limited operating range due to combustion dynamics, which is favorable in lean combustion.

The entirety of this project was meant to develop a highly stable Lean Direct Injection combustor concept, which can operate at a wide range of fuel-to-air ratios. The fully lean combustor will produce low NO\textsubscript{x}, as well as avoid any possibilities of auto-ignition through direct
injection of the fuel into the flame front, allowing it to be suitable for high pressure ratio engines. However, lean combustion coupled with improper fuel/air mixing due to direct fuel injection is favorable to undesirable blowouts and high levels of CO and UHC. So the main goal was to ensure flame stability and entrainment of the fuel, which was accomplished through atmospheric testing done by Jacob Haseman [19]. This thesis will review the development of the LDI concept designed at the University of Cincinnati and to present emission characteristics derived from combustion conducted at elevated inlet air temperatures and pressure.

2.1 Design Consideration

To stabilize the flame, LDIs are designed with miniature swirlers, each fitted with its own fuel nozzle. This allows for a more uniform distribution of the fuel as well as a highly turbulent reaction zone, which will result in the rapid vaporization and fuel-air mixing needed for flame anchoring and entrainment of the fuel within the recirculation zones. However, this causes multipoint LDI to have highly complicated fuel circuits and swirler arrays. NASA[14,16,20] have developed several LDIs with two of their concepts are displayed in figure 2-1. They have noted that for most cases, the only feasible method of manufacturing their concepts are through micro-laminating the swirlers, fuel nozzles/circuit, and the combustor dome plate as one integrated unit. This is problematic as it makes it impossible to replace components such as damaged fuel nozzles or study the effect of various configurations like the use of different swirler types or nozzle insertion depth without having to manufacture an entirely new combustor. The LDI concept developed at the University of Cincinnati was designed to avoid all these complications as well as being optimal for integration with aviation gas turbine engines. All components of the combustor are assembled independently and can be replaced or modified as needed. Among other things, this
allowed for the understanding of the effects on flame stabilization due to different types of swirlers and nozzle insertion depths.

![Figure 2- 1: A 49 point multiplex (left) and a 36 point sector (right) LDI concepts developed by NASA [15,16].](image)

2.2 Design/Approach

As depicted in figure 2-2, this LDI concept consisted of a 9-point fuel injection system setup in a 3 by 3 array, where each point is made of a fuel nozzle fitted into a counter-rotating radial-radial swirler. It is inferred that each 3 by 3 array of swirlers makeup a conventional cup of the combustor, so the size of each swirler and the entire combustor assembly was designed small enough to maintain practical and convenient weight and volume for a jet engine. Thus, the exit flare diameter of all swirlers used was 1 inch and the center to center spacing is maintained to 1.5 inches. This allowed the array to be fitted into a square combustion chamber with a 4.5 x 4.5 in. inner dimensions.
Tangirala et al. [22] conducted experiments to determine the effects of high swirl on flame stabilization. They suggested that better mixing occurs with the increase in swirl up to an approximate swirl number (SN) of 1. At even higher swirl number, the turbulence begins to decrease, which leads to poorer air/fuel mixing and flame stability. Their study infers that SNs ranging from 0.6 and 1.0 are optimal in gas turbine combustion. Two counter-rotating, radial-radial swirlers with varying swirl intensity were used for this project. Swirler-1 is a high strength swirler with a swirl number of 1.03 and was used to anchor the flame. Swirler-2 is a low strength swirler with SN of 0.60 and was used to reduce the flame residence time (reduce NOx). Cross-sectional models of the swirlers are shown in figure 2-3. Three swirler configuration were studied to correlate their effects on the aerodynamic flow field and on flame anchoring/stability. As illustrated in figure 2-4, the baseline configuration consisted entirely of the low-SN swirlers. Configuration 2 use eight low strength swirlers, with one high strength swirler in the center of the array. Configuration 3 uses three high strength swirlers in the center row and six low strength swirler is the first and third row of the array. Both swirlers have similar effective flow area of
approximately 0.13 in$^2$.

Figure 2-3: Solid models of the high strength swirler (left) and low strength swirler (right) [19]

To allow for a wide range of operating conditions, the fuel line assembly allowed for fuel staging of three independent circuits. Circuit one, two, and three supplied fuel to the central nozzle, the four side nozzles, and the four corner nozzles, respectively. All simplex nozzles used had a 72° spray angle. Two nozzle insertion depths were also studied as depicted in figure 2-6. Full direct
injection of the fuel was conducted through the “shallow” insertion depth and partial prefilming on the tip of the venturi was done through the “deep” insertion depth.

Figure 2-5: Three independent fuel circuits consisting to the central nozzle (1), the four side nozzles (2), and the four corner nozzles (3)

Figure 2-6: Injection pathways due to the two nozzle insertion depths [19]

2.3 Aerodynamic Behavior of Swirling Flow

The aerodynamics results of the swirlers were studied individually prior to conducting the experiment of the LDI array. Cold flow Particle Imaging Velocimetry (PIV) of the high strength swirler was conducted by Kao [23] and the low strength swirler was done by Brennan [24]. Both
cases used a confinement ratio of 5.1 under a 4% pressure drop at atmospheric conditions. The results are as follows:

![Figure 2-7: PIV results of the high strength swirler [23] (left) and the low strength swirler [24] (right)](image)

Under similar conditions, the aerodynamics of the LDI configurations were conducted by Endicott [21]. Baseline configuration utilizing nine low strength swirlers resulted in uneven recirculation zones (RZ) between the central, the four side, and the four corner swirler(s), which were all ultimately exhibiting weaker swirl intensity than the single swirler tested by Brennan. The single low strength swirler shown in figure 2-7 had approximately a 3 in. long RZ, a reverse flow magnitude of 15 m/s, an axial velocity exiting the swirler of 40 m/s, and an expansion angle of 46°. When nine of these swirlers are placed in an LDI array, the RZ lengths of the central, side,
and corner swirlers are shrunken to approximately 1 in., 1.5 in. and 2 in., respectively. The reverse flows, axial flow exiting the swirlers, and the expansion angles also decreased to 12 m/s, 30 m/s, and 27°, respectively. Essentially the entirety of the nine swirling flows is being destroyed, but not to the extent of diminishing any of their respective recirculation zones. Since the swirlers are all rotating in the same direction, each one will oppose the flow of neighboring swirlers resulting in the reduction of the tangential momentum of one another as they interact. The swirl intensity is determined by the ratio of axial flux of the tangential momentum to the axial flux of the axial momentum [25]. That is:

\[ SN = \frac{G_\theta}{G_z R} \quad 2.1 \]

Where R is typically considered as the exit radius of the swirler and the axial flux of the tangential and axial momentum can be calculated by integrating the velocity components of the flow field as follows:

\[ G_\theta = \int_0^\infty \rho u_z u_\theta r^2 dr \quad 2.2 \]

\[ G_z = \int_0^\infty (\rho u_z^2 + p)r^2 dr \quad 2.3 \]

So the swirl destruction in the LDI is caused by the diminishing of the tangential momentum due to neighboring swirler interactions. Since the central interacts with four swirlers, it experienced the most destruction, followed by the side swirlers and then the corner ones. PIV results of the axial velocities for swirler configuration 1 in shown in figure 2-8.
The intended use of the central swirler was to anchor the flame. So having a minuscule central RZ was not acceptable as there needed to be sufficient residence time within that region to stabilize the flame during ultra-lean combustion. So to remedy the problem of configuration one, a high strength swirl was placed in the center of the array. Doing so had a significant difference to the central RZ. The expansion angle grew to 57° and the length of the RZ became approximately 2.5 in. long. This central RZ is considered sufficient to establish a well-anchored flame. Furthermore, the remaining swirlers maintained a small RZ length of approximately 1 in., which
would be useful in mitigating NO\textsubscript{x} by decreasing their flames’ respective residence time. Results of swirler configuration 2 is shown in figure 2-9.

Figure 2-9: PIV results for swirler configuration 2 (1 high strength swirlers in the center) [21]

Although the attributes of configuration 2 were considered acceptable as to provide flame stability as well as low NO\textsubscript{x}, it is less practical for adaptation in an annular combustor. That is, the central row would require alterations between high and low strength swirlers, which will create asymmetry of the flame and likely lead to complications such as an increased temperature gradient due to that lack of uniformity in air distribution. This will in turn lead to increased NO\textsubscript{x}, CO, and
UHC. So the third configuration was tested using high strength swirlers in the entire middle row. The results of this configuration, shown in figure 2-10, indicate a much more uniform flow pattern across all nine RZ with approximate lengths of 1.5 in. The central RZ expanded more radially and remains slightly longer than the other RZs, which indicate its ability to continually be a good source of a well anchored flame. Furthermore, healthy and uniform shear layers were created across the array. These shear layers are fundamental in fuel vaporization and fuel-air mixing. From the aerodynamics results, it was expected that this swirler configuration is optimal, but nevertheless, atmospheric combustion was conducted on all three configurations for better assessment on flame stabilization of the this LDI concept.

![Figure 2-10: PIV results for swirler configuration 3 (3 high strength swirlers) [21]](image)
2.4 Atmospheric Combustion Results

Atmospheric combustion of the LDI were conducted by Haseman [19]. All atmospheric combustion tests used a 4% pressure drop with a preheated air temperature of 400°F. The atmospheric combustion results were used to determine the equivalence ratios at ignition and lean blowout (LBO) of only the central circuit lit (pilot) and the steady state combustion of various fuel circuit configurations. Table 2-1 shows the ignition and LBO results for the three swirler configuration and the two nozzle insertion depth. The data indicates that a locally rich pilot flame is needed for ignition and stable operating condition. As the other circuits become active, the pilot flame is maintained slightly richer to enhance flame anchoring and stability. Initial testing was of the shallow nozzle insertion depth (full direct injection) of all swirler configurations. This section will discuss the shortcomings of the shallow insertions and how they were avoided through the deep insertion, as well as the optimal swirler configuration for flame stability and anchoring.

<table>
<thead>
<tr>
<th>Ignition/LBO Equivalence Ratios - Global $\phi$ (Local $\phi$)</th>
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<tr>
<td><strong>Configuration 1</strong></td>
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<td><strong>Deep Insertion</strong></td>
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<td></td>
</tr>
<tr>
<td><strong>Shallow Insertion</strong></td>
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*Table 2-1: Equivalence ratio of LBO and ignition for the pilot flame [19]*

Also shown in table 2-1, is that a much richer flame is needed for a sustained pilot flame arising from the shallow insertion as opposed to the deep insertion. For example, the pilot flame’s
local equivalence ratio needed to be 3.16 for the shallow insertion depth to maintain a stable flame, whereas a 1.56 local equivalence ratio was needed for the deep insertion depth of swirler configuration 2. Images of the flame developed through the shallow insertion shown in figure 2-11 indicate a highly unstable combustion for all swirler configurations and active fuel circuits. Significant amount of diffusion yellow flame indicates lack of entrainment of the fuel within the recirculation zones. This is a result of improper fuel-air mixing and fuel vaporization. Although combustion can be sustained in this configuration, it will lead to a highly inefficient flame, which will produce a significant amount of carbon monoxide (CO) and unburned hydrocarbons (UHC). Furthermore, a considerable amount of NOx is expected to emanate due to the stoichiometric burning conditions needed to sustain the flame as well as the substantial temperature gradient created by the uneven distribution of the fuel.

The flame was shown to be more stable with the use of the higher strength swirler configurations. To sustain a pilot flame, the equivalence ratio of swirler configuration 1 (all low SN) needed to be much higher than with the other two configurations. Furthermore, it appears that diffusion flames are more prominent in configuration 1, which is likely due to the higher axial velocity and shrunken recirculation zone of the central swirler due to its low swirl strength, which would mitigate the flame zone’s residence time and force the fuel to more easily escape it. Shown from the aerodynamic data in figure 2-8, the tangential momentum and the overall recirculation zone of the central swirler are experiencing severe destruction due to its interactions with the surrounding swirlers and this is overcome by replacing it with a high strength swirler. The effects of this phenomenon emanates in the combustion behavior as the central flame of configuration 1 is overwhelmingly weak when operating with 5 or 9 active cups. Furthermore, swirler configuration 3 had the lowest LBO limit and was capable of operating at a wider range of fuel-
to-air ratios, indicating that the more high-strength swirlers used, the better the flame anchoring becomes.

Figure 2-11: Flame images of the shallow nozzle insertion depths and the variable swirler configurations [19]
To solve the lack of fuel entrainment and diffusion flames created from the shallow nozzle insertion depth, the aerodynamics were revisited. Firstly, note that the counter-rotating swirlers used in this project were purposefully selected for their shear layers; the shear layers were intended to be used to optimize fuel vaporization and mixing. Also, the swirlers were compactly placed to generate a secondary counter-rotating shear layer due to swirler-to-swirler interactions. That is, by having all the swirlers rotating in the same direction, each swirler will inherently oppose the flow of any adjacent swirler as they interact, thus creating the secondary shear layer. What appears to have happened with the shallow insertion depth is that the fuel injected completely bypassed the primary shear layer and partially entered the secondary shear layer, which was highly insufficient in breaking down the fuel. As a solution, the nozzles were retracted to allow for fuel injection to follow the path of the air into both the primary and secondary shear layers. To allow for optimal injection, the fuel needed to partially prefilm on the tip of the venturi, which ultimately led to even better flame stability and anchoring. The schematic in figure 2-12 shows the shear layers as well as the differences between the two insertion depths.

*Figure 2-12: Shallow insertion (left) allowed the fuel to bypass the shear layers, whereas the deep insertion (right) allow for perfection injection into the shear layers [19]*
As shown in figure 2-13, the difference in flame structure and characteristics were remarkable between the two insertion depths. For all swirler configurations, the deep insertion produced a uniform, short blue flame. The equivalence ratio of LBO of the pilot flame was significantly reduced, and it appears that proper fuel entrainment within the recirculation zones was achieved for all conditions. A discrete square flame structure is developed within each cup, showing how the radial jet momentum from adjacent swirlers are self-destructive, leading to the inherently round swirl structure to flatten out. Overall, the deep insertion is superior in all conceivable fashion than the shallow insertion, and all proceeding tests utilized it.

Figure 2-13: Flame images of the deep nozzle insertion depths and the variable swirler configurations [19]
Interestingly, a sizable amount of the flame (hot gases) tended to recirculate into adjacent swirlers only if those swirlers are inactive. When considering just one active cup, the flame appears to be very weak or nonexistent in the center of swirler despite the locally rich condition. When the side circuits are lit, the flame covers the entire central swirler, although the flame is transitioned to lean condition. Similarly, the center of the side swirlers are void until the corner circuits are lit. This phenomenon forced the use of richer local swirlers for every fewer active circuits. It is believed that this is caused by the expansion of combustion products due to high flame temperature, which would force the gases to prefer the lower density/pressure of the cold adjacent flow fields.

Regardless, the deep insertion depth results reiterate that the use of higher-strength swirlers is optimal for flame anchoring. The high SN swirlers are shown to produce a more distinguished and brighter flame, which can burn leaner than the low SN swirlers. This is expected as the residence time within the recirculation zone is increased due to the decrease in axial momentum. This is projected to result in higher NO\textsubscript{x} production. However, flame stability and an increased range of operating conditions was considered more desirable than the expected higher NO\textsubscript{x}, so swirler configuration 3 was determined to be the most appropriate for high pressure combustion and emission analysis.

### 2.5 Goals and Expectations of High Pressure Combustion

The goal of the high pressure tests was to understand the combustor’s behavior at variable pressures and inlet air temperatures. The LDI exhibited combustion dynamics (thermoacoustic instabilities) in atmospheric combustion, which was further investigated to understand how the
operating pressure affects its onset. Three emission probes were deliberately placed in locations to allow for reliable understanding of the profile distribution of the emitted pollutants. Gases measured were NOx (both NO and NO2), CO, O2, and UHC. The combustor was tested at pressures ranging from 2±0.1 to 7±0.27 atm. (29.4±1.5 - 102.9±4 psia) and with inlet air temperatures of 400±10˚F and 500±10˚F. The pressure drop was maintained at 4±0.3%. These tests were intended to demonstrate the low NOx potential of the combustor as well as predict its characteristics at engine operating conditions. Emissions results indicate high dependency on local equivalence ratios, which is characterized by the fuel staging strategy. The work presented in this thesis will explore this phenomenon by explaining the three distinct emissions patterns arising from the three staged fuel circuits.

3. Experimental Facility

The high pressure (HP) rig was intricately designed to control and acquire relevant data via LabVIEW (LV). The HP rig consists of two distinct flow passages, a hot flow passage consisting of the plenum, and the test section, and an annular cooling flow passage surrounding it. Two air flows, total and main, were controlled through pneumatic valves via the LV code. The total air flow indicates the air flowing through the cooling air and plenum, while the main air flow indicates the flow only through the plenum. Two orifice plate flowmeter were used to determine their respective flow rates. A 192-KW air heater, which is installed in line with the main air flow, was used to preheat the air to required levels prior to reaching the combustion chamber. The air compressor can supply a maximum flowrate of 0.91 Kg/s (2 lb/s) and maximum pressure of 11.9-atm (175-psia). Due to the large flowrate needed for the LDI, the flowrate was exacerbated at a
maximum pressure of 7-atm (102-psia), which would be the maximum pressure tested.

The fuel flowrates were manually controlled. However, the flowrates were continually transmitted to LV, allowing monitoring of instantaneous equivalence ratios. A custom designed hydrogen ignition system was implemented to develop a flame kernel. A transformer supplying 15,000-V was used to generate an electrical spark in conjunction with hydrogen flow supplied via a pressurized tank to generate the ignition flame. City water supplied at 4-atm (60 psi) was passed through a pump to supply a maximum pressure of 15-atm (220 psi) of which it was used to rapidly cool the exhaust gases in the water jacket and the emission probes. The pressure of the rig was controlled via a back pressure regulator, which was controlled in LV. Temperatures and pressures were continually monitored at various locations and used to trigger an emergency shutoff if necessary. Image of the combustor’s integration into the rig and the overall experimental schematic and are shown in figures 3-1 and 3-2, respectively.

![Figure 3-1: Integration of the LDI into the high pressure rig](image-url)
Figure 3-2: Experimental schematic of the high pressure testing

3.1 Hydrogen Ignition Flame

Developing a flame kernel was difficult to accomplish with the combustor in the high pressure chamber. In atmospheric combustion, a propane torch was used to ignite the pilot flame (central circuit) and stabilizing it prior to activating the other circuits. An electric spark is typically solely used to ignite the fuel and air mixture in the high pressure rig, as using a propane torch is obviously not feasible. However, igniting the pilot flame with a spark required the electrodes to be inserted well into the combustor to reach the central circuit. This would require the electrodes to pass by the side and corner circuit, which lack fuel injection. Doing so would result in deterrence of the flame as well as possible damage and melting of the electrodes. An option of igniting the
side circuit, rather than the central circuit was avoided as this would require a vast amount of fuel to escape through the exhaust as well as enter the emission probes, which would later lead to imprecision in measurements.

Due to this problem, an ignition flame was needed. The amount of energy required for ignition of a fuel depends on the temperature, pressure, thermal diffusivity, and laminar flame speed. That is:

\[ E_{\text{ign}} = 61.6 \, P \, \frac{C_p}{R_b} \left( \frac{T_b - T_u}{T_b} \right) \left( \frac{\alpha}{S_l} \right)^2 \]  

At standard atmospheric stoichiometric combustion, the flame speed for hydrogen is 230 cm/s, whereas for methane and propane the flame speeds are 35 cm/s and 38 cm/s, respectively [26]. That is, hydrogen-air mixtures typically require less than 200 times the amount of the energy needed to ignite propane-air mixtures. Alongside being much easier to ignite, the high flame velocities allows the hydrogen-air mixtures to sustain combustion in higher axial velocity flow fields. So due to its desirable characteristics, hydrogen was used to generate an ignition.

An electrical transformer taking 120 Volts-AC and outputting 15000 Volts-DC was used to develop a spark and ignite the hydrogen-air mixture within the combustor. Ultra-high temperature tungsten wire was used as an electrode for its ability to withstand temperatures up to 5000°F. The tungsten was fed through a hollow non porous high alumina ceramic rod, which acted as a high temperature electrical insulator. The ceramic rod was teed into the combustor alongside the hydrogen inlet. A small concentric gap between the rod and a stainless steel tube allowed for both an air resistant to generate the spark and for the hydrogen inlet into the combustor, where it was meant to ignite. A schematic of the ignition system is shown in figure 3-3.
Figure 3-3: Hydrogen ignition system

3.2 Emissions Sampling Probe

Grove et al. [27] showed that NO\(_x\) is conserved in 316 stainless steel and quartz tubes at temperatures of 300°C and 400°C, respectively. At higher temperatures, reduction of NO\(_x\) is significantly increased (20-85%) if oxygen and high levels of HC are present. Cooling when operating at temperatures of 200°C or higher is required for more accurate reading.

Yanagi [28] suggested that a large probe width will produce significant error when used in a flow with a concentration gradient. This error tends to diminish with high sampling rates and decreasing probe width. The probe should be less than 10% of the characteristic diameter of the combustion chamber. Furthermore, in heterogeneous flows where liquids, solids, or gaseous
gradients of varying densities exist, sampling must be done at the stream velocity for correct readings.

To maximize accuracy based on Grove and Yanagi’s suggestions, an in-house emission probe was designed and integrated into the combustion chamber, ensuring that no leakage in or out the chamber occurs and the chemical reaction is quickly quenched within the probe. The probe designed consists of three concentric 316 stainless steel tubes of outer diameters of 0.125 in., 0.25 in, and 0.375 in. Emitted gases would flow through the inner 0.125 in. tube and into the analyzer via a heated hose, which would prevent water condensation. Cooling water, which would flow through the 0.25 in tube and out of the 0.375 in tube, rapidly quenches the chemical kinetics of the gases as they pass through the inner tube. A schematic of the design and operability of the probe is shown in figure 3-4.

![Figure 3-4: Design and operability of the emission probe](image)
Based on the aerodynamic results, the recirculation zones of configuration 3 diminishes at approximately 1.5-inch downstream of the swirler array and beyond that the flow becomes relatively uniform at approximately 3-inch downstream [21]. However, the recirculation zones of reacting flows are typically larger and exhibit more strength than non-reacting flows. Reacting flows exhibit higher velocities than non-reacting flows due to the increase of temperature causing the air to expand [22]. According to Yanagi, for reliable global emission measurements, samples should be collected at locations where the flow is uniform [28]. Samples should be collected well after the recirculation zone to ensure uniformity. Based on this, the probes were placed 6-inch downstream of the swirlers.

![Image of sampling locations of the three independent probes.](image)

*Figure 3-5: Sampling locations of the three independent probes.*
To further acquire reliable emission measurements, three emission probes were used to collect localized combustion byproducts of the three fuel injection locations. It was expected that by having a uniform flame with all nine active circuits, the pollutants would also be uniformly distributed at the exit of the combustor. However, the cold air flowing through inactive cups from the other two fuel staging strategies is expected to produce a highly irregular flue gas distribution. By having three independent probes measuring emissions of a side, corner, and the central regions of the combustion chamber, a better understanding of this lack of uniformity due to fuel staging can be achieved. The three sampling locations are illustrated in figure 3-5. Furthermore, a reliable global emission measurement can be calculated by the weighted average of three measurement points. That is:

\[ X_{i-global} = \frac{X_{i-A} + 4X_{i-B} + 4X_{i-C}}{9} \]  

3.2

3.3 The LDI Combustor

The design of combustor was described in section 2.3 and this section will discuss in more details its relations to high pressure combustion. Based on atmospheric combustion, the swirler configurations consisting of three high strength swirlers (SN=1.03) in the central row and six low strength swirlers (SN=0.60) in the first and third row was considered to be optimal [19]. This configuration provided a wider range of flame stability, which is required of LDI combustion for aviation engines. However, the use of three high strength swirlers is expected to increase the residence time of the flame, which in turn will increase NO\textsubscript{x} pollutant. Regardless, this configuration was chosen for high pressure combustion as it was considered more important to enhance combustion efficiency.
Haseman [19] individually measure the effective area for all of the swirlers that were used (equation 3.3).

$$A_{eff} = \frac{\dot{m}}{\sqrt{2\rho \Delta P}}$$

The average effective flow area of the high strength swirlers is 0.13 in.$^2$. This would correspond to a flowrate of 51 lb/hr at atmospheric pressure with an inlet temperature of 400°F and a 4% pressure drop. The low strength swirlers used had an average effective flow area of 0.14 in.$^2$. The flow rate at 400°F and 4% pressure drop to the atmosphere is 55 lb/hr. Summing the overall effective area of the nine swirlers results to 1.2 in.$^2$, which correspond to a flowrate of 485 lb/hr. However, the flowrate measured was consistently 445 lb/hr. The 8% discrepancy is indicative that the swirlers are incapable of attaining their maximum flowrates due to them being placed in a highly compact orientation. The inlet diameter of the high and low strength swirler are 1.25 in. and 1.42 in., respectively. The closest spacing between two low strength swirlers, two high strength swirlers, and a high strength with a low strength swirlers are 0.08 in., 0.25 in., and 0.165 in., respectively. The minute spacing between any two low SN swirlers is believed to cause the reduction of effective flow area. This was more readily seen with swirler configuration 1 (9 low SN swirlers), where a 20% discrepancy in flowrate exists between measured and expected values.

Quartz glass was placed on one of the sides of the combustor. Haseman [19] designed the combustor as to allow for that side to align with a window in the high pressure rig. This allowed the flame to be visible during testing and so images/videos of the flame were taken for every operating condition. The operating pressure and equivalence ratios had significant alterations to the flame behavior, and these images/videos are used to shed light on the matter.
Figure 3-6: Cross-sectional model of the combustor tightly spaced swirler array, central probe placement, and the alignment of the combustor to one of the rig’s windows

3.4 Emission Measurement

The California Analytical Instruments Model 600 CLD emission analyzer was used for dry determination of oxides of nitrogen through the chemiluminescent method. The analyzer operates in two modes allowing measurements all oxides of nitrogen (NO$_x$ mode) within the sampled gas or only considering the amount of nitric oxide (NO mode). In NO mode, the NO in the sample is quantitatively converted to NO$_2$ by gas phase oxidation with molecular ozone produced by ultraviolet reaction of compressed dry air. A small percentage of these NO$_2$ molecules are elevated to an electronically-excited state followed by a decline to a non-excited state alongside emission of photons. A photodiode collects these photons to generate a low DC current directly proportional to the NO available within the sample. The current is amplified by a precision electrometer which is used to determine the levels of NO. In NO$_x$ mode, the NO$_2$ within sample gas is reduced to NO,
then the NO mode procedure is followed again to determine to total amount of NOx.

The California Analytical Instruments Model 600 NDIR/Oxygen multicomponent emission analyzer is used for the analysis of CO and O₂ gases. The method of operation to measure CO is based on the infrared absorption characteristics of the gas. The analyzer delivers a stable measurement through the use of the infrared beam to measure the gas concentration. The beam, which is controlled by a chopper system, travels through a cell of predetermined length containing the sample gas. The data is analyzed as the beam passes through the cells absorbing some of its energy. The transmittance develops from the cell and travels through the front chamber of the infrared microflow detector which is filled with the sample gas. The beam experiences the energy absorption which increases the pressure in chamber. The difference in pressure from the front and the back chamber of the detector enables the flow between the two chambers. The mass-flow meter, then, detects and converts the flow into molar fraction of the CO [29]. The analyzer uses paramagnetic method to determine the percent level of oxygen within the sample gas. Unlike most other gases, oxygen is attracted to strong magnetic fields. The analyzer includes a component which generates a magnetic field. Within this component, there exists a mirror connected to two nitrogen filled glass spheres. The attraction of the oxygen into the magnetic field allows its collision with the nitrogen spheres, which in turn causes the mirror to rotate. Light is reflected of this mirror into a pair of photocells to generate a current directly proportional to the mirror’s rotation, which is dictated by the percent oxygen within the sample gas.

Prior to every test, the analyzers were calibrated to ensure accurate measurements. The California Analytical Instrument ZAG-6L Zero Air Generated was used to provide “zero air” to calibrate the NOx and CO analyzers to their zero reference point. A specialty pressurized tank consisting of 300 ppm of NO balanced with nitrogen was used to calibrate the second sample
reference point to generate the linear correlation. For the CO calibration sample, the specialty tank consisted of 9% CO, 22.5% O₂, and also balanced with nitrogen.

A heated hose, maintained at 350°F (177°C), was connected downstream of the emissions probe to prevent water condensation. Then this hose leads the gases to an air chiller, in which the water was condensed and rejected from the sample. The remaining gases are fed into the analyzers for dry-base emission measurements.

4. Test Matrix

The optimal equivalence ratio ranges for the three fuel stages based on the results of atmospheric combustion and preliminary high pressure testing are as follows: the overall equivalence ratio of the pilot flame (central circuit only) ranged from just above LBO (ϕ=0.125) to 0.19. For the 5 cup-staging (central and 4 side circuits) and the 9 cup-staging (all circuits), the ranges tested were from 0.26 to 0.36 and 0.44 to 0.60, respectively. These values correspond to self-sustaining flame within each active swirler. The higher limits for the 5 and 9 cup-staging were reduced for lower pressures due to thermo-acoustic instabilities. Interestingly, the equivalence ratio for the onset of acoustic instabilities increased with the increase in pressure and inlet air temperatures. For example, with inlet air temperature of 400°F, the equivalence ratios for onset of the 9 cup-staging at 2-atm, 3-atm, and 4-atm are 0.52, 0.55, and 0.6, respectively. The on-set equivalence ratio value continues to increase with higher pressures and this phenomenon is discussed in the section regarding thermos-acoustic instabilities.

To best maintain consistency with emission measurements, the relative distribution of the fuel remained the same for the various staging strategies. For the 5 cup-staging, 21% of the fuel
was supplied to the central nozzle and the remaining 79% went to the four side nozzles. For the 9 cup-staging, 12.2%, 43.5%, and 44.4% of the fuel went to the central, side, and corner nozzle(s), respectively. Summary of the operating limit and fuel distribution strategy is shown in table 4-1.

<table>
<thead>
<tr>
<th></th>
<th>φ-Limits</th>
<th>% of Fuel Supplied</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>φ_Lower</td>
<td>φ_Higher</td>
</tr>
<tr>
<td>Pilot</td>
<td>0.125</td>
<td>0.19</td>
</tr>
<tr>
<td>5 Cup-Staging</td>
<td>0.26</td>
<td>0.36</td>
</tr>
<tr>
<td>9 Cup-Staging</td>
<td>0.44</td>
<td>0.6</td>
</tr>
</tbody>
</table>

*Table 4-1: Operating limits and fuel distribution strategy for the three fuel staging*

Overall, the operating pressures ranged from 2-atm to 7-atm. Two inlet air temperatures were tested, which were 400°F and 500°F. For the 400°F cases, the pressures tested were 2, 4, 5, and 7-atm as shown in table 4-2. Whereas for the 500°F cases, the pressures were 2, 3, 4, 5 and 6-atm. A summary of these test of operating conditions tested is shown in table 4-2. Initial plans of testing higher inlet air temperatures were avoided for safety concerns, as the 500°F case led to excessive temperatures of the rig’s shell. However, the data collected proved to be sufficient in understanding the combustor’s emission characteristics. Furthermore, the pressure drop was maintained at 4% of operation pressures for all tests conducted.

<table>
<thead>
<tr>
<th>T₃ (F)</th>
<th>2-atm</th>
<th>3-atm</th>
<th>4-atm</th>
<th>5-atm</th>
<th>6-atm</th>
<th>7-atm</th>
</tr>
</thead>
<tbody>
<tr>
<td>400</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td></td>
<td>✓</td>
</tr>
<tr>
<td>500</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
</tr>
</tbody>
</table>

*Table 4-2: LDI test matrix for inlet air temperatures and corresponding operating pressure*
5. Emission Index Calculations

The emission index was used to characterize the combustor’s behavior. There are various methods of calculating the emission index, which can result significantly different values. For example, Tacina et al. [16] used the following formula to estimate the emission index of NO$_x$ (EINO$_x$):

\[
EI_{NO_2} = \frac{MW_{NO_2}}{MW_{Products}} \frac{1+F/A}{F/A} \times 10^{-3}
\]

The formula considers that the molecular weight of NO$_x$ is the same as NO$_2$. The majority of EINO$_x$ calculations employ this strategy and is sometimes called EINO$_x$ as NO$_2$ or EINO$_2$. The reasoning, aside from simplification and consistency of measurements, is that nitric oxides are expected to react with atmospheric gases and transform into NO$_2$. The problem with equation 5.1, however, is the use of the fuel-air ratio. By using the fuel-air ratio, the EINO$_x$ is then based on the grams of NO$_x$ produced per Kg of fuel injected, rather than fuel burned. Which ultimately does not allow any consideration of combustion inefficiency. Furthermore, this formula does not consider any heterogeneous distribution of the flue gases as the calculation is independent of concentrations of species measured alongside NO$_x$.

The emission index calculations used in this report will instead be based on the concentrations of emitted gases. Based on a health assessment report conducted by JP-Morgan, the average molecular weight of jet fuel is 170 [30]. So dodecane (C$_{12}$H$_{26}$) was used as a surrogate for the Jet-A fuel. The chemical reaction of the overall combustion process will then be:

\[
C_{12}H_{26} + aO_2 + 3.76aN_2 \rightarrow 12CO_2 + 13H_2O + bO_2 + 3.76aN_2
\]
Where $a$ and $b$ are coefficients determined by the fuel-air ratio. The emitted gases were measured on dry-bases. That is, the water was condensed and removed from the products prior to measurements. Based on adapted derivations discussed by Turns [3], $a$ and $b$ are calculated by:

$$a = 12 + \left(1 - X_{O_2,\text{dry}}\right) \times \frac{26}{4}$$

$$b = \frac{X_{O_2,\text{dry}} \times (12 + 3.76a)}{1 - X_{O_2,\text{dry}}}$$

Where $X_{O_2,\text{dry}}$ is the measured molar fraction of oxygen in the sampled gases. Using $a$ and $b$, the molar fraction of CO$_2$ was determined by the following formula:

$$X_{CO_2,\text{dry}} = \frac{12}{12 + b + 3.76a}$$

The EINO$_x$ can then be calculated using the following formula:

$$EI_{NOx} = \frac{X_{NO_x,\text{dry}}}{X_{CO,\text{dry}} + X_{CO_2,\text{dry}} \times \frac{12\text{MW}_{NOx}}{170}} \times 1000$$

Two EINO$_x$ will be presented in this report, one will be based on molecular weight of NO$_x$ being weighted average of NO and NO$_2$, which can be calculated by the following formula:

$$MW_{NOx} = \frac{30 \times X_{NO,\text{dry}} + 44 \times X_{NO_2,\text{dry}}}{X_{NO,\text{dry}} + X_{NO_2,\text{dry}}}$$

The other will simply take the molecular weight of NO$_x$ as NO$_2$, and this will be used as comparison with other combustors. In this report, EINO$_x$ will be the weighted average calculations and EINO$_2$ will be NO$_x$ based as NO$_2$. A comparison of EINO$_2$ values derived from the experimental data using equation 5.6 and Tacina’s formula (5.1) is shown in figure 5-1. Although
there exists slight variation, the increase in accuracy will be more useful in deriving a more representative correlation of emission measurements.

![Figure 5-1](image)

*Figure 5-1: Discrepancies between Tacina’s formula and the before mentioned derived calculation of EINO₂*

The EICO will use the same procedure leading to equation 5.6, but will of course use the measured molar fraction and molecular weight of CO instead of NOₓ. Furthermore, the UHC was measured based on a carbon number of 1, so the EI-UHC will be calculated based on the molar fraction and molecular weight of methane (CH₄).
6. **NO\textsubscript{x} Results and Discussion**

The NO\textsubscript{x} results are organized to initially describe the localized emission measurements based on the three sampling points and its implications on the overall combustion characteristics. Global NO\textsubscript{x} measurements will then be presented with intentions to clarify the effects of pressure, inlet air temperature, flame length, fuel staging, and fuel type on the production of the pollutant. A comparison of NO\textsubscript{x} produced by this LDI concept with data acquired for LPPs and other LDIs tested at similar conditions will follow. Correlations derived from the data collected will then be presented.

6.1 **Localized Emission Measurements**

Sample results of localized emission values with variable fuel staging is shown in figure 6-1. The pilot flame was tested with a global equivalence ratio ranging from 0.12 to 0.2, which equate to 1.02 to 1.8 local values, respectively. The optimal value was considered to be 0.16 (1.44 local), as it is highly stable and does not permit excessive hot spots. However, this means that the pilot flame is essentially acting as an RQL as the local rich flame is diluted to the lean combustion. The quenching process is assumed to be highly rapid due to quick interactions with the cold air flowing through adjacent swirlers. This may mitigate excessive formation of NO\textsubscript{x}, but test results show that relative to the global fuel-to-air ratio, high levels of the pollutant are being formed from this pilot flame. As expected more NO\textsubscript{x} and less O\textsubscript{2} were always present at the central probe location but, surprisingly, not by an excessive amount. The relatively well distributed pollutants reiterate that the surrounding air is well utilized to burn off excess fuel from pilot nozzle.

The major concern with the pilot flame was high levels of CO and UHC. Both these pollutants are highly prevalent in ultra-lean and rich combustion, which make up the pilot flame
operating condition. Although, the transition from rich to lean generates high level of NO\textsubscript{x}, this process is highly useful in burning of excess CO and UHC. As it stands, the pilot flame fairs well in not producing excessive CO and UHC, and the values produced are comparable with RQLs at low power conditions (idle)[1].

Highly irregular and undesirable emission profiles were typically arising due to the 5 cup-staging. In this staging strategy, the equivalence ratios tested ranged from 0.25 to 0.4, which corresponds to local center and side values of 0.52 to 0.77 and 0.48 to 0.71, respectively. The transition to fully lean combustion significantly reduced NO\textsubscript{x}, relative to the global fuel-to-air ratio, but have also led to major combustion inefficiency. The emission profile indicate that substantial amount of fuel is escaping the side circuits’ recirculation zones and entering both the central and corner swirlers’ airfield. This can be seen in figure 6-3, where the central flame is extended and the side flames is exhibiting weak characteristics. This phenomenon has led the central circuit to produce considerably more NO\textsubscript{x}, but more concerning is the amount of CO and UHC entering the corner RZs. It appears that the cold air flowing through the corner swirlers is effectively freezing the pollutants, disallowing them to oxidize. All tests showed the corner probe acquiring much higher CO and UHC, which contribute to combustion inefficiency, but the overall value significantly decreases with higher pressure, equivalence ratio or inlet air temperature. That is, higher pressures reduce dissociation of CO\textsubscript{2} to CO and the higher equivalence ratio and air temperature increase the burning rates leading to more rapid oxidation of the pollutants.
Figure 6-1: Local emissions values corresponding to variable fuel staging (global values in red).
Figure 6-2: Side image of the pilot flame at 2-atm, $T_3 = 500^\circ F$, $\phi = 0.16$.

Figure 6-3: Side image of the 5 cup-staging flame at 4-atm, $T_3 = 500^\circ F$, $\phi = 0.3$. 
The optimal staging condition was through the 9 cup-staging. Sample image of the flame is shown in figure 6-4. This condition produces the lowest NO\(_x\) relative to the fuel-to-air ratio as well as having a more uniform emission profile. The equivalence ratio range tested for the 9 cup-staging is from 0.45 to 0.6, which correspond to central, side, and corner local values of 0.49 to 0.66, 0.44 to 0.59, and 0.45 to 0.6, respectively. Having overall low equivalence ratios across the combustor greatly mitigates hot spots (low NO\(_x\)) as well as create a highly uniform flue temperature profile. Most importantly, very little CO and UHC is emitted through this configuration. Since hot gases are present across the combustor, which include high levels of O\(_2\), almost all CO and UHC oxidize despite any lack of entrainment within the RZs from which they arise. Thus, the 9 cup-staging produces a highly efficient stable lean flame, with low temperature (low NO\(_x\)), and a highly symmetrical temperature profile.
The distribution of the NO\textsubscript{x} and CO in the flue gases is better seen in figures 6-5 and 6-6. The graphs show the percent variance of the pollutants collected in the central probe with respect to the weighted average determined by all three sampling probes. The deviation of NO\textsubscript{x} and CO developed from the pilot flame (0.1<\phi<0.2) has remained within 20% of average values, indicative of a uniform distribution. The deviations of NO\textsubscript{x} increases with the equivalence ratio, while CO decreases. When solely active, the pilot flame tends to recirculate to the adjacent swirler, which can be seen in figure 6-2. This behavior has led to the uniform distribution. The increased hot spots and flame length emanating from the increase of equivalence ratio is believed to be the cause for higher deviation in NO\textsubscript{x}.

The disproportionality of the NO\textsubscript{x} and CO arising from 5 cup-staging (0.25<\phi<0.35) is severe. The amount of NO\textsubscript{x} entering the central probe has reached nearly twice as much as the average value. The lack in uniformity is flame structure as shown in figure 6-3 is the cause for such large deviations. This may be resolved by varying the fuel distribution within the two circuits. That is, increasing the amount of fuel injected through the side nozzles and decreasing the amount injected through the central nozzles. This may be challenging as the central circuit is meant to anchor the flame, but finding a proper distribution is necessary as to increase efficiency of the flame and develop a more uniform temperature profile to mitigate heat stress to the turbine. To maintain consistency in emission measurement, the redistribution of fuel was not undertaken proceeding the discovery of this problem.

As previously mentioned, the distribution of NO\textsubscript{x} is typically very uniform through 9 cup-staging (0.45<\phi<0.6) with typical variance of less than 10% entering the central probe. The cause for this uniformity can be seen in Figure 6-4, which indicate well distribution of the flame structure within all nine swirlers. On the other hand, the deviation in CO is miscued by the graph as typical
values (in ppm) are within double or even single digits, so any insignificant deviation can result in high variance.

![Graph]

Figure 6-5: %-variance of NO\textsubscript{x} to the weighted average (eq. 3.2) based on emissions collected from the central probe for variable pressures and T\textsubscript{3} = 500°F

![Graph]

Figure 6-6: %-variance of CO to the weighted average (eq. 3.2) based on emissions collected from the central probe for variable pressures and T\textsubscript{3} = 500°F
6.2 Effects of pressure on NO$_x$

The molar fractions of NO$_x$ corrected to 15% oxygen, EINO$_x$, and EINO$_2$ for tests utilizing an inlet air temperature of 400°F are shown in figures 6-8, 6-9, and 6-10, respectively and the results for inlet air temperature of 500°F are shown in figures 6-11 to 6-13. The graphs show three distinct patterns for each fuel staging condition. Although the pilot flame operates at the leanest conditions (0.12<$\phi$<0.20), it produces the largest amounts of NO$_x$ per weight of fuel burned. This is due to the flame’s behavior as an RQL (1.08<$\phi_{local}$<1.8), which is known to produce large amount of NO$_x$. It was expected that the most NO$_x$ would be emitted at near stoichiometric local combustion and further increasing the fuel would result in fewer NO$_x$. However, the richer the pilot flame burned, the more NO$_x$ was produced. This indicates that regardless of the fuel flow rate, the surrounding air arising from adjacent swirlers is heavily used in burning of the fuel arising from the pilot flame. This is likely the reason why the LBO exists just above local stoichiometric combustion ($\phi_{local}$≈1.05) and for the uniformity in emission distribution collected across the three probes. So, by increasing the equivalence ratio of the pilot flame, there exists larger hot spots as the fuel transitions away from the central swirler’s air.

[Figure 6-7: Image of the pilot flame at variable pressures with $\phi$=0.16 and $T_3$=500°F]
In general, the pilot flame produced more NO\textsubscript{x} with higher pressures. However, there appears to be excessive amounts of the pollutants when operating at 2-atm for both 400°F and 500°F inlet air temperatures cases and it is not clear why this is occurring. From the images shown in figure 6-7, the pilot flame appears to be less condensed at higher pressures and there is a gradual increase in yellowish flame characteristics, which is indicative of rich combustion. It is likely that the quenching process is delayed with higher pressures and that the high levels of NO\textsubscript{x} at 2-atm is due to excessive hotspots arising from its specific quenching course.

Figure 6-8: Molar fraction of NO\textsubscript{x} corrected to 15% oxygen at variable pressures with inlet air temperature of 400°F
Figure 6-9: Emission index of NO\textsubscript{x} at variable pressures with inlet air temperature of 400°F

Figure 6-10: EINO\textsubscript{2} at variable pressures with inlet air temperature of 400°F
Figure 6-11: Molar fraction of NO\textsubscript{x} at variable pressures with inlet air temperature of 500°F

Figure 6-12: Emission index of NO\textsubscript{x} at variable pressures with inlet air temperature of 500°F
The ranges of EINO\(_x\) for the 5 cup-staging corresponds to almost the same ranges of the 9 cup-staging with respect to operating pressures and temperatures. What appears to be happening is the EINO\(_x\) lacks dependency on the global equivalence and is rather determined by the local operating conditions of the each swirler. The ranges of local equivalence ratios for the 5 and 9 cup-staging are 0.45 to 0.68 and 0.44 to 0.66, respectively. So by distributing the fuel evenly across the combustor and mitigating the temperature gradient, low NO\(_x\) can be produced despite operating with high equivalence ratios.

For the both the 5-cup and the 9-cup flames, a clear NO\(_x\) dependency on pressure is seen. This dependency can be shown by examining equation 1.15 \(\frac{d[NO]}{dt} = k_g [N_2][O_2]^{\frac{1}{2}}\). The concentration of nitrogen and oxygen can be calculated by:
\[ [N_2] = \frac{P \times X_{N_2}}{R \times T} \]
\[ [O_2] = \frac{P \times X_{O_2}}{R \times T} \]

These equations indicate that nitric oxide is more rapidly produced with the increase in pressure. This is conducive with the data, despite points showing higher NO\textsubscript{x} existing with lower pressures. These discrepancies typically arise at the leanest points of both the fuel staged flames, which correspond to the least efficient combustion (discussed in chapter 7). It is probable that experimental errors play a role with these points, but the inefficiency is likely to exacerbate this error.

### 6.3 Effects of Inlet Air Temperature on NO\textsubscript{x}

The increase in air temperatures produced rather interesting and strange results. It was expected that NO\textsubscript{x} formation would increase with the rise of inlet air temperature. Like pressure, higher temperatures lead to increased rate of the chemical reactions needed to create NO\textsubscript{x}. For example, the global reaction coefficient (K\textsubscript{g}) of equation 1.15 is exponentially proportional to temperature. So it was assumed that by raising the inlet air temperature, the adiabatic flame temperature would subsequently increase leading the reactions for nitric oxide formation to accelerate. This was in fact seen at higher pressures as shown in figure 6-14, which is the NO\textsubscript{x} emission index test results using inlet air temperatures of 400°F and 500°F at 5-atm pressure.

On the contrary, the increase in inlet temperatures appears to have decreased NO\textsubscript{x} when operating at lower pressures. This can be seen in figure 6-15, which is the EINO\textsubscript{x} using inlet air temperatures of 400°F and 500°F at 2-atm pressure. Anderson [33] tested an LDI concept with inlet air temperatures varying from 1520°F (1100K) to 1790°F (1250K) at 2.5-atm. He noticed the
same phenomenon of the decrease of NO\textsubscript{x} with higher inlet air temperatures. He attributed this to enhanced fuel vaporization due to higher temperatures, which would ultimately allow better fuel-air mixing of the directly injected fuel. This would decrease the flame length and produce fewer hotspots leading to less NO\textsubscript{x}. His data strongly agrees with the data presented when considering the lower pressures testing. However, the rapid vaporization and fuel-air mixing arising from operating at higher pressures began to overcome any significant affect arising from the increase in inlet temperature. Thus at higher pressures, the increase in temperature have led to an increase in NO\textsubscript{x}.

![Figure 6-14: Emission index of NO\textsubscript{x} with variable inlet air temperatures and inlet pressure of 5-atm](image-url)
Figure 6-15: Emission index of NO\textsubscript{x} with variable inlet air temperatures and inlet pressure of 2-atm

\begin{align*}
P_3 &= 2\text{-atm}, T_3 = 400^\circ\text{F}, \phi=0.47 \\
P_3 &= 2\text{-atm}, T_3 = 500^\circ\text{F}, \phi=0.47
\end{align*}

Figure 6-16: Difference in flame structure due to the increase of inlet air temperature at 2-atm
The effects of inlet temperature on fuel vaporization is more readily seen in figure 6-16, which illustrates the difference in flame structure due to the increase of inlet air temperature at 2-atm. At 400°F, the flame is extended well beyond three inches (cutoff of image) with homogenous characteristics beyond the shear layers. This exemplifies the lack of rapid fuel vaporization and fuel-air mixing needed to entrain the flame zone within the recirculation zones. By increasing the inlet temperature to 500 °F, the flame length is drastically decreased with the majority of the fuel being burned by 1.5 inches of the inlet. Furthermore, the hotspots (indicated by the red arrows) are much vaster with the lower air temperature conditions, which alongside the increased flame length, resulted in more NO\textsubscript{x} being developed. The increase in temperature continued to decrease the flame length at higher pressures. However, the length of the flame was observed to be shrinking with pressure as well (figure 6-23), and so the effects of inlet temperatures became less influential in vaporization and this emanated by the increase of NO\textsubscript{x} due to the increase in temperature beyond 5-atm pressure.

6.4 CHEMKIN Simulation

Asides from operating lean to avoid NO\textsubscript{x} formation, the LDI intends to produces a very short flame to ultimately disallow sufficient residence time for excessive formation of the pollutant. The rate of chemical reactions is increased with pressure and temperature, so the rate to achieve equilibrium NO\textsubscript{x} will increase, counteracting the low residence time flame. A simulated CHEMKIN model of the LDI was developed to understand how the residence time affects NO\textsubscript{x} irrespective of operating temperature or pressure. This was done by maintaining a constant operating pressure, inlet temperature, and fuel-to-air ratio, but changing the overall size of the
flame zone. The residence time was increased with respect to the increase in flame zone as to maintain constant gas velocity. GRI-MECH 3.0 was used for this model, which incorporates 325 reactions mechanisms associated with natural gas combustion. As these mechanisms don’t include any heavy hydrocarbon suitable as a surrogate for jet fuel, the CHEMKIN simulation is not meant to perfectly model the LDI, but rather give insight of un-testable conditions. Methane and propane were instead used as fuel for these simulations. Initial results will define the effect of using various fuels on NOx formation.

Figure 6-17: Schematic of the CHEMKIN model.

The condition modelled is the same as an experimental point operating with inlet pressure and temperature of 3.85-atm (56.45-psia) and 400°F, respectively. The overall equivalence ratio was maintained at 0.6 for all conditions, which resulted in adiabatic temperatures of 1822-K, 1805-K, and 1831-K for Jet-A, methane, and propane, respectively. Figure 6-17 shows schematic of the CHEMKIN model, which consists of three sets of three perfectly stirred reactors (PSR) meant to
simulate the flame zone, and a plug flow reactor (PFR) to simulate the post flame region prior to reaching the probe. Each set of PSRs is meant to simulate the three fuel stages, which include the central, four side, and four corner circuits. The flame volume of the side and corner circuits were 4 times greater than the central circuit, but the residence time was maintained constant. Within each set, three PSRs are used characterize the fuel distribution associated with each injection circuit. PSR-1 takes into account the prefilming on the ventruri surface, which assumes 20% of the total fuel and air flows at $\phi = 0.60$. The temperature of this PSR is maintained at 620°F (600-K), which disallowed combustion and so allowed this reactor to act just as a mixer.

The lack of uniform fuel-air mixture due to direct fuel injection is simplified by the other two PSRs. PSR-2 has an inlet supplying 40% of the total air at $\phi=0.80$, as well as the outlet of PSR-1, which would ultimately assume 60% of the total air operating at $\phi=0.73$. The remaining 40% of the air is supplied to the inlet of PSR-3, which operates at $\phi=0.40$, thus allowing the overall equivalence ratio to be 0.60. After the reaction zone, a nonreactive mixer is used to congregate the flue gases of the PSRs, which proceed to a plug flue reactor (PFR), where any remaining fuel is burned as well as taking account any post flame thermal NO$_x$ formation. Test conditions and NO$_x$ emissions are shown in Table 6-1 and Figure 6-18 illustrates nitric oxide production vs. PFR axial distance from propane, methane, and the actual experimentally measured value for Jet-A.

<table>
<thead>
<tr>
<th></th>
<th>Actual-Jet-A</th>
<th>CK-Methane</th>
<th>CK-Propane</th>
</tr>
</thead>
<tbody>
<tr>
<td>NO (ppm)</td>
<td>47</td>
<td>22</td>
<td>37</td>
</tr>
<tr>
<td>Tad (K)</td>
<td>1822</td>
<td>1805</td>
<td>1831</td>
</tr>
<tr>
<td>$\phi_{\text{global}}$</td>
<td>0.59</td>
<td>0.6</td>
<td>0.6</td>
</tr>
<tr>
<td>Pressure (Kpa)</td>
<td>371.5</td>
<td>371.5</td>
<td>371.5</td>
</tr>
</tbody>
</table>

*Table 6-1: Variance between CHEMKIN and experimental data*
Since the LDI was tested with Jet-A liquid fuel, it is expected to produce more NO\textsubscript{x} than if it was used with a lighter gaseous fuels. That is, the rate of the NNH mechanism and the NO\textsubscript{2}-intermediate mechanism, which tend to be prominent route of NO\textsubscript{x} formation in lean combustion, tend to increase with higher carbon-to-hydrogen ratio fuels [11]. The simulations indicate a 68% rise in NO when transitioning from methane to propane. The results show that not all the fuel is burned within the PSRs, and the remaining amount is rapidly combusted within 5-mm (0.2 in.) of the PFR. Afterwards, approximately 0.85-ppm of NO is generated in the post flame region for both fuels, which correspond to a similar amount calculated for Jet-A using equation 1.15, which is 0.75-ppm. Most importantly, this data shows that for proper comparison, the combustor needs to be compared with other combustors using Jet-A fuel, as other fuel types significantly affect NO\textsubscript{x} production.

\textit{Figure 6-18: Nitric Oxide production due to different fuel type based on CHEMKIN simulation.}
The actual experimental flame was approximately 1-inch long with a residence time of 1.25-ms. Using the CHEMKIN model, the effects of this flame length and residence time were further investigated to see how much impact they have on nitric oxide formation. The model used propane as fuel and modified to determine the amount of nitric oxides formed due to flame lengths of 0.33 in., 0.5 in., 1 in. (baseline), and 2 in. The results are shown in figure 6-19, with the data superimposed on a clearer image of the flame operating at 6-atm and $\phi = 0.47$. 

![Diagram of a reaction process with labels: Inlet Source → Perfectly Stirred Reactor → Plug Flow Reactor → Exhaust.](image)

**Figure 6-19:** CHEMKIN simulation showing the effects of flame length on NO.
The simulation shows a 22% increase of NO formed by doubling the flame length and a 20% reduction by halving it. The flame temperature was maintained below 1860-K for all experimental tests as to disallow any significant post-flame thermal NO\textsubscript{x} from forming. Based on calculations and further reiterated by the CHEMKIN model, post-flame thermal NO\textsubscript{x} is negligible. However, super equilibrium levels of radicals’ O and OH are associated with the flame zone as discussed in section 1.2.5. Since these radicals are heavily associated with thermal NO\textsubscript{x} formation, the increase in residence time have led to further nitric oxide development in the flame zone arising from the Zeldovich pathway.

6.5 Comparison to Other Combustor Concepts

Tacina [34] compiled emission results for various LPP and LDI concepts. He plotted the EINO\textsubscript{2} versus the adiabatic flame temperatures for these combustors. Figure 6-20 shows the results derived from the high pressure testing for the 9 cup-staged flame were superimposed on Tacina’s charts to better illustrate the low NO\textsubscript{x} capabilities of this LDI concept.

Aside from Hussain’s data, the charts indicate that this LDI concept produces fewer NO\textsubscript{x} than the other LDI concepts. Alkabie tested his concept at atmospheric pressure with an inlet temperature of 600-K (620˚F). Anderson tested at 2.5-atm with inlet air temperature to 1100-K (1520 ˚F) and 1250 (1790 ˚F). The data indicated minimal dependency on inlet air temperature, and as previously mentioned, Anderson witnessed fewer NO\textsubscript{x} arising from the increase in inlet temperature due to more rapid fuel vaporization. So the LDI is rather more dependent on pressure and adiabatic flame temperature, rather than inlet air temperature. The data acquired from the 5-atm tests using 500 °F (533-K) inlet air temperatures are highly similar EINO\textsubscript{2} values acquired by the before mentioned authors, with respect to the adiabatic flame temperature. This occurs despite
operating at 5 times higher pressure than Alkabie and 2 times higher than Anderson. Included in the chart in the data from 3-atm and 500°F (533-K) inlet air temperature, which is similar, although slightly higher operating pressures as Anderson. This data indicates fewer NOₓ is achieved as compared to the other concepts, despite the higher operating pressure.

Furthermore, the concept appears to be behaving more as an LPP in terms of NOₓ production. The data Tacina acquired for the LPP is referenced to various types of fuel used. Anderson’s LPP data was acquired using propane, Semerjian and Cooper used Jet-A, and Tang used Hexane. As the type of fuel has significant impact on the amount of NOₓ produced, which was shown in the previous section using CHEMKIN simulation, the comparison of the LDI will only focus on referenced results of Semerjian and Cooper. Semerjian’s data shows the impact of post-flame thermal NOₓ as he varied the residence time prior to collecting emission samples from 2-ms to 4-ms. This can be seen by the deviation in emission index between his two sets of data arising from the increase in adiabatic temperature. Although interesting, Semerjian’s measurements were conducted at atmospheric pressure, which results in the fewest NOₓ production in the chart. It is difficult to infer if his data will reach the levels acquired by Cooper if operating at the same pressure of 3-atm. Regardless, assuming that the trend of NOₓ measured for the LDI at 3-atm continues in the same fashion as it does for the 5-atm case, then the data indicates that the LDI is achieving NOₓ values highly similar to Cooper’s LPP.
6.6 NO\textsubscript{x} Correlations

Based on the acquired data, nonlinear regression was used to correlated the EINO\textsubscript{x} and the EINO\textsubscript{2} to the operating pressure, inlet temperature, and equivalence ratio. The correlations are based on a 4% pressure drop as all tests used this value. The maximum adiabatic temperature tested was 2890\textdegree F (1860-K). This produced a calculated value of 2.18 ppm of post flame NO, which correspond to 2.7% of the overall measured NO\textsubscript{x}. Since post flame thermal NO\textsubscript{x} was mainly avoided during these tests, the correlation will only assume validity for flame temperatures less
than 2960 °F (1900-K). Furthermore, due to the excessively high levels of CO and UHC as a result of combustion inefficiency, a consistent and reliable correlation was not achieved for the pilot flame and the 5 cup-staged flame. The correlations for the 9 cup-staged flame are as follows:

\[
EINO_x = 1.97 \times 10^{-6} \times P_3^{0.527} \times T_3^{0.895} \times \exp(\phi \times 8.245)
\]

\[
EINO_2 = 7.1 \times 10^{-6} \times P_3^{0.517} \times T_3^{0.73} \times \exp(\phi \times 8.52)
\]

Where pressure is in kilopascal and temperature is in Kelvin.

The proportion of variance (R^2) for both equations is 0.93. Plotted in figures 6-21 and 6-22 are the actual experimental values plotted against correlation calculated values for EINO_x and EINO_2, respectively.

![Figure 6-21: Experimental values of EINO_x vs. values calculated from correlation 6.1](image-url)
Results display NO\textsubscript{x}’s dependency on pressure. Maughan et al. [31] tested a well-mixed natural gas combustor at variable pressures. They maintained a range of adiabatic flame temperature between 1227-K to 1310-K. Their results indicated that at higher temperatures, the proportionality of pressure on the rate of NO\textsubscript{x} formation, increases. They attributed this phenomenon to higher percentages of NO\textsubscript{x} being produced by prompt NO\textsubscript{x} and nitrous oxide mechanism at lower temperatures, which are not dependent on pressure. Whereas at higher temperatures, more thermal NO\textsubscript{x} is produced, which is dependent on pressure as shown in equation 1.13, where the concentrations of nitrogen and oxygen increase with pressure. Furthermore, higher pressure and temperatures are associated with faster chemical reactions, resulting to equilibrium levels of pollutants being achieved much more rapidly. The flame produced through 9 cup-staging can be assumed to be well mixed and uniform. The flame temperature of this configuration ranged from 1550-K to 1865-K, which would disallow any impactful post-flame NO\textsubscript{x} production, but according Maughan, this range in flame temperature will result in an approximate square root
dependency to pressure, if the flame is truly well distributed. Based on the nonlinear regression, it was determined that the NO\textsubscript{x} developed by this LDI concept does achieve this proportionality with the pressure exponent of 0.517.

The approximate square root dependency is further justified for lean direct injection by Tacina et al. [16,20] and He et al. [32]. Tacina developed EINO\textsubscript{2} correlations for a 49-point multiplex LDI concept and a 36-point integrated LDI module combustor concept. The exponent of the pressure for the 49-point varied from 0.45 to 0.59 based on three established correlations. For the 36-point sector, the exponent was 0.594. He tested a Parker Hannifin’s multipoint LDI concept at elevated pressures and inlet air temperatures. They noticed a change in NO\textsubscript{x} when the flame temperature reached levels higher than 1810-K, which they attributed to the introduction of post-flame thermal NO\textsubscript{x}. For flame temperatures less than 1810-K, the exponent of pressure for their correlation was 0.46 and for greater than 1810-K the exponent was 0.516. Regardless, the exponents of pressure shown in equations 6.1 and 6.2 are highly consistent with various LDI concepts.

6.7 Summary of NO\textsubscript{x} Results

The combustor exhibited interesting behaviors with respect to the NO\textsubscript{x} it emitted. Three distinct emissions pattern were observed for each fuel staging strategy, which reiterated that the formation of NO\textsubscript{x} is more dependable on local operating equivalence ratios, rather than on the global. To stabilize the pilot flame, a locally rich flame is needed as the air arising from adjacent swirlers are aggressively involved in the combustion process and sufficient fuel is need to allow enough hot gases to recirculate within the central swirler and initiate combustion of the next batch
of fuel and air. This results in an RQL behavior of the pilot flame, with excessive hot spots due to the transition from rich to lean, and thus it was observed that the pilot flame produced the largest emission indices of NO$_x$.

The transition to lean combustion resulted in significant reduction of NO$_x$. The 5 and 9 cup-staging showed highly similar emission characteristics, which is indicative of the fact that both operate at nearly identical local equivalence ratios. The problem arising from the 5 cup-staging is high levels of fuel escaping the side recirculation zones and entering the center. This produced an elongated central flame and a very weak side flame, which corresponded to highly irregular emission profile. It can be inferred from these results that a lack of uniformity in the temperature profile exists as a result of this flame. The well distribution of the flame arising from the 9 cup-staging allowed this configuration to operate very lean across all the swirlers. This resulted in the optimal fuel staging in regards to NO$_x$ as it produced a very short flame which was discrete to each individual swirler.

From the results, it is evident that there are many aspects that contribute NO$_x$ formation. The primary contributors are of course pressure, inlet temperature, and equivalence ratio. These parameters have significant impact on secondary contributors such as fuel vaporization and mixing, flame length and structure, chemical reaction rate, and adiabatic flame temperature. With an increase in pressure, better fuel-air mixing and shorter flame length are achieved, which as shown by CHEMKIN, will result in fewer NO$_x$. However, the impact of pressure on chemical reaction rate leading to NO$_x$ formation heavily outweighs the enhanced fuel-air mixing and shorter flame length achieved. This was clearly shown by the data, as the increase in pressure led to higher levels of NO$_x$. Figure 6-23 shows decrease in flame length and structure with the increase in pressure.
Based on the correlation developed through nonlinear regression, NO\textsubscript{x} is proportional to the square root of pressure. According to Tacina, who has done experimental analysis at much higher inlet pressures (~28-atm) and temperatures (~810-K), the square root dependency appears to not significantly change with operating pressures.

The inlet air temperature significantly aided in fuel vaporization, which in turn led to better uniformity in fuel distribution and mitigation of hot spots. These parameters will lead to fewer
NO\textsubscript{x} production. However, the increase in air temperature also correspond to an increase in chemical reaction rate and adiabatic flame temperature, which are proportional to NO\textsubscript{x} formation. Due to the extended flame existing at lower pressures, the aid in fuel vaporization had significant impact on flame structure leading to a reduction in NO\textsubscript{x}, which was greater than the expected increase due to higher reaction rate and flame temperature. So fewer NO\textsubscript{x} was produced from the increase in temperature at lower pressures. However, as the flame length became small enough due to operating at higher pressure, the impact on vaporization and flame structure from the increase in air temperature became insignificant. This has led to the increase of NO\textsubscript{x} due to higher flame temperatures and rate of chemical reactions, which began to overcome the decrease due to enhanced vaporization and flame uniformity.

When compared to other combustors, this LDI concept appears to be behaving more as an LPP, rather than an LDI. This is likely due to the method of fuel injection, which allows for prefilming on the venturi and impingement of the fuel into the shear layers created by the counter rotating swirlers, as well as adjacent swirlers interactions. As shown in figure 6-24, this has forced rapid fuel vaporization and mixing with the air, which have led to excessively short flames existing within the shear layers. This is of course enhanced by higher pressures and temperatures, but nevertheless, the method of fuel distribution appears to be optimal for NO\textsubscript{x} reduction.
Figure 6-24: Image of the flame at 5-atm, shows a highly condensed burning region indicative of the shear layers as shown by the aerodynamics results.

7. CO and UHC Results and Discussion

Due to direct fuel injection, there were fears that the high momentum of the fuel exiting the nozzles would result in the fuel escaping the burning zones unburned. The combustion work conducted at atmospheric pressure achieved a short blue flame, with minimal yellow diffusion burning. All of which were highly concentrated within the recirculation zones of each swirler. Initial inspection of the flame assumed high efficiency and stability, but were not quantified. The purpose of the carbon monoxide and unburned hydrocarbon measurements were to assess the combustion efficiency.

7.1 Effects of Pressure and Inlet Temperature on CO and UHC

The combustor’s dependency on pressure is further seen from the CO and UHC results. EICO and EIHC for inlet temperature of 400°F are shown in figure 7.1 and 7.2, respectively.
Decrease in CO and UHC were observed with the increase in pressure for all fuel staging conditions except for the 5-cup flame operating at 2-atm. This is likely experimental error arising from improper measurement calibration. Also note that prior to conducting the 7-atm tests, a large offset was witnessed with the UHC analyzer, leading to highly sporadic and inconsistent measurements, and so the data for that condition is neglected. Due to higher burning rates and flame temperature, further decrease in the pollutants was observed with the increase in equivalence ratio within respective to the fuel stages. However, an increase in the pollutants was expected to occur with respect to the equivalence ratio when operating the pilot flame. This is because of the locally rich flame (needed for stabilization) as well as direct fuel injection, which was feared to cause the fuel to bypass the RZs and lead to excessively high levels of CO and UHC. The fact that a decrease occurred indicates the influence of the surrounding cold air on the combustion process. It was observed that the majority of the pilot flame recirculated outwards into the side swirlers, hence the need for locally rich combustion to allow for sufficient hot gases to be recirculated within the central swirler and sustain the flame. So through richer combustion, more of the fuel is being burned in the side swirler’s RZ resulting in higher temperatures and burning rates within them, which ultimately resulted in fewer CO and UHC.

Transitioning to fully lean combustion drastically changed the characteristics of CO and UHC production. The 5 cup-staged flame produced the greatest amount of the pollutants, which indicate the highest combustion inefficiency. This is caused by the freezing of the combustion process generated within the side swirlers by the cold air flowing through the corner swirlers. Burning excessively lean within the side circuit exacerbated this effect. This is due to reduced burning rates within the side circuit, which allowed tremendous amount of the pollutants to escape into the cold corner circuit.
Figure 7-1: Emission indices of CO at variable inlet pressures

Figure 7-2: Emission indices of UHC at variable inlet pressures
However, at $\phi=0.35$, there exists sufficient burning rates to drop the emissions of CO and UHC to acceptable levels within the 5 cup-stages. At this level, the local operating condition is $\phi=0.7$. Conventionally operating at this condition with a single swirler would result in near minimal production of CO and UHC (see section 1.1). So continuing to operate near this level is expected to achieve adequate combustion efficiency, especially when operating at higher pressure and temperatures.

Low levels of UHC and insignificant amounts of CO were achieved with the 9 cup-staged flame. This is impressive considering the highly lean conditions that were tested. This is likely due to the uniformity of the flame within the combustor. Regardless if the pollutants transitioned from its injected circuit to an adjacent recirculation zone, they are bound to a flame region consisting of high levels of oxygen, which will ultimately allow for optimal conditions of CO and UHC oxidation.

Overall, pressure have proved to be extremely useful in mitigating CO and UHC, which is as expected. The increased level of reaction due to higher pressure ensures more of the pollutant oxidize as well as disallow dissociation of CO$_2$. The impact of pressure was greatest on the 5-cup flame. This is because the pollutants only needed to escape into the corner cold air to avoid oxidation, whereas for the 9-cup flame, the pollutants needed to escape axially out of the RZs. That is, it is far more difficult to escape oxidation for the 9-cup flame and so the effect of pressure on CO and UHC will be less severe. As for the 5 cup-staging, the higher reaction rates due to higher pressure will more readily counteract the short amount of time needed for pollutants to escape oxidation within the side circuit prior to interacting with cold corner air.
CO and UHC have also decreased with the increase in inlet air temperature. Just like pressure, the increase in reaction due to higher temperatures will result in a more likelihood for these pollutants to oxidize. Furthermore, higher temperatures will result in enhanced fuel vaporization. Lefvebre [1] noted that a sizable amount of the combustion volume is occupied by fuel evaporation, which ultimately allows less volume for chemical reaction. By evaporating the fuel more rapidly through the increase in temperature, and mixing it by the increase in pressure, there is a higher likelihood for the oxidation of CO and UHC.

![Figure 7-3: Emission indices of CO at variable inlet temperature](image)

Figure 7-3: Emission indices of CO at variable inlet temperature
7.2 Summary of CO and UHC Results

Fear over combustion inefficiency was justified by the emission measurements of CO and UHC. The 9 cup-staged flame produced little of the pollutants despite operating very lean. However, the pilot and the 5 cup-staged flames showed worrying levels of the pollutants, especially at their lower operating limits. To better understand their performance, their efficiency was calculated based on the following formula:

$$\eta_{comb} = 100 - 0.12 \times EIHC - 0.023 \times EICO$$

The equation was taken from Turgut [35] and is used to approximate combustion efficiency. The efficiency results for 400°F inlet air temperature are shown in figure 7-5.
The efficiency for the 9 cup-staging ranged from 98.5-99.96% and was always more than 99% when operating greater than $\phi=0.5$. Furthermore, it reached >99.9% at 5-atm. The 5 cup-staging had an efficiency ranging from 86.9-99.3% with only the greatest point being above 99%. The pilot flame fared much better than the 5 cup-staging, with an efficiency ranging from 95.8-99.3%. Considering the LDI is meant to operate at much higher pressures and inlet air temperatures than what was tested, it is expected that the pilot flame and the 9 cup-staging will continually achieve >99% and >99.9% efficiencies, respectively. However, there is concerns whether the 5 cup-staging will consistently achieve the desirable level of efficiency. There are a few methods to remedy this problem. One method is to simply operate greater than $\phi>0.33$, where higher pressures and inlet temperatures will assure >99% efficiency. The other option is to search for a better performing fuel distribution. Since the distribution of fuel between the central circuit and side
circuits were maintained constant as to better correlate the emissions, it is highly likely that by
changing it, a more efficient flame can be achieved. The new distribution should increase the
amount of fuel injected through the side circuit as to increase the burning rate within it, where it is
believed the inefficiency arises.

8. Conclusion

The data presented characterized emissions and combustion behavior at elevated pressures
and inlet temperatures of an LDI concept developed at the University of Cincinnati. To optimize
flame stability, the LDI was made up of a highly compact swirler array, which would enhance the
turbulence needed for proper fuel vaporization and fuel-air mixing. Due to the fuel injection
directly into the shear layers created by the counter rotating swirlers and by swirler-swirler
interaction, rapid combustion was achieved and was mainly limited to the length of the said shear
layers. The combustor achieved an operating equivalence ratio range of 0.12 to 0.6 when tested at
higher pressures and can reach higher values if needed to accommodate required engine
performance. This was accomplished through three fuel stages (central, side, and corner).

These fuel stages dictated the emissions and combustion behavior of this combustor. By
operating the central pilot flame solely, a rich combustion was needed for stabilization due to the
air from the surrounding swirlers, which actively quenched the flame. This resulted in an RQL
behavior of the pilot flame, which resulted in the largest amount of NO\textsubscript{x} production with respect
to the fuel being burned. By activating the central and side flame (5 cup-staging), the emission
index of NO\textsubscript{x} was shown to highly coincide with that of activating all of the circuits (9 cup-
staging), despite operating much leaner. This phenomenon showed that NO\textsubscript{x} formation is more
dependent on the local operating conditions of the swirler, rather than the global. Furthermore, due
to fuel injection into the shear layers as well as partial prefilming on the venturi, the 9 cup-staged flame behaved highly like a premixed flame and NOx emissions were more consistent with LPPs rather than LDIs.

Combustion efficiency was a major concern because of the direct fuel injection. However, data indicated that the pilot and the 9 cup-staged flame can easily achieve greater than 99% efficiency at engine operating conditions. On the other hand, the 5 cup-staged flame requires higher pressures and inlet air temperatures as well as operating at an equivalence ratio greater than 0.33 to achieve the 99% efficiency. Varying the fuel distribution of this staging strategy can result in better performance. Overall, this LDI exceeded expectations in producing a short, stable, and low NOx flame.

9. Continued Development

9.1 Combustion Dynamics

Although all intended research of this LDI concept has been completed, further research of certain phenomena that were observed could render useful information on thermo-acoustic instabilities. The combustor exhibited acoustic instabilities for the 5 cup-staged and 9 cup-staged flames with all swirler configurations. However, the onset of dynamics varied with operating pressures and temperature. At 2-atm and T3 = 400°F, the onset for the 9-cup flame occurred at $\phi = 0.49$, which is why all tests conducted at this pressure and temperature were limited to that value. At 4-atm and T3 = 400°F, the onset was observed at $\phi = 0.6$. The value continued to rise at higher pressures, however, it is unclear where the exact points of onset are. Furthermore, at 2-atm and T3 = 500°F, the onset was observed at approximately $\phi = 0.52$. Rayleigh’s criteria indicate flame
dynamics will exist when the heat release fluctuation is coupled with the pressure fluctuation. That is:

\[ \frac{1}{T} \int_0^T \dot{Q}'(t)P'(t)dt > 0 \]  

8.1

What is believed to be happening is that the increased combustion reaction (the heat release) rates due to higher pressures and temperatures, drastically disrupts the heat release fluctuation. Pressure and temperature will distort pressure fluctuation as well, but not as greatly as their effect on the heat release. That is:

\[ \dot{Q}' \propto P_3T_3 \] \[ P' \propto P_3T_3 \]  

8.2

This phenomenon is more readily seen in the images of figure 6-23, where a drastic decrease in flame length occurred due to operating at higher pressures with inlet air temperature of 500°F. From the images, the blue flame is extended (≈1.5-2 in.) at 2-atm and begins to gradually decrease in size with higher pressures until almost disappearing at 6-atm. Assuming the heat release mainly arises from the visible flame, then these images indicate the huge influence on the heat release structure due to pressure. Since the heat release structure dictates its fluctuation habits, then it is evident that the operating pressure greatly impacts the heat release fluctuations. Through similar reasoning, temperature has the same type of impact on the heat release fluctuations.

So if Rayleigh’s criterion is fact, then there exists uneven changes of the heat release and pressure fluctuations due to variable pressures and temperatures. It is believed that the greater change is towards the heat release fluctuations, but what has been presented is solely based on observations. Dynamic pressure measurements of the LDI at various pressures and temperatures needs to be conducted to determine their effect on the pressure oscillations. Dynamics were avoided during previous tests due to safety reasons, however, replacing the chamber’s window
with a steel plate will easily allow it to endure the instabilities. Prior to replacing the window, a photomultiplier should be used to better assess the pressure’s and temperature’s effect on heat release.

Furthermore, all tests were conducted with the same percent fuel distribution of the three circuits as to maintain consistency for emission measurements. That is, for the 9 cup-staging, 12.2%, 43.5%, and 44.4% of the fuel went to the central, side, and corner nozzle(s), respectively. Further studies of the effects on dynamics due varying the fuel distribution may be highly useful.

### 9.2 Emissions with Variable Inlet Temperature

The decrease in NO\textsubscript{x} with higher inlet temperatures at low operating pressures should be further investigated. This can be done through atmospheric combustion. It is likely that further increasing the inlet air temperature beyond 500˚F will yield lower NO\textsubscript{x}. However, at a certain temperature, the more rapid fuel vaporization and mixing should cease to have a profound effect on reducing NO\textsubscript{x} generation and there should be a rise in the pollutant beyond this point. The goal should be to find the optimal inlet air temperatures with respect to various fuel-to-air ratios and perhaps continuing to do so with various operating pressures.
References


