Experimental Investigation on The Influence of Liquid Fuels Composition on The Operational Characteristics of The Liquid Fueled Resonant Pulse Combustor

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In this study, the response of a liquid-fueled resonant pulse combustor to changes in liquid fuel composition was investigated. Experiments were performed with gasoline-ethanol, gasoline-diesel, and gasoline-heptane mixtures selected to produce meaningful variations in the ignition delay time. A review of ignition quality tester (IQT) data provided an expected increase in the overall $\tau_{\text{delay}}$ for gasoline-ethanol mixtures with increasing ethanol concentrations, and a decrease for gasoline-diesel mixtures with increasing diesel concentrations in the mixture. By taking the phase of the ion signal as an indicator of heat release timing, the experimental results showed an agreement of gasoline-ethanol cases with the IQT data with a near linear increase with increasing ethanol concentrations. However, for gasoline-diesel, there exit no linear relation with the IQT data. For the case of gasoline-heptane mixtures, the results showed a linear decrease in $\tau_{\text{delay}}$ with increasing heptane concentrations. Furthermore, it was shown that small changes in the physical properties of the fuel can significantly influence the cold-start operation of the combustor and alter the coupling between the unsteady heat release and resonant acoustic pressure wave during resonant operation. Dynamic combustion chamber pressure, stagnation temperature and pressure are recorded after a fixed warm-up time to characterize the performance and operation of the device. Results are interpreted in the context of fuel sensitivity and performance optimization of a resonant pulse combustor for pressure gain turbine applications.
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Chapter 1

Introduction

Gas turbines (GTs) are one of the main sources of electrical power generation in the world. The history of this machinery goes back to 1791, when the first patent for a gas turbine was issued to John Barber in the United Kingdom. However, it was not until a century after that, in 1904, that the first actual gas turbine with an axial compressor was built by Franz Stolze in Berlin. Two years after that, Armengaud Lemale from France created a gas turbine with a centrifugal compressor but with no useful power output. Following that, was the Brown Boveri company in Switzerland, which created the world's first gas turbine for power generation in 1939. Since then, the development of gas turbines has continued with the goal of improving performance by increasing efficiency and thus reducing fuel consumption and emissions. Two main approaches are currently used to increase the overall efficiency of conventional gas turbine engines: increasing pressure ratio in the compressor or increasing the turbine inlet temperature. However, after many years of development using these two approaches, efficiency is getting close to the theoretical efficiency limit, causing the development process to become costly and researchers to seek an alternative. One of the major limitations for gas turbine efficiency can be found in the combustion chamber. Loss in pressure due to the complexity of the fluid path, friction, and other effects limit the thermodynamic availability of the working fluid, which has a significant impact on overall efficiency. The reported pressure loss was between 4-8% of the stagnation pressure delivered by the compressor [8]. One of the state-of-the-art solutions is to replace the conventional Brayton cycle (constant pressure heat addition
process) with the Humphrey thermodynamic cycle (constant volume heat addition process). Furthermore, entropy generation is less with the isobaric process Figure 1.1 which results in additional work output that can be extracted by the turbine.

![Humphrey vs Brayton cycles](image)

**Figure 1.1: Humphrey vs Brayton cycles**

### 1.1 Combustion Regimes in Gas Turbines

When discussing about pulse combustors, it is important first to understand the underlying combustion process and where the concept of pulse combustion originates from. In combustion, two important regimes exist [8]:

- Detonation
- Deflagration

#### 1.1.1 Detonation

Detonation waves are reactive shock waves. They are considered to be high density/high velocity (supersonic) waves with speeds of approximately 1 to 4 km/s. The high burning rate gives the detonation process an advantage over the deflagration process by increasing the overall stagnation pressure. The difficulty comes when coupling such a process with a conventional gas turbine component which is designed to operate in a subsonic, steady environment. In addition, the undiluted exhaust temperature
from these devices exceeds the material limitations of conventional turbine blade materials and the unsteady flow inherent in an isochoric process can cause significant performance reduction in the turbine. Two examples of detonation-based combustion processes are pulse detonation engines (PDE) and rotating detonation engines (RDE). Figure 1.2 shows the Nicholls et al. [1] design for a PDE engine. PDE uses detonation waves to ignite the fuel-air mixture. The term pulse derives from the way in which the mixture is fed into the system after each detonation wave. A pulse detonation engine cycle starts with the injection of the fuel-air mixture into the system. The mixture is then ignited, creating a detonation wave that travels downstream at a supersonic speed. Air is then injected to purge the chamber and the cycle is completed. Utilizing PDEs for conventional gas turbines is currently far from being applied. Creating inlet valves for this system is challenging, given the high frequency that the system operates on, which can reach 1000Hz. RDEs, on the other hand, are similar to PDEs in operation, but RDEs do not purge the chamber as PDEs do. The high velocity wave (continuously cycled around the chamber) makes it more suitable for implementation in GTs than PDEs as the exhaust flow is continuous. Figure 1.3 shows the operation process of RDE.
1.1.2 Deflagration

In deflagration process, the waves propagate at a subsonic speed (resulting in low performance compared to detonation process), and require less than one millisecond to complete 80% of the combustion. The flame propagates through the unburned mixture. The combustion products of hydrocarbon fuel-air mixtures in a deflagration regime have low density and the normal flow velocity is below 1 m/s. Deflagration is utilized in pressure gain as follows. Deflagration is divided into wave rotor engines and resonant pulse combustors (PC). Wave rotor engines have an array of channels arranged around the axis of a cylindrical drum. As schematically shown in Figure 1.4, the drum rotates between two stationary end plates, each of which has a few ports or manifolds, controlling the fluid flow through the channels. Through rotation, the channel ends are periodically exposed to differing port pressures, initiating compression and expansion waves within the wave rotor channels. However, wave rotor engines have a drawback, mainly of a mechanical nature which involves sealing and thermal expansion issues.
The other type of deflagration process is the resonant pulse combustor. The PC uses a thermo-acoustic wave to generate self-compression and natural aspiration, and to initiate combustion. The next section provides a thorough overview of pulse combustors.

1.2 History of Pulse Combustors

The technology of pulse combustion was first explained by Lord Rayleigh [9] who stated that, “If heat be periodically communicated to, and abstracted from, a mass of air vibrating in a cylinder bounded by a piston, the effect produced will depend upon the phase of the vibrating at which the transfer of heat takes place. If heat be given to the air at the moment of greatest rarefaction, or abstracted at the moment of greatest condensation, the vibration is encouraged.”. Some applications of this phenomenon are Rijke tubes and pulse combustors.

1.2.1 The Rijke Tube

A Rijke tube is a thermo-acoustic wave generator. It was first created by professor Petrus Leonardus Rijke, [4] an experimental physicist at the university of Leiden, in 1859. The tube contains wire gauze in the first quarter of the lower half as shown in Figure 1.5. When heating the tube till red hot, air flows upward inside the tube due to convection heating from the gauze. When air passes through the gauze, it heats up and that causes the pressure to rise, which forces the air to flow outward. As a result,
Figure 1.5: Rijke tube operation process [4].

the pressure inside the tube decreases, and the air then flows from both directions into the tube. Some of the air in the lower portion was heated from the previous cycle, so the pressure remains constant. However, the other air is cooled, and when it passes through the gauze, it causes a sudden increase in pressure which thermo-acoustic (sound) waves. The cycle repeats itself until the gauze heat dissipates. Figure 1.5 shows the process graphically.

Pulse Combustor

A pulse combustor consists of three main parts: an air inlet valve, a combustion chamber, and an exhaust pipe. Esnault ans Pelterie [5] were the first people to attempt to design a pulse combustor in 1906 (Figure 1.6). They used a mechanical (flapper) valve for the air inlet section (more about the classification of inlet valves in the next section). However, the first and most well-known application for pulse combustors was the German V-1 Buzz Bomb in World War (Figure 1.7). The pulse combustor was used as a propulsion system. This demonstrates that this technology is not new, and, in fact, combustion-driven oscillations were reported 100 years before the Esnault-Pelterie pulse combustor by Byron Higgins as described by Putnam [5].
1.3 Classifications of Pulse Combustors

There are two main types of pulse combustors [6]:

1. Mechanical Valved Pulse combustors, which are classified into two types:

   (a) Flapper-valved, and reed-valved pulse combustors

   (b) Rotary-valved pulse combustors

2. Aerovalved pulse combustors

1.3.1 Reed (Passive) Valved Pulse Combustor

One of the early designs for a pulse combustor using a passive valve was the Esnault-Pelterie combustor (Figure 1.6), which happens to be the first actual attempt at designing pulse combustors. After that Karavodine (in 1906) received a patent for a flapper valved pulse combustor, Figure 1.8 shows the combustion chamber design
used in the Karavodine pulse combustor. It is important to distinguish between the flapper and reed valves. They are similar in terms of their motion mechanisms and both allow one-way motion of the fluid inside the combustion chamber, which restricts the flow inside the combustor to exit through the tail pipe and creates a forward thrust. However, they differ significantly in their design. The motion of reed valves is controlled by a spring force that keeps it closed when no force is applied, while flapper valves are freely driven by pressure fluctuations and are not normally closed. Figure 1.8 shows one example of a flapper valve.

In the 1946, Tenney and his co-workers developed the Dyna-Jet pulse combustor which is still being produced today by the Curtis Automotive Devices company. The
Dyna-jet red head pulse combustor (Figure 1.10) uses a reed valve, a one-way valve, in which the movement is restricted to inward motion as previously mentioned. This inhibits combustion gases from flowing back to the intake valve during the compression phase of the cycle. The pressure difference between the combustion chamber dynamic pressure and the atmospheric pressure is the driving force that provides sufficient air for the combustion to accrue inside the combustion chamber. However, to start resonant operation, it is necessary to force starting air into the inlet. Once combustion is established, forced air is no longer required and the combustor is capable of operating in a naturally aspirated state. The fuel injector is located upstream of the combustor inside the air venturi inlet. The fuel is mixed with air upstream before being pulled inside the combustion chamber.

![Figure 1.10: Dyna-Jet pulse combustor.](image)

One of the advantages of passive valved pulse combustors is that the resulting dynamic and stagnation pressures are high compared to those demonstrated by current rotary valved pulse combustors. However, reed valve pulse combustors have a downside compared to rotary valves. The operational time is short due to the short life span of the reed itself, as reported in the literature. Reed valve pulse combustors can operate up to half an hour before failure and the resulting failure can be attributed to two causes. First is the physics which occurs during the combustion process which requires the valve to move continuously and rapidly. Second, the resulting temper-
ture values are too high for the reed material to withstand for a longer period of time.

1.3.2 Rotary Valved Pulse Combustor

The second type of pulse combustor is the rotary valve pulse combustor. The most significant advantage of the rotary valve pulse combustor is its robust valve system. In 1971, Muller [10] studied the system of a rotary pulse combustor (Figure 1.11). He used a conical rotary valve and visualized multiple sets of pulse combustors which were timed by the rotary valve to fire in sequence. However, no further work was reported beyond this study. In 2016, Lisanti [11] studied the performance and operational characteristics of a novel actively valved resonance pulse combustor Figure 1.12. He identified three modes of operation where a resonance can be sustained. The operational frequency where resonance occurred were between 200 Hz and 290 Hz. He concluded that for a given fuel mass flow rate the combustion chamber peak pressure increases with decreasing the frequency. Also, for a fixed inlet valve frequency he found that the combustion chamber peak pressure increases with increasing the fuel flow rate.
1.3.3 Aerovalved (Valveless) Pulse Combustor

The main difference between aerovalved pulse combustors and the mechanical valved pulse combustor is the existence of an inlet valve unit. Aerovalve combustors use geometry instead of a mechanical valve to direct the majority of the exhaust products down the tail pipe. The development of the aerovalved pulse combustors began in the 1920s by Tesla. He proposed the use of his “valvular conduit” in aerovalved applications for gas turbines, however, the work didn’t continue after that time. One of the major contributors in this field of pulse combustors is the french aerospace engines manufacturer SNECMA. They started to work in the same year that the V-1 project started in 1943. Three years later, the first aerovalved pulse combustor was in operation. Since then, several designs of aerovalved pulse combustors have been developed. Figure 1.13 shows the different designs of valveless combustors.
1.4 Principle of Operation

All types of combustors use the same principle of operation. They operate under a constant volume heat addition process with a slight difference between them. In general, four phases occur during one complete cycle as shown in Figure 1.14.

- Phase 1 Ignition Process: Fuel and air are brought inside the combustion chamber and the mixture is then ignited using a spark-plug. The use of a spark-plug is limited only to the beginning of the process, and after that, combustion continues for the proceeding cycles using the residual products. This results in a rapid pressure and temperature rise inside the combustion chamber.

- Phase 2 Expansion Process: Due to the rapid expansion of hot products, the inlet valve closes (one-way valve) and the hot gases travel towards the lower atmospheric pressure down the tail pipe.

- Phase 3 Recharge Process: The hot products travels down the tail pipe causing a drop in the combustion chamber pressure which forces the inlet valve to open. As a result, air and fuel are then drawn inside the combustion chamber.

- Phase 4 Compression Process: A portion of the hot product is reflected back from the tail pipe due low pressure inside the combustion chamber. The hot product is then mixed with the fresh reactant increasing its temperature and pressure. The mixture is then ignited and the cycle is repeated.

[Figure 1.14: Pulse combustor principle of operation]
1.5 Advantages and Disadvantages of Pulse Combustors

Although pulse combustors are highly complex devices in terms of the complexity of fluid flow inside the combustion chamber with the interactions between wave propagation and heat release, they have significant advantages which make them a better replacement for conventional combustion chambers in gas turbines. The advantages and disadvantages of pulse combustors as reported in the literature\[6\] are as follows:

1. Advantages

   • Higher combustion efficiency with reduced energy consumption
   • Increased heat and mass transfer rates by a factor of 2-5
   • Reduced NOx and CO pollutant emissions by a factor up to 3
   • Improved uniformity of the temperature field in the combustor
   • Reduced space requirement for the combustion equipment

2. Disadvantages

   • The nature of pulse combustion which is an intrinsic feature of pulse combustors and practically difficult to avoid
   • Vibrations of metal walls of the pulse combustors as flow-induced resonance in the components of pulse combustors and nearby structures can cause intense mechanical vibrations
   • The velocity difference between flue gases flowing out from the combustor and stagnant ambient air.
   • Mechanically valved pulse combustors produce extra noise due to the rapid motion during operation
   • The short life span of the reed valve pulse combustors which results in frequent replacement.
1.6 Operational Characteristics

The phenomenon of combustion driven by pressure drive-oscillations (thermo-acoustic waves) is characterized by the phase relation between unsteady heat release and pressure oscillation. Rayleigh's criterion states that if unsteady heat release occurs at the maximum pressure peak point, the resulting pressure amplitude increases, and if heat release occurs at the point of minimum pressure, the resulting pressure amplitude decreases. A graphical representation by Keller [12] is shown in Figure 1.15. Q represents unsteady heat release and the sinusoidal wave represents the combustion chamber pressure wave. Number 1 in the figure represents the case in which heat release occurs at the point of maximum pressure \((3\pi/2)\) which increases amplitude. Number 2 represents the case in which heat release occurs at the minimum pressure \((\pi/2)\) resulting in a decrease in amplitude. Number 3 represents the case in which heat release occurs at a point after the minimum pressure \((\pi)\) which increases the frequency with no effect on the amplitude. Number 4 represents the case in which heat is released after the peak pressure \((2\pi)\), and frequency decrease with no effect on amplitude.

![Figure 1.15: Graphical representation of Rayleigh's criterion.](image)
Chapter 2

Literature Review

This project utilizes a passive (reed) pulse combustor to study the influence of liquid fuel composition on its operational characteristics and performance. Early designs of the passive valved pulse combustor were first introduced by Esnault and Pelterie [5] (Figure 1.6) who used a reed valve pulse combustor in a gas turbine application. Two combustion chambers worked in phase opposition to achieve more continuous outflow through the nozzle and into the turbine wheel. In 1931, Paul Schmidt patented a pulse combustor in Germany (Figure 2.1). The novelty of his design (which was later known as the Schmidt pulse combustor) is due to the position in which the fuel is injected. He placed the fuel injector downstream of the passive valve instead of upstream of the combustor with a uniform cross-sectional area of the tube as shown in the figure (Figure 2.1). Later, Schmidt designed another pulse combustor firing Benzene fuel, but the project was unsuccessful. This failure led Schmidt to collaborate with Diedrich and Staab. The result was the creation of the V-1 buzz bomb propulsion system. The system consisted of a fuel nozzle venturi set.

Figure 2.1: Schmidt pulse combustor.
The literature lacks studies regarding the impact of fuel compositions on the performance of passive (reed valve) pulse combustors. In 1986, J. O. Keller [13] investigated the response of a pulse combustor Figure (2.2) with a change in the homogeneous chemical ignition delay time. Two gaseous fuels with different ignition delay times were used, pure methane and a mixture of 85% methane and 15% ethane. The results from their study showed a variation in the chemical ignition delay time between the two fuels. These changes in ignition delay resulted in a modification of the phase relationship between the unsteady heat release and the pressure oscillation. The heat release for the methane-ethane fuel mixture occurred earlier compared with pure methane fuel. The combustion chamber pressure for pure methane fuel was 119 kpa, while the methane-ethane fuel mixture was 115 kpa. This alteration also affected the lean stability limit between the two fuels. The lean stability limit for the pure methane fuel was found to be lower ($\phi = 0.473$) than the lean stability limit of methane-ethane ($\phi = 0.522$). The limit for the mixed fuel was defined by insufficient coupling between the unsteady heat release and the pressure oscillation which caused the pulsation to cease. For the pure methane fuel, the premixed flame was extinguished and the steady flame diffusion was re-established.

Figure 2.2: Keller passive pulse combustor.

In 1989, Keller [12] examined the response of a passively valved Helmholtz-type
pulse combustor to changes in the relative timing between the resonant pressure wave and the instantaneous heat release. Four time scales were considered (t_{species}, t_{mixing}, t_{kinetic}, and t_{resonance}). t_{species} represents the characteristic time required to mix the fuel with the oxidizer, t_{mixing} is the characteristic time required for the reactants to mix with products, t_{kinetic} is the characteristic time for combustion to occur, and lastly, t_{resonance} is characteristic resonance time. They assumed that the phase relation between heat release and pressure oscillation is restrained by the magnitude of the total ignition delay time (t_{total} = t_{species}, t_{mixing}, t_{kinetic}, and t_{resonance}) and the resonant pressure wave. Each time scale was studied individually to investigate the magnitude of its contribution to the total ignition delay time. The chemical kinetic time scale was investigated by adding small amounts of N\textsubscript{2} or CO\textsubscript{2} that would significantly lengthen t_{kinetic}. The addition of N\textsubscript{2} resulted at first in an increase in the magnitude of the pressure oscillations until a maximum value was obtained in which the magnitude of P_{rms} start decreasing monotonically. The dilution was found to enhance the coupling between the unsteady heat release and the pressure oscillations. Furthermore, they found that the pressure oscillations and the frequency of operation were a stronger function of phase relationship than the magnitude of the heat release. t_{mixing} was modified by changing the mass injection rate which, in return, changes the injection velocity. Bramlette showed that t_{mixing} scaled with r_o/u_o, where r_o is the jet radius and u_o is the jet velocity. The combustor (Figure 25 add the reference) operated under a premixed condition which eliminated t_{species} from the equation, so the total ignition delay time was assumed to be a function of three characteristic times [t_{total} = f(t_{species}, t_{mixing}, t_{kinetic})].
John and Leng [7], utilized a commercially available non-premixed pulsed combustion heater to study the effect of gas composition on the operational characteristics of the combustor (Figure 2.4). Two gaseous fuel mixtures were used in the experiment, methane-hydrogen and methane-propane and three main operational characteristics were investigated: pressure amplitude, frequency, and phase difference. The experiment was performed in two ways: 1) burning different methane/hydrogen or methane/propane mixtures, keeping the volumetric flowrate of the fuel constant and 2) burning different methane/hydrogen or methane/propane mixtures, keeping methane flowrate constant and gradually adding hydrogen or propane to the fuel. Using pure methane resulted in an increase in the amplitude of pressure oscillations with increasing fuel flowrate. Moreover, the phase difference between the heat release and pressure oscillations decreased when increasing the flowrate of the fuel. Adding a small amount of hydrogen to the fuel caused a decrease in the amplitude of pressure oscillation with an increase in the operational frequency and also an increase in the phase difference between the heat release and pressure oscillations. Furthermore, combustor had a difficulty sustaining the combustion when the phase relation approached 90. On the other hand, when mixing propane with methane fuel, the
performance of the combustor was reduced with a low pressure rise compared with the case of pure methane. Conversely, the operational frequency increased (compared with pure methane) with increasing propane ratio. Finally, the phase angle also had a linear increase with the propane ratio, which is opposite of pure methane, where the phase angle decreased linearly with increasing fuel flowrate (Figure 2.5). Kushari et. al [14], investigated the effect of heat content and fuel composition on the performance of a passively valved pulse combustor (Figure 2.6). Two main fuels where used,

![Figure 2.4: John and Leng [7].](image)

![Figure 2.5: John and Leng [7].](image)
methane and hydrogen, with the addition of helium and CO$_2$ as diluent. The results showed a difference in the behavior of the combustor between methane and methane-hydrogen mixture. Methane fuel showed a slight increase in the pressure oscillation and frequency of operation with heat content. However, increasing the heat content by replacing 20% of methane fuel with an equivalent amount of hydrogen resulted in decreasing the pressure amplitude. Furthermore, frequency showed an increased in the beginning followed by a plateau after. In addition, an investigation of the fuel chemistry was done by keeping the properties of the fluid mechanics constant to determine the effect of chemical ignition delay time on the operational performance. The results showed that a reduction in the pressure oscillation and frequency of pulsation were mainly due to a change in the combustion chemistry once hydrogen was added to the fuel. Also, the addition of hydrogen resulted in an operational increase in the rich limit of the pulse combustor (Table 2.1).

![Figure 2.6: Kushari et al.](image)

<table>
<thead>
<tr>
<th>H2 %</th>
<th>( \phi ) at rich limit</th>
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<tbody>
<tr>
<td>0</td>
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<tr>
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<tr>
<td>30</td>
<td>1.085</td>
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<tr>
<td>40</td>
<td>1.135</td>
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Table 2.1: Kushari et al.
In 2007, Wu Zhonghua and Arun Mujumdar [15] developed a computational fluid dynamic model (Figure 2.7) to investigate the combustion characteristics of a pulse combustor under the use of a wide range of gaseous fuels. The purpose of their study is irrelevant to this study, but the utilization of different types of gaseous fuels in pulse combustors is relevant to the study. The fuels were split into two groups. The first group consisted of gaseous fuels (propane, methanol and butane), ethanol, and fuel oil. Their molecular weights varied from low (methane, propane, and butane) to high, as with fuel oil. The results showed that the frequency, pressure amplitude, average gas temperature inside the combustion chamber, average flue gas velocity and the power and thrust output to power input all showed that this group has similar pulse combustion performance. The differences lie in their specific impulse and the fuel flow rate between the low heating and high heating value fuels. Methanol and ethanol (low heating) tend to have a higher fuel flowrate (0.4789 g/s) than propane fuel (0.1929 g/s). As a result of its self-breathing function, pulse combustors have the ability to automatically adjust the fuel intake under a certain range of fuels, which means that the fuels can be changed with a no major effect on performance.

![Figure 2.7: Wu Zhonghua and Arun Mujumdar](image)

The second fuel group studied was mixed (biogas) fuels. The fuel compositions were mainly methane (55%-75%) and carbon oxide (24%-45%). Three ratios were used in their simulation (70:30, 60:40, 40:60), and were compared with pure methane. Figure 2.8 shows the combustion chamber pressure oscillations for the different ratios.
It is clear that as the ratio of methane-carbon oxide decreased from 100:0 to 40:60, the peak pressure amplitude increased from 15 kpa to 20 kpa which indicates that the combustion process intensified with the ratio being lowered.

Figure 2.8: Wu Zhonghua and Arun Mujmadar
2.1 Research Objective

The reported studies have already established the importance of fuel choice on the performance and operation of acoustically resonant pulse combustors and demonstrated the significant influence that $\tau_{\text{delay}}$ has on the coupling between the pressure wave and unsteady heat release. Most pulse combustors in the literature operate using gaseous fuels and most of studies have focused on the importance of the gaseous fuel composition on the performance of the pulse combustors, such as Keller [13] and Zhonghua [15]. However, there is a lack of study in the literature when it comes to understanding the influence of liquid fuels on pulse combustor operation. Therefore, the objective of this study is to investigate the effect of liquid fuel composition on the operational characteristics and performance of pulse combustors, and to highlight the parameters of importance when utilizing liquid fuels for pressure gain combustion. Furthermore, this study aims to characterize liquid fuels to understand the underlying influence of their physical properties on the cold start operation of the reed valved pulse combustor. To understand the influence of liquid fuels on the operation and performance of the pulse combustor, four different fuels were utilized, two of which are commercially available: ARAMCO 91 octane gasoline fuel and CORYTON US-2D diesel fuel. The second two fuels were ethanol and n-heptane. These were included in order to better understand the effect of changing the fuel auto-ignition delay time.
Chapter 3

Experimental Apparatus

The purpose of these experiments is to determine the influence of the different fuel compositions on the performance and operational characteristics of the passively (reed) valved pulse combustor. A liquid fuel passive valved resonant pulse combustor was utilized to characterize performance.

3.1 Pulse Combustor

The combustor utilized for the experiment is a commercial hobby scale resonant pulse combustor (Figure 3.2) designed by the Dyna-Jet Corporation. The Combustor consists of three main components, the head (or the valve), the combustion chamber, and the tailpipe. The combustor body and exhaust pipe are manufactured from stainless steel 304. The valve is made of aluminum and the deforming component (Reed type) is made of blue steel (Figure 3.1).

Figure 3.1: Reed valve
3.2 Measuring Instrumentation

3.2.1 Ignition System

Combustion was initiated using a high voltage spark plug, Figure 3.3, to provide the sufficient temperature to initiate the process.

3.2.2 Ion Sensor

Ignition delay time was detected by a spark-plug ion sensor. The spark-plug (similar to Figure 3.3) detected the ion signal inside the combustion chamber. A bias voltage was applied to the gap at spark-plug tip. Once ignition occurred, ions were generated filling the gap which results in a drop in resistance, causing the flow of the ion-current signal.

Figure 3.3: Spark-plug
3.2.3 Pressure Measurements

Static pressure, dynamic pressure, and stagnation pressure were measured during the experiments. To capture the time-resolved dynamic pressure in the combustion chamber, the frequency of operation, and the peak-to-peak fluctuations, a piezoresistive pressure sensor type Kistler type 4049A water cooled absolute pressure sensor (Figure 3.4a) was used. The sensor was coupled with a single conditioning amplifier (type Kistler 4665) which was used to amplify the output signal between 0-10V (Figure 3.4b). To simplify the connection of the sensor analog signal (V) a National Instrument shielded connector block (type BNC-2120) was used which then transferred the data to a NI PICe-6363 data acquisition device Figure 3.7a,b. The average static and stagnation pressures were measured at the exit of the pulse combustor.
using a United Sensor Corp. PAC-T-8-K-KL-HT pitot prop (Figure 3.7a). The stagnation pressure is monitored with an Omega PX2300-1BDI differential pressure transducer (Figure 3.6).

![Figure 3.6: Differential pressure transducer](image)

### 3.2.4 Temperature Measurement

The average stagnation temperature was measured using a United Sensor Corp. PAC-T-8-K-KL-HT pitot prop (Figure 3.7a) which was mounted at the centerline of the combustor exit cross sectional area with a Type-K thermocouple (15 sec response time) embedded inside it. The thermocouple used a junction of two different metals (nickel-chromium) that generate temperature-dependent voltage signals (mV) as a result of a thermoelectric effect. A type-K thermocouple was chosen due to its high temperature capability (-200 C to 1250 C) and its ability to withstand a harsh environment. The thermocouple was then connected to a Red Lion (type IAMS0011) transmitter (Figure 3.7b) with a response time of 0.4 sec for the input voltage to eliminate any electrical noise, provide a reference junction compensation, and to amplify the output signal (20V to 0-10 V). The transmitter has also has a built-in function to transform the signal from voltage to temperature. The output signal was then transmitted to the National Instrumentation PICe-6363 data acquisition device.
3.2.5 Emission Analyzer Instruments

Gas samples were carried via heated lines to a HORIBA Continuous Emissions Monitoring (CEM) system. The CEM includes a THERMO (FID ES) analyzer coupled with a HORIBA (VS-E-3005) sample conditioning unit for the measurement of unburned hydrocarbons, a HORIBA (VA-3022) Dual CLD analyzer for NO/NO₂ measurement, and a HORIBA (VA-3116) analyzer for CO/CO₂/O₂ measurement.

The in-situ probe samples were collected along the centerline aft of the ejector. Heated lines (up to 200 Celsius) were used to transport nitrogen as a carrier gas through the GSV and into the sample conditioning unit for the CEM.

Non-Dispersive Infrared Absorption (NDIR)

Concentrations of CO and CO₂ (subscript 2 for co2) were measured with the Dual-beam Non-Dispersive Infrared Absorption Method (NDIR). In this sensor the sample gas flowed through the measurement cell and a beam of infrared energy was directed through the sample gas and onto an infrared detector. Some of the beam energy was absorbed by the measured gas and the concentration was related to the reduction in energy through the Beer-Lambert Law.
**Flame Ionization Detector (FID)**

The presence of unburned hydrocarbons in the exhaust flow was measured with a Flame Ionization Detector (FID). This device detects ions formed during the combustion of organic compounds in a hydrogen flame. The quantitative measurement of these ions allows for the concentration of organic species in the sample gas stream, and its generation of the ions is proportional. Hydrocarbons generally have molar response factors that are equal to the number of carbon atoms in their molecule, while oxygenates and other species that contain heteroatoms tend to have a lower response factor.

**Chemiluminescence Detector (CLD)**

Concentrations of NO are measured with a Chemiluminescence Detector (CLD). In this sensor a reaction between NO and O3 (ozone) emits light which is detected on a photo multiplier tube (PMT). The output voltage is then calibrated to correlate the NO concentration.

### 3.3 Fuel Characterization Apparatus

Characterizing liquid fuels is a crucial part in understanding the influence of the physical properties on operational performance of pulse combustors. Therefore, all binary mixtures were characterized to obtain the physical properties which include distillation temperature, surface tension, density and viscosity.

**Distillation Temperature**

Distillation Temperature for the commercial market fuels (ARAMCO gasoline fuel and CORYTON US-2D diesel fuel) and fuel blends Table 4.4 were measured using the standard ASTM D7344 method via MINIDIS Xpert instrument provided by Garbner.
Instrument. A boiling stone was placed in a sample cup. The cup was then filled automatically with a fuel sample and heated using an electrical heater. The resulting vapor was then measured with a thermo-electric sensor. The condensate volume was measured using an optical system with condenser chamber.

**Surface Tension (SFT)**

Surface tension measurements were carried out by the Wilhelmy plate method using KRUSS K100 Tensiometer provided by KRUSS Scientific. A platinum plate was vertically suspended and injected into a sample using a sample stag motor. When the plate touched the liquid surface a force acted upon the surface. The surface tension was then calculated using a correlation formula that relates the force with surface tension and contact angle.

**Density**

The density of the fuel mixtures was computed mathematically using the known volume fraction of each component in the mixture and its corresponding density.

**Viscosity**

The viscosity of the main fuels was measured using EMS-1000 Electromagnetically Spinning Viscometer which contains two magnets coupled to a rotor which creates a rotating magnetic field that generates a torque that rotates an aluminum sphere injected with the sample. The difference in the angular velocity of the magnetic field and that of the sphere is proportional to the torque. The viscosity is then determined by its linear relation to the angular velocities.
3.4 Experimental Procedure

3.4.1 Fuel Choice

Fuel choice was based on characterizing two main operational periods. First, we aimed to understanding the influence of liquid fuels on the operational characteristics and performance of pulse combustor. Therefore, Ignition Quality Tester (IQT) data, were taken in order to qualitatively compare it with the obtained experimental results. ARAMCO octane 91 gasoline and CORYTON US-2D diesel were used as commercially available market fuels. The addition of ethanol as an octane enhancer and n-heptane as a zero-octane rating were introduced and mixed with gasoline to understand the influence of altering ignition delay time on the coupling between instantaneous heat release and pressure oscillations. Furthermore, all fuels were characterized to understand the mechanism behind the limiting operation of the combustor.

3.4.2 Physical Properties of Base Fuels

The physical properties of the four fuels listed below were provided by their production companies.

<table>
<thead>
<tr>
<th>Test</th>
<th>Result</th>
<th>Test</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average MW</td>
<td>89.794</td>
<td>Density kg/m³</td>
<td>754.73</td>
</tr>
<tr>
<td>Density kg/m³</td>
<td>91</td>
<td>Octane Number</td>
<td>89.794</td>
</tr>
<tr>
<td>Octane Number</td>
<td>754.73</td>
<td>Cetane Number</td>
<td>843.4</td>
</tr>
<tr>
<td>Viscosity at 40 C mm²/s</td>
<td>2.268</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3.1: ARAMCO Gasoline

<table>
<thead>
<tr>
<th>Test</th>
<th>Results</th>
<th>Test</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>MW</td>
<td>46.07</td>
<td>Density kg/m³</td>
<td>684</td>
</tr>
<tr>
<td>Density kg/m³</td>
<td>790</td>
<td>Viscosity at 40 C mm²/s</td>
<td>0.494152</td>
</tr>
<tr>
<td>Viscosity at 40 C mm²/s</td>
<td>1.0139</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3.2: Ethanol

<table>
<thead>
<tr>
<th>Test</th>
<th>Results</th>
<th>Test</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average MW</td>
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<td>Density kg/m³</td>
<td>100.205</td>
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<tr>
<td>Density kg/m³</td>
<td>89.794</td>
<td>Viscosity at 40 C mm²/s</td>
<td>684</td>
</tr>
<tr>
<td>Octane Number</td>
<td>754.73</td>
<td>Cetane Number</td>
<td>843.4</td>
</tr>
<tr>
<td>Viscosity at 40 C mm²/s</td>
<td>2.268</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3.3: US-2D CORYTON Diesel

Table 3.4: Heptane
3.4.3 Fuel Mixing and Characterization Procedure

In this study, gasoline fuel was used as the baseline fuel in which each diesel, ethanol, and n-heptane were mixed individually with an increment of 5% between cases for gasoline-ethanol blends and gasoline-diesel blends and an increment of 10% in the first three cases of gasoline-heptane blends. The rest of the gasoline-heptane blends were mixed with a 20% increment as shown in Table 3.5. Each fuel blend in the table is denoted by the first letter of the fuel followed by a number that represents the volume fraction (v/v%) of that fuel in the mixture (E30 is a 30% ethanol, D5 5% is a diesel, and H15 is a 15% n-Heptane, etc.). Table 3.5 shows the range of mixtures used in the study. For the gasoline-ethanol mixtures, the maximum operational envelope in which the combustor operates is 30%-70%, beyond that the combustion process cannot be initiated. The gasoline-diesel mixtures have an operational limit of 25%-75% and the gasoline-heptane mixtures have an operational limit of 100% gasoline to 100% heptane.

<table>
<thead>
<tr>
<th>Gasoline-Ethanol</th>
<th>Gasoline-Diesel</th>
<th>Gasoline-Heptane</th>
</tr>
</thead>
<tbody>
<tr>
<td>E5</td>
<td>D5</td>
<td>H5</td>
</tr>
<tr>
<td>E10</td>
<td>D10</td>
<td>H15</td>
</tr>
<tr>
<td>E15</td>
<td>D15</td>
<td>H25</td>
</tr>
<tr>
<td>E20</td>
<td>D20</td>
<td>H45</td>
</tr>
<tr>
<td>E25</td>
<td>D25</td>
<td>H65</td>
</tr>
<tr>
<td>E30</td>
<td>-</td>
<td>H85</td>
</tr>
</tbody>
</table>

Table 3.5: Labeling of fuel blends

3.4.4 Pulse-Jet Experimental Operation Procedure

The experiment setup is illustrated in Figure 3.9. To start the experiment, intake air has to be supplied to the combustor which initiates combustion using a high voltage spark plug. The fuel is drawn into the system using a venturi effect which operates on the Bernoullis Principle. Figure 3.8 show the injection mechanism. Pressure inside
the combustion chamber drops below the atmospheric pressure. This drop forces the fuel to inject through a 2 mm hole and mixed with air. The mixture is then forced to be injected into the combustion chamber due the difference in pressure. Once resonant combustion is initiated, the igniter is turned off.

![Figure 3.8: Injection Process](image)

![Figure 3.9: Experimental set-up](image)
3.5 Data Acquisition

Data acquisition taken from the PICe-6363 software was transferred to the LabVIEW software where a code was developed. The code collected a 200000 sample with a rate of 30000 samples per second. The data were cycle averaged to obtain Ion probe and peak pressure curves in order to obtain the operational characteristics.
Chapter 4

Experimental Results

The first part of the experimental results focuses on the analysis of the behavior of the combustor in the cold-start region to understand the relationship between the fuel’s physical/chemical property and the ignition process.

4.1 Fuel Characterization and Cold-Start Operation

The combustor operational limit for each fuel blend set (E-blends, D-blends, and H-blends) is shown in Table 4.4 with gasoline-diesel blends having the lowest operational limit and gasoline-heptane blends having the highest operational limit. Therefore, different physical properties for the utilized fuels were evaluated to understand their influence on the operation of the combustor.

4.1.1 Distillation Temperature

Distillation temperature determines the volatility of the fuel. The most volatile components in the fuel tend to evaporate first. The T10 region in the distillation curve is relevant to the cold start operation [17]. With respect to gasoline-ethanol blends, Figure 4.1 shows that a 10% distilled volume of E5 has the lowest distillation temperature, but when the ethanol content was increased by E10 the distillation temperature also increased. Finally, the T10 temperature at E30 exceeded the gasoline T10 temperature. This is analogous to the "hydrogen bonding dilution effect" [18]. The attractive forces between gasoline (Van der Waals) fuel is weaker than the hydrogen-
bonding forces in ethanol. When mixing a small amount of ethanol with gasoline, ethanol tends to escape more easily due to the weak attractive "Van der Waals" forces between ethanol and gasoline. The increase in the T10 temperature (thus decreasing vapor pressure) for E15 and E30 is related to the presence of large quantities of ethanol in the mixture which corresponds to the presence of hydrogen bonding that makes it difficult for ethanol to evaporate. This could be the reason that resonant combustion could not be established in cases of more than E30 under the cold-start condition. On the other hand, distillation curves for gasoline-diesel blends show similar temperatures at T10 this, which can be related to the less heavy fuel gasoline being evaporated first. Gasoline-heptane cases show an increase at T10 with the increase of heptane content until they reach heptane BP at H85. Temperatures at T10 are listed in Table 4.5

![Figure 4.1: Distillation Curves](image-url)
<table>
<thead>
<tr>
<th>Fuel Blend</th>
<th>T10 (°C)</th>
<th>Fuel Blend</th>
<th>T10 (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>E5</td>
<td>53.9</td>
<td>H15</td>
<td>57.9</td>
</tr>
<tr>
<td>E15</td>
<td>56.6</td>
<td>H25</td>
<td>61</td>
</tr>
<tr>
<td>E30</td>
<td>62.7</td>
<td>H45</td>
<td>77.6</td>
</tr>
<tr>
<td>D5</td>
<td>62.8</td>
<td>H65</td>
<td>91.2</td>
</tr>
<tr>
<td>D15</td>
<td>57</td>
<td>H85</td>
<td>99.5</td>
</tr>
<tr>
<td>D25</td>
<td>57</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 4.1: Temperatures when 10% volume is distilled

4.1.2 Surface Tension and viscosity

To investigate the sensitivity of the pulse combustor operation to physical properties of the fuel, surface tension and viscosity were determined and evaluated (Table 4.5). Heptane fuel was seen to have the lowest surface tension when compared to the other cases, followed by gasoline and ethanol, and a maximum surface tension with diesel. The variation between E-blends corresponds to change in the intermolecular forces, as discussed in 4.1.1. Furthermore, the increase of the ethanol content reduced the $\sigma$ of the mixture. On the other hand, the variation of D-blend cases had a noticeable change on the $\sigma$ of the mixture which can influence the atomization process inside the combustion chamber. It is worth noting that H-blends were not measured due to the low variation of $\sigma$ between heptane and gasoline. Furthermore, Viscosity has an effect on spray form. High viscosity alters the atomization process by increasing droplet size which increases the evaporation rate and thus results in poor atomization [19]. For example, ethanol viscosity is three times that of gasoline and heptane while diesel has the largest viscosity. When comparing those results with the base line fuel gasoline we can see that changes of $\sigma$ had an influence on the cold-start operation and the changes between the viscosity of gasoline fuel compared to Ethanol and diesel may also influence the cold-start operation of the pulse combustor.
Density

The third important aspect of the atomization of fuel/air mixtures is density. In Figure 4.2 the density of gasoline is set as the reference density. It can be seen that H-blends densities lie below the baseline. Diesel and ethanol blends are all above the reference line with a maximum at D25.

4.1.3 Cold-Start Operation

After conducting a characterization of the utilized fuel set, it was found that the passive valved pulse combustor is sensitive to small changes in the liquid fuel composition, specifically in the cold-start region. The obtained results revealed that the cold-start operation is controlled by the physical and chemical properties of the mixture. E-blends showed that resonant combustion could not be established beyond
Figure 4.2: Density of the different fuel mixtures

E30 due to the presence of strong hydrogen-bonding associated with ethanol which affected surface tension forces and made it difficult for the mixture to combust under atmospheric conditions [18]. Combined with $\rho$ and $\mu$, this may have resulted in this operational limit. The chemical properties of the E-blends could also have influenced the operational limit. Diesel is known for being a high reactive fuel compared to gasoline, therefore, the presence of diesel with gasoline increased the reactivity of the mixture. However, when operating the combustor beyond D25, the resonant combustion could not be established, which indicates that the physical properties ($\sigma$, $\rho$, and $\mu$) of the mixture clearly influenced the cold-start operation. Finally, the high reactivity of heptane fuel coupled with low physical properties (low $\sigma$, low $\rho$, and low $\mu$) resulted in operational capability extending to pure heptane.

4.2 Pulse Combustor Experimental Results

4.2.1 Combustion Chamber Peak Pressure

The influence of fuel composition on peak pressure and heat release delay were experimentally investigated. An addition of E-Blends (Figure 4.3) was found to shift the peak pressure further in the cycle and decrease the amplitude. Ion probe signals
showed an agreement with the IQT data [16] with an increase in IDT with increasing ethanol content. Maximum pressure amplitude was found at E5. D-Blends (Figure 4.4) exhibited an agreement with IQT data with a decrease in IDT with increasing diesel content in the mixture. Maximum pressure amplitude of D-Blends was found at D25. The high reactivity of heptane fuel corresponds to fast IDT. The addition of heptane with gasoline resulted in decreasing IDT of the mixture. Figure 4.5 shows agreement with the convention with a maximum pressure amplitude at H25.

Figure 4.3: Combustion chamber pressure vs. heat release as a function of E-blends

Figure 4.4: Combustion chamber pressure vs. heat release as a function of D-blends
Figure 4.5: Combustion chamber pressure vs. heat release as a function of H-blends

4.3 Operational Analysis

4.3.1 Stagnation Properties

Stagnation pressure and temperature were both measured. It was found that the composition of the fuel influenced stagnation pressure Figure 4.6. A reduction in stagnation pressure was found by increasing ethanol blends with gasoline with a minimum at E30. This can be analogous to the coupling between the instantaneous heat release and pressure oscillations. On the other hand, an increase in diesel concentration with gasoline increased stagnation pressure with a maximum at D25. For H-blends, the variation in stagnation pressure was considerably lower compared to E-blends and D-blends. The stagnation temperature was also found to be influenced by the fuel composition. All three blends exhibited the same trend with a decrease in stagnation temperature by increasing their contents in the mixture (Figure 4.7).
Figure 4.6: Stagnation pressure as a function of volume fraction

<table>
<thead>
<tr>
<th>Fuel Blend</th>
<th>Avg. $P_o$ (bar)</th>
<th>Fuel Blend</th>
<th>Avg. $P_o$ (bar)</th>
</tr>
</thead>
<tbody>
<tr>
<td>E5</td>
<td>1.0919</td>
<td>D20</td>
<td>1.0917</td>
</tr>
<tr>
<td>E10</td>
<td>1.0883</td>
<td>D25</td>
<td>1.0927</td>
</tr>
<tr>
<td>E15</td>
<td>1.0862</td>
<td>H25</td>
<td>1.0920</td>
</tr>
<tr>
<td>E20</td>
<td>1.0869</td>
<td>H45</td>
<td>1.086</td>
</tr>
<tr>
<td>E25</td>
<td>1.0838</td>
<td>H65</td>
<td>1.0822</td>
</tr>
<tr>
<td>E30</td>
<td>1.0779</td>
<td>H85</td>
<td>1.0856</td>
</tr>
<tr>
<td>D5</td>
<td>1.088</td>
<td>Heptane</td>
<td>1.0815</td>
</tr>
<tr>
<td>D10</td>
<td>1.0900</td>
<td>Gasoline</td>
<td>1.088</td>
</tr>
<tr>
<td>D15</td>
<td>1.0918</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 4.4: Stagnation Pressure of fuel blends

Figure 4.7: Stagnation temperature as a function of volume fraction
<table>
<thead>
<tr>
<th>Fuel Blend</th>
<th>Avg. $T_o$ (C)</th>
<th>Fuel Blend</th>
<th>Avg. $T_o$ (C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>E5</td>
<td>880.7</td>
<td>D20</td>
<td>913.9</td>
</tr>
<tr>
<td>E10</td>
<td>868.8</td>
<td>D25</td>
<td>873.3</td>
</tr>
<tr>
<td>E15</td>
<td>853.7</td>
<td>H25</td>
<td>865.4</td>
</tr>
<tr>
<td>E20</td>
<td>858.5</td>
<td>H45</td>
<td>854.5</td>
</tr>
<tr>
<td>E25</td>
<td>847.1</td>
<td>H65</td>
<td>821.7</td>
</tr>
<tr>
<td>E30</td>
<td>848.7</td>
<td>H85</td>
<td>839.9</td>
</tr>
<tr>
<td>D5</td>
<td>922.2</td>
<td>Heptane</td>
<td>849.6</td>
</tr>
<tr>
<td>D10</td>
<td>920.7</td>
<td>Gasoline</td>
<td>911.4</td>
</tr>
<tr>
<td>D15</td>
<td>890</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 4.5: Stagnation temperature of the fuel blends

### 4.3.2 Pressure Gain

The importance of implementing pressure gain devices (for example, pulse combustors) in gas turbines is correspondent to the resulted gain in pressure at the exit of the combustion chamber. Since the experiments were performed under atmospheric conditions, pressure gains using average stagnation pressure (Table 4.4) were measured relative to the atmospheric pressure. Table 4.6 shows the maximum pressure gain with each fuel blend. The maximum gain for all cases was found at D25.

Figure 4.8: Average stagnation pressure
### 4.3.3 Frequency of Operation

Rayleigh theory states that if heat release occurs prior to peak pressure the frequency of oscillations will increase. Interchangeably, if heat release occurs after peak pressure, the frequency will decrease. Qualitatively coupled with IQT data [16], the addition of ethanol with gasoline resulted in an increase in the octane number thus increasing ignition delay time. This indicated that the frequency of operation would decrease by increasing ethanol content. Moreover, IQT data also showed that increasing diesel content with gasoline would increase the mixture’s cetane number, thus decreasing ignition delay time which resulted in increasing the operating frequency. With heptane fuel having a zero-octane rating, it is expected that by increasing its content with gasoline, ignition delay time will decrease, thus increasing the frequency. Figure 4.9 show the agreement with IQT data.

![Graph showing frequency of operation as a function of volume fraction](image)

**Figure 4.9: Frequency of operation as a function of volume fraction**

<table>
<thead>
<tr>
<th>Fuel Blend</th>
<th>Pressure Gain %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gasoline</td>
<td>7.9</td>
</tr>
<tr>
<td>E5</td>
<td>8.14</td>
</tr>
<tr>
<td>D25</td>
<td>8.29</td>
</tr>
<tr>
<td>H25</td>
<td>8.24</td>
</tr>
</tbody>
</table>

Table 4.6: Pressure gain
4.3.4 Emissions

Emission measurements displayed a large variation across the evaluated cases. However, even with the existence of bypass flow in the exhaust pipe, the results can still provide useful information. It is very important to consider NOx emissions when analyzing emissions. This radical forms at relatively high flame temperature. The experimental results showed a low concentration for all studied cases. Furthermore, a reduction at E-Blends and H-Blends with increasing volume fraction was observed. UHC emission results for D-Blends showed large concentrations when compared to E-blends and H-blends, which were relatively small. The presence of UHC corresponds to incomplete combustion due to insufficient mixing of fuel with the oxidizer. In the case of CO emissions, the experiments showed large concentrations (200-700 ppm). This can be analogous to incomplete combustion and poor fuel/oxidant mixing. Furthermore, E-blend cases showed a decrease in CO with increasing ethanol v%, which maybe due to one of two causes. First, is the increase in IDT which resulted in reducing the frequency of operation which thus allowed sufficient time for the combustor to force more mass flux into the combustion chamber. Second, the presence of oxygen atoms in the basic form of ethanol may have led to a more complete combustion by increasing its content with gasoline. D-Blends and H-Blends exhibited the same trend as ethanol, however, with low gradient. This maybe be related to the decrease in IDT by increasing the volume fraction of the fuel which reduced the mass flux drawn inside the combustion chamber.
Characteristic Evaporation Time ($t_{\text{evaporation}}$)

Section 4.1.3 highlighted the sensitivity of pulse combustors to changes in the physical properties of the fuel in the cold-start region. However, during the resonant operation, emission measurements revealed that the operation consisted of high levels of carbon monoxide, which is a result of incomplete combustion. Two possible causes for this outcome were discussed in section 4.10. Table 4.7 shows the approximated injection duration for each case. Nearly all cases clustered around 1.3 ms. This outcome leads to the question of how physical properties influenced resonant operation. Going back to the literature, Keller [20] studied the quantification of the two main influencers of total ignition delay time, the fluid dynamics mixing time and the chemical kinetics delay time. He concluded that the fluid dynamic mixing time was on the order of approximately 90% of the total ignition delay time and that homogeneous chemical
kinetic ignition delay time made up the remaining fraction. However, their experiments were performed on gaseous fuels only. With liquid fuels, the ignition delay time is longer due to the presence of the evaporation process. This leads to the possibility of proposing a new characteristic time:

\( t_{\text{evaporation}} \)

This time corresponds to the evaporation rate of liquid fuel during injection and the influence of physical properties of the fuel (e.g., volatility) on evaporation rate can be significant.

<table>
<thead>
<tr>
<th>Fuel Blend</th>
<th>Injection Period (ms)</th>
<th>Fuel Blend</th>
<th>Injection Time (ms)</th>
</tr>
</thead>
<tbody>
<tr>
<td>E5</td>
<td>1.324</td>
<td>D20</td>
<td>1.328</td>
</tr>
<tr>
<td>E10</td>
<td>1.310</td>
<td>D25</td>
<td>1.396</td>
</tr>
<tr>
<td>E15</td>
<td>1.319</td>
<td>H25</td>
<td>1.353</td>
</tr>
<tr>
<td>E20</td>
<td>1.321</td>
<td>H45</td>
<td>1.351</td>
</tr>
<tr>
<td>E25</td>
<td>1.336</td>
<td>H65</td>
<td>1.333</td>
</tr>
<tr>
<td>E30</td>
<td>1.347</td>
<td>H85</td>
<td>1.326</td>
</tr>
<tr>
<td>D5</td>
<td>1.334</td>
<td>Heptane</td>
<td>1.330</td>
</tr>
<tr>
<td>D10</td>
<td>1.332</td>
<td>Gasoline</td>
<td>1.353</td>
</tr>
<tr>
<td>D15</td>
<td>1.317</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 4.7: Injection time of fuel blends
Chapter 5

Conclusion

An experimental study was conducted to analyze the influence of liquid fuel composition on the operational performance of a liquid fuel pulse combustor. Four different fuels were utilized, ARAMCO gasoline fuel, US-2D diesel fuel, ethanol, and heptane. Gasoline was taken as the baseline fuel which was mixed with the other fuels on a volume basis. The operational limit for E-Blends was at E30, D-Belnds at D25, and H-Blends at H-100. The study revealed that pulse combustors are sensitive to small changes in the physical properties of the fuel. Moreover, two regions were investigated, the cold-start regime and the resonant operational regime. In the cold-start regime, physical properties were found to dominate the operation. The resonant operation region agreed with the reported IQT data for the E-blend cases. By increasing ethanol content with gasoline, heat release shifted further in the cycle. In the case of D-blends and H-belnds, heat release shifted earlier in the cycle by increasing the volume fraction of the fuel. Furthermore, pollutant emissions showed that all cases resulted in incomplete combustion with a global decrease in emissions by increasing the volume percent of base fuels with gasoline.
Chapter 6

Future Work

• A study of the influence of fuel composition on the operational performance of pulse combustors under elevated pressures is necessary to investigate the performance characteristics in extreme conditions.

• An experimental investigation for quantifying the characteristic evaporation time and its contribution to the total ignition delay time is necessary to investigate the influence of liquid fuels on the total ignition delay time.
REFERENCES


7 Papers Submitted and Under Preparation
