Narrow-throat Pre-chamber Combustion with Ethanol, a Comparison with Methane

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ABSTRACT

Combustion Characteristics of Ethanol as Main Chamber Fuel with Methane-fueled Pre-chamber

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Pre-Chamber combustion systems are gaining popularity in Internal Combustion Engines (ICE) with the increasing emissions regulations due to their advantages in improving fuel economy by increasing the lean limit and cutting emission, especially NO_x. In pre-chamber Combustion, flame jets shoot out from the pre-chamber orifices into the main chamber and generates several ignition points that promote a rapid burn rate of the lean mixture (air-excess ratio (λ) >1) in the main chamber. This work focused on studying two different fuels in the main chamber, lean limit, combustion efficiency (η_c), and emissions. A single-cylinder heavy-duty engine equipped with a narrow throat active pre-chamber was used. Two fuels were tested in the main chamber, Methane (CH₄) and Ethanol (C₂H₅OH), the first fuel is used as a baseline, while keeping the pre-chamber fueled by Methane only. The engine was operated at a fixed speed, intake pressure, and spark-timing. The amount of fuel injected was varied to attain different global λ , then at each global λ ; the amount of fuel injected to the Pre-chamber was varied to observe the effect of the pre-chamber λ . Different air intake temperatures were tested to see the effect on combustion efficiency. Results from the study showed an increase in the lean-limit using Ethanol in the main chamber compared to using only Methane in both chambers. However, lower η_c than that of the Methane was reported; this is due to a combination of the narrow-throat feature and the high heat of vaporization of Ethanol, η_c showed improvement when the air intake temperature increased.

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.

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Chapter 1 Introduction

1.1 Internal combustion engine history

The internal combustion engine emerged as an energy conversion device in the 1860s. It was Etienne Lenoir (1822-1900), who is responsible for commercializing the internal combustion engine. He patented his engine in 1860 [1], shown in Figure 1.1, his engine had a double-acting piston and did not utilize compression, and it used vaporized gasoline. Lenoir engine's power range was between half and six horsepower, and their reported efficiencies were around four percent. At that time, efficiencies around four percent were the level of steam engines, though it turned out that Lenoir engine efficiency was much lower than that. Around 500 engines were sold during the 1860s which was a success at the time.

Figure 1.1 Lenoir gas engine at the Vienna Technical Museum[2]

Lenoir's engine inspired Nicolaus Otto to design his engine in 1876 [3]. He recognized that he needs to compress the mixture before combustion and to do that, a four-stroke engine was invented. The four-stroke principle is used in almost every combustion engine today.

The four strokes are:

1. **Intake stroke:** The stroke begins at the top dead center (TDC), as the piston is moving down, the intake valve is opened to allow the fresh air-fuel mixture to enter the chamber due to the resulting difference in pressure. As the piston reaches the bottom dead center (BDC), the intake valve is closed.
2. **Compression stroke:** As the piston is moving from BDC to TDC, it works on compressing the charge trapped in the chamber. The end of the compression stroke marks the completion of the first revolution.
3. **Power stroke:** While the piston is at TDC, a spark ignites the compressed charge. The resulting combustion will return the piston to BDC, and hence work is produced.
4. **Exhaust stroke:** Whilst the piston is at BDC, the exhaust valve will open to allow the residual gas resulting from the combustion to exit the chamber. The piston pushes the residual gasses as it moves towards TDC. The second revolution ends by the end of the exhaust stroke. [4]

Otto's engine raised the internal combustion engine efficiencies up to 14% from Lenoir's 4%. Over 5000 engines were sold, which put the internal combustion engine in competition with steam engines.

Rudolf Christian Karl Diesel (1858-1913) started developing his engine in 1893[5], he used Otto's four-stroke principle but with a slight modification. The engine would compress pure air instead of the air-fuel mixture, and injecting fuel at TDC. Instead of the spark, the high pressure and temperature of the compressed air would cause an auto-ignition of injected fuel. Diesel settled on doing an isobaric-combustion instead of an isothermal-combustion, as the latter concept does not use almost 75% of the

available air and hence limit the maximum power. Diesel's engine shown in Figure 1.2 achieved an efficiency of 30%[5], some modern diesel engines can attain an efficiency of 62%[6], marking the diesel engine as one of the most efficient energy converters.

Figure 1.2 First fully functional diesel engine, displayed in the Deutsches Museum, München.[7]

1.2 The Pre-Combustion Chamber

To reduce the effect of global warming, governments and environment agencies have been **increasing emissions** regulations on carbon emissions and other greenhouse gasses[8]. To reduce the greenhouse gasses, one high potential candidate is the automotive industry, and more specifically, the internal combustion engine (ICE). Therefore, the engine research community focused on finding means to reduce ICE

emissions by exploring novel combustion concepts to increase efficiency and consequently decreasing emissions.

While producing the same energy, a higher efficiency ICE will have a lower fuel consumption, and therefore lower carbon emissions compared to a lower efficiency ICE. One way to achieve a higher efficiency ICE is to operate the engine with a lean mixture, that is to say, the air-fuel ratio is higher than stoichiometry. Increasing the thermodynamic efficiency will result in an improvement in the overall efficiency; theoretically, based on Otto cycle efficiency, that can be achieved by two methods [6]. The first is to increase the compression ratio (r_c), but this is limited by heat transfer and friction losses. The second method is increasing the ratio of specific heats (γ), which can be accomplished by increasing the air-excess ratio (λ), put differently operating the engine with a lean mixture.

Although increasing γ will increase temperature and pressure and, therefore, the losses, the higher air-excess ratio will offset heat transfer losses by reducing the temperature of combustion; furthermore, NO_x formation will be reduced. However, if the mixture is extremely lean, cycle to cycle variations would increase because of a significant increase in ignition energy requirement. Increasing the air-excess ratio will slow down the flame speed and make it difficult to ignite.

Thus, higher ignition energy and a more costly system are required to ignite a leaner mixture. Besides, a lack of stability in the ignition system is one of the major causes of unstable combustion. Therefore, in the past years, there has been an interest in ignition system development by engine researchers.

The pre-combustion chamber or pre-chamber is one of the promising means of reducing the ignition difficulties of lean mixtures. In the pre-combustion chamber, the fuel is added and ignited in the pre-chamber, then the partially burned jets shoot out

into the main chamber. The pre-combustion chamber's inner volume is small in comparison to the clearance volume of the main chamber. The main and pre chambers exchange mass and energy through multiple nozzle holes. There are two kinds of Pre-chambers, passive and active pre-chambers. In the passive pre-chamber, there is no separate fuel supply line; therefore, there is not much difference in the air-excess ratio between the main and pre chambers. In such a case, the pre-chamber depends on the lean mixture in the main chamber to flow through the nozzle holes during the compression stroke, to be ignited by the spark plug. In the active pre-chamber, the pre-chamber is fueled by a separate fuel supply line. Usually, the pre-chamber is supplied with a stoichiometric or rich air-fuel mixture to be ignited by the spark plug located in the pre-chamber.

The fuel supplied to the active pre-chamber is not necessarily the same as that of the main chamber. The active type of pre-combustion chambers is used in this study. After ignition, the resulting pressure increase from the combustion event will force the combustion products out through the nozzle holes, and into the main chamber as turbulent flame jets. These flame jets will introduce several ignition points on the surface of the jets; therefore, assisting a distributed heat release. Pre-combustion chamber engines are more stable than standard spark ignition engines when it comes to ultra-lean air-fuel mixtures [9]. This quality is due to the vast number of ignition points compared to the single ignition point of the spark plug.

Reviews of the research in pre-chamber combustion were published by Toulson et al.[10] and Alvarez et al. [11]. L. A Gussak, was one of the early researchers who investigated the effect of pre-chamber design parameters, L. A Gussak et al. [12] concluded that a pre-chamber of the size of two to three percent of the clearance volume and holes area between 0.03 and 0.04 cm² per one cm³ of the pre-chamber volume is optimum for fuel consumption, combustion stability, and power. The effects

of pre-chamber volume and nozzle diameter were also studied by Shah et al. [13] in a single-cylinder engine; the results show a good agreement with Gussak findings. Attard et al. used a light-duty engine to assess the performance of a gasoline-fueled pre-chamber. Very low NO_x emissions were reported while pushing the lean limit to $\lambda = 2.1$. Toulson et al. studied the effect of different fuels in the pre-chamber on the lean limit, combustion stability, and emissions in a CFR engine [14]. They concluded that the main chamber lean limit was primarily affected by the choice of fuel in the pre-chamber, but that is not the case for the emissions. Besides, Toulson et al. [14] showed that using hydrogen in the pre-chamber increased the lean limit the most. Costa et al. [15] used hydrous ethanol as fuel in a passive pre-chamber single-cylinder optical engine. The lean limit was extended to $\lambda = 1.4$, reduction of 22% specific fuel consumption, and 52% in NO_x emissions were reported.

1.3 Why Ethanol?

In order to further reduce emissions, renewable sources such as biofuels offer some advantages over fossil fuel. Ethanol is the most widely used liquid biofuel in the spark-ignition engines and can be made from sugarcane, corn, and other types of crops [16]. Lower CO₂ emissions, when considering the total life cycle, is one of the advantages of using Ethanol, along with faster laminar flame speed. Besides, it has a higher octane number and latent heat of vaporization than that of gasoline, which makes it a desirable fuel for spark-ignition engines. However, Ethanol is known for its problems with cold-start-ability due to the high latent heat of vaporization [17].

The main question to be answered was how a high-octane liquid fuel would affect a pre-chamber engine compared to a high-octane gaseous fuel? The difference between Ethanol and Methane will be studied to answer this question. Many questions will follow the main question, what is the effect of the main chamber fuel on the

combustion in the pre-chamber? Intuition suggests that the combustion in the pre-chamber should be independent of the main chamber fuel. Is the response to the pre-chamber enrichment is the same? What about the level of dilution in the main chamber? How is the liquid main-chamber fuel going to affect the engine efficiencies compared to the gaseous fuel? Hence, the study was conducted to answer these questions.

1.4 Definitions

1.4.1 Global air-excess ratio:

$$\lambda_{Global} = \frac{\left(\frac{A}{F}\right)_{actual}}{\left(\frac{A}{F}\right)_{Stoich.}} \quad (1)$$

The global air-excess ratio is the defined as the actual air-fuel ratio to the stoichiometric air-fuel ratio of both chambers.

1.4.2 Pre-chamber air-excess ratio:

$$\lambda_{Pre-chamber} = \frac{\left(\frac{m_{a \rightarrow}}{m_{f \rightarrow} + m_{f,pre-chamber}}\right)}{\left(\frac{A}{F}\right)_{Stoich.}} \quad (2)$$

Where $m_{a \rightarrow}$ and $m_{f \rightarrow}$ are the mass of air and fuel respectively being pushed from the main chamber into the pre-chamber during the compression stroke.

1.4.3 Gross indicated mean effective pressure:

$$IMEP_g = \frac{\int_{-180^\circ}^{180^\circ} (P \cdot dv)_{main\ chamber}}{V_d} \quad (3)$$

Where V_d is the displacement volume.

1.4.4 Coefficient of variation of :

$$COV_{IMEP_g} = \frac{\sigma_{IMEP_g}}{\overline{IMEP_g}} = \frac{\sqrt{\frac{\sum_{k=1}^N (IMEP_{g,k} - \overline{IMEP_g})^2}{N}}}{\overline{IMEP_g}} \quad (4)$$

Where N is the number of cycles.

1.4.5 Combustion efficiency:

$$\eta_c = \frac{(m_f \cdot LHC_f)_{MainChamber} + (m_f \cdot LHV_f)_{PreChamber} - Q_{emissions}}{Q_{fuel,total}} \quad (5)$$

Where $Q_{emissions}$ is calculated from the measured emission species by the analyzer, the methodology is detailed in [18].

1.4.6 Thermodynamic efficiency:

$$\eta_T = \frac{IMEP_g}{QMEP} = \frac{IMEP_g}{FuelMEP \cdot \eta_c} \quad (6)$$

Where QMEP is the heat mean effective pressure, a quantity to represent the heat generated by combustion. It can be noticed, that decreasing the combustion efficiency while keeping $IMEP_g$ and FuelMEP constant, will increase the thermodynamic efficiency.

1.4.7 Gross indicated efficiency:

$$\eta_{i,g} = \frac{IMEP_g}{FuelMEP} = \eta_c \cdot \eta_T \quad (7)$$

1.4.8 Flame development angle

The Flame development angle is defined as the difference between crank angle degree (CAD) of spark timing and CAD of 10% of accumulated heat release.

1.4.9 Combustion phasing (CA50)

Combustion phasing (CA50) is defined as CAD of 50% of accumulated heat release.

1.4.10 Combustion duration

Combustion duration is defined as the difference between CAD of 10% and 90% of accumulated heat release.

Chapter 2 Experimental Setup and Methodology

2.1 Experimental setup

2.1.1 *The Engine*

The set of experiments were carried out on a D13C500 Volvo Truck engine, shown in Figure 2.1, which was modified by MESA Engine Solutions into a single-cylinder research engine. The original diesel six-cylinder engine was converted into a single-cylinder engine by disabling the other five cylinders. The compression ratio was set to 11.5 for the Pre-Chamber experiments by installing a new piston with a different shape to avoid auto-ignition. The experimental engine specifications are shown in Table 2-A. An intake manifold with two pockets for port fuel injectors was designed and installed for injection of the liquid main chamber fuel. A pressure regulator was used to reduce the lab air supply line from pressure of 45 bar to the required intake pressure of one and a half bar and is fed to the engine. An ABB electrical motor was used to operate the engine at a constant speed and, at the same time, act as a dynamometer. A LabView program and a NI system were used to run the engine and collect the measured data from the engine. Additionally, a Horiba MEXA 1700 exhaust gas analyzer was used to measure the emissions.

Figure 2.1 KAUST Heavy-duty metal engine, Volvo D13C500

Table 2-A Engine Specifications

Engine Model	Volvo D13C500	
Piston Shape	Bowl-in-piston	
Valve Mechanism	Single Overhead Cam	
Number of Valves	2-intake 2-exhaust	
Bore	131	mm
Stroke	158	mm
Connecting Rod Length	255	mm
Compression Ratio	11.5	[-]
Displacement Volume	2.1	Liters
Intake Valve Timing	Open	-330 CAD aTDC
	Close	-130 CAD aTDC
Exhaust Valve Timing	Open	180 CAD aTDC
	Close	-340 CAD aTDC

2.1.2 The Pre-Chamber

The original engine had a diesel injector mounted inside the diesel injector pocket at the combustion chamber's center. Usually, the pre-chambers features a larger throat diameter than that of the typical diesel injector diameter to reduce the pressure drop associated with flow losses and to prevent choking. Subsequently, the central injector hole is usually drilled into a larger diameter to allow for the larger diameter pre-chamber. Nevertheless, the KAUST narrow throat pre-chamber was designed with minimizing changes on the engine cylinder head in mind; in order to examine the possibility of a quick upgrade of the existing conventional diesel [19]. To minimize the engine cylinder head changes, the pre-chamber and all of its auxiliaries must be able to fit within the injector pocket, which led to many constraints on the pre-chamber

assembly. The final design of the pre-chamber is somewhat unique compared to other pre-chambers described by literature since it features a narrow throat design, as shown in Figure 2.2.

Three volume classes were designed to study the effect of pre-chamber volume; each was denoted by a letter (A, B, and C). Moreover, three sub-classes were generated to study the effect of the area of the holes; a number denoted each sub-class. The sub-classes were designed based on the area of the holes to the pre-chamber volume ratio. KAUST Narrow-throat Pre-chamber matrix is shown in Table 2-B.

Table 2-B KAUST Narrow-throat Pre-chamber Matrix

		A	B	C
	% of clearance Volume (V_c)	1.6	2.5	3.4
	Actual Pre-Chamber Volume [cm^3]	3.245	5.070	6.896
	Nozzle Area to Volume ratio $\left[\frac{\text{cm}^2}{\text{cm}^3}\right]$	Nozzle Diameter [mm](12 holes)		
1	0.025	0.93	1.16	1.35
2	0.035	1.10	1.37	1.60
3	0.045	1.24	1.56	1.81

The classes and their sub-classes were determined according to the findings of Gussak [12], where it was concluded that the optimal pre-chamber volume to main-chamber volume ratio is from 2% to 3%, and the area of the optimal holes to pre-chamber volume to be from $0.03 \text{ cm}^2/\text{cm}^3$ to $0.04 \text{ cm}^2/\text{cm}^3$. This is only true for the B2 configuration.

Thus, B2 configuration was used for both fuels, Ethanol, and Methane. B2 inner volume is 5.070 cm^3 , which corresponds to 2.5% of the clearance volume; the area of the holes to the pre-chamber volume is $0.035 \text{ cm}^2/\text{cm}^3$, the B2 configuration is shown in Figure 2.2.

Figure 2.2 KAUST Narrow-throat pre-chamber, B2 configuration (left: front view, right: section view)

The pre-chamber assembly consists of three main parts: The pre-chamber, the adaptor, and the enclosure. The pre-chamber assembly is presented in Figure 2.3.

Figure 2.3 The pre-chamber assembly [19]

The pre-chamber itself is made of INCONEL 625 to handle the high-temperature loads that could be subjected to it. To meet the area of the holes to the pre-chamber volume ratio, and due to the small diameter of the pre-chamber tip, the number of holes was decided to be twelve, which is divided into two rows, six holes in each row. The two planes were phased with an angle of 15 degrees to improve the distribution of the pre-chamber jets.

The pre-chamber is screwed into an adapter. The adapter is an integral part of the assembly as it connects the pre-chamber to the spark-plug, piezoelectric pressure sensor, and the fuel supply line. The adaptor was made out of bronze; such material will allow the pre-chamber to screw tightly into the adapter and to prevent leaking. Additionally, the bronze has a sufficient thermal conductivity which assists in transferring heat from the pre-chamber during combustion. A miniature check-valve is installed in the fuel supply passage to prevent backflow during the pre-chamber combustion event. The fuel supply line connects the adapter to a solenoid gas injector.

The adaptor is connected to the enclosure with the help of four small headless screws. The enclosure purpose is to protect the other pre-chamber assembly components from the fuel gallery within the cylinder head. Similarly, the enclosure is sealed from the top to prevent the lubricating oil from flooding the assembly. It is also used to fix the pre-chamber assembly in its place by clamping it down from the top.

2.1.3 The Fueling system

Figure 2.4 Experimental set-up: (1) Engine, (2) Intake manifold, (3) PFIs, (4) Common rail, (5) Pre-chamber, (6) Cardan shaft, (7) Dynamometer, (8) Back pressure valve, (9) EGR cooler, (10) EGR valve, (11) Main air valve, (12) Pressure regulator, (13) Control air valve, (14) Intake throttle valve, (15) Air flow meter, (16) Air heater, (17) Mixing tank, (18) Main Methane valve, (19) fuel filter, (20) Valve, (21) Pressure damping vessel, (22) Mass flow controller, (23) Pre-chamber injector, (24) HORIBA exhaust analyzer, (25) Vent, (26) Ethanol tank, (27) Fuel balance, (28) pump.

The engine utilizes two fuel systems in order to fuel the engine. First, to fuel both the pre and the main chambers with Methane, the gas fuel supply line was divided into two lines, each of them will fuel a separate chamber. In order to reduce pressure fluctuations, each line goes into a damping cylinder. The line out of both damping cylinders is going into a mass flow controller to control and measure the mass rate of fuel going to the engine. After each mass flow controller, there is an injector responsible for fueling its particular chamber. The system was fitted with A SP-021 injector from the Clean AirPower to fuel the pre-chamber. For the main chamber, the

system was fitted with Two Bosch NGI-2 Port Fuel Injectors (PFIs) in the intake manifold. To change the main chamber fuel to Ethanol, two Bosch EV14 PFIs will replace the Methane PFIs in the intake manifold. They are supplied with liquid fuel through a common rail. A submerged pump is used to pump the fuel from a tank to the common rail with a low pressure of 4 bar. To measure the fuel flow from the tank, a precise mass balance was used. A throttle-controlled valve is installed in the intake line and is kept in an adjustment such that the intake line pressure is 1.5 bar.

2.2 Operating Conditions and Methodology

The methodology used in the Methane study by Hlaing et al. [20] was used to allow for comparison between using Ethanol and Methane as the main chamber fuel. The amount of fuel injected in both chambers was gradually decreased to achieve lean conditions while keeping the air intake pressure constant at 1.5 bar. Likewise, the exhaust pressure was kept constant at 1.7 ± 0.02 bar. At each global air-excess ratio, the pre-chamber air-excess ratio ($\lambda_{\text{pre-chamber}}$) was varied from rich to lean conditions. The methodology is essentially a load variation that covers low to medium loads.

Methane with 99.5% purity was used in the pre-chamber, and Ethanol-water solution with a concentration of 96 % by volume was used in the main chamber. Methane with the same purity was used for both chambers in the referenced paper. Both Fuels were injected at the gas exchange TDC. An intake and exhaust valve overlap of 10 CAD which can be negligible. The injection pressure of Methane was set at 7 bar, while the main chamber fuel injection pressure was set at the common-rail pressure of four bar. Both Methane and Ethanol experiments were carried out at a fixed engine speed of 1200 rpm. The engine was operated with an air intake temperature of 30°C for both groups of experiments and was additionally varied from 30°C to 70°C for some Ethanol data points to see the effect on combustion efficiency.

The lean limit was set at five percent of the coefficient of variation of gross indicated mean effective pressure (5% CoV of IMEP_g). Knock intensity of half a bar was specified as the knock limit, and the knock intensity was defined as the maximum amplitude of the main chamber pressure signal after band-pass filtering. For Both Ethanol and Methane experiments, the spark ignition timing was fixed at -15 CAD aTDC, the operating conditions can be found in Table 2-C.

Table 2-C Operating Conditions

Engine Speed	1200	RPM
Coolant Temp.	78 ± 2	°C
L.O Temp	80 ± 2	°C
Intake Temp.	30 ± 1, 60 ± 1, 70 ± 1	°C
Intake Press.	1.5 ± 0.01	bar
Exhaust ΔP	0.2 ± 0.01	bar
MC Fuel	Ethanol (96% purity)	
MC Fuel SOI	-360	CAD aTDC
MC Fuel Press.	4	CAD aTDC
PC Fuel	Methane (99.5% purity)	
PC Fuel SOI	-360	bar
PC Fuel Press.	7	bar

2.3 Heat Release Analysis

The heat release analysis in pre-chamber combustion is more complicated than that of other modes due to the presence of two volumes and the mass transfer between them. Heat release of two-chambers has been studied by Duong et al. [21], applying a first-law analysis of the two control volumes based on pressure measurements in the pre-chamber and main chamber. Following the same approach in [21], the first law analysis will produce the following rate of heat release equations for both main and pre chambers

$$\frac{dQ_{MC}}{d\theta} = \frac{\gamma \cdot P_{MC}}{\gamma - 1} \frac{dV_{MC}}{d\theta} + \frac{V_{MC}}{\gamma - 1} \frac{dP_{MC}}{d\theta} - C_p \cdot T_{PC,MC} \frac{dm}{d\theta}$$

$$\frac{dQ_{PC}}{d\theta} = \frac{V_{PC}}{\gamma - 1} \frac{dP_{PC}}{d\theta} + C_p \cdot T_{PC,MC} \frac{dm}{d\theta}$$

The subscripts (MC) and (PC) refer to the main chamber and pre-chamber, respectively. The pressure (P) was measured using pressure sensors in both chambers. The main chamber volume (V_{MC}) was estimated with the help of an encoder, while the pre-chamber volume (V_{PC}) is a fixed quantity. In order to calculate the specific heat ratio (γ), along with heat capacity at constant pressure (C_p), the Mass fraction of the fuel in the main chamber and temperature of the mixture were used. The temperature of the mixture was determined by using the ideal gas law and the main chamber pressure data recorded at the inlet valve closing. The following equation can be used to calculate the mass transfer between the two volumes [21]

$$dm = \frac{C_d \cdot A_T \cdot P_{PC}}{\sqrt{R \cdot T_{PC}}} \left(\frac{P_{MC}}{P_{PC}} \right)^{\frac{1}{\gamma}} \left(\frac{2\gamma}{\gamma - 1} \left(1 - \left(\frac{P_{MC}}{P_{PC}} \right)^{\frac{\gamma-1}{\gamma}} \right) \right)^{1/\gamma}$$

The mass transfer rate (dm) sign is positive when the mass is transferred into the main chamber from the pre-chamber and vice versa. The sign of the mass transfer rate can be found by observing the pressure difference (ΔP) between the two chambers; namely, dm is positive when ΔP is positive. The pre-chamber temperature should be used as long as the mass is transferred from the pre-chamber[18]. Likewise, the main chamber temperature is used if the mass is transferred into the pre-chamber.

If the pressure difference (ΔP) is negative, that is to say; the pre-chamber pressure is less than the main chamber pressure. Then, the pressure ratios in the equation should be inverted, and the pressure and temperature in the first term in the equation are the main-chamber pressure and temperature. The pre-chamber's discharge coefficient is denoted by (C_d) and is calculated using a model in GT-POWER [19]. The

total area of the holes is denoted by (A_T). The complete heat release derivation can be found in a paper published by Duong et al. [21].

The total heat release, however, can be calculated as follows

$$\frac{dQ_{total}}{d\theta} = \frac{dQ_{MC}}{d\theta} + \frac{dQ_{PC}}{d\theta} + \frac{dQ_{HT}}{d\theta} + \frac{dQ_{Cr}}{d\theta} + \frac{dQ_{Bl}}{d\theta}$$

The first two terms are the heat release rate of the main and pre chambers. The heat transfer losses (dQ_{HT}) were calculated based on the Woschni heat transfer model [22], while the crevice losses (dQ_{Cr}) and blow-by losses (dQ_{Bl}) were tuned to yield a zero heat release during the motoring cycles.

2.4 Estimation of Pre-chamber Air-Excess Ratio

Determining the Pre-chamber's air-excess ratio ($\lambda_{pre-chamber}$) during the spark-timing is not a straight forward task. Two estimations need to be applied; first, the amount of fuel mass that is trapped inside the pre-chamber after injection. Second, the amount of air and fuel that is pushed from the main chamber into the pre-chamber during the compression stroke until spark timing

When a high amount of fuel is being injected into the pre-chamber, the injected fuel's total volume starts to exceed the pre-chamber's inner volume, and hence the some of the injected fuel would be spilled into the main chamber. Hlaing et al. [20] presented two possibilities that could happen to the spilled fuel, based on terminologies borrowed from two-stroke engine gas exchange:

- 1- **Perfect mixing of spilled fuel:** As the spilled fuel reaches into the main chamber, it mixes instantaneously with the main chamber air-fuel mixture.
Furthermore, the pre-chamber can only trap an amount of fuel that is

equivalent to or less than its actual inner volume, and any amount of fuel more than that is lost in the main chamber mixture.

- 2- **Perfect displacement of spilled fuel:** The spilled fuel is pushed back into the pre-chamber during the compression stroke. Thus no fuel is lost to the main chamber, and the pre-chamber recovered all of the spilled fuel.

The latter was used in this paper to simplify the calculations since the spilled fuel cannot be measured in this engine set-up. Although in reality, none of the two ideal scenarios is achievable, and the real case should be in between.

Chapter 3 Results and Discussions

In order to compare Ethanol with methane, some Ethanol operation conditions tested were the same as the Methane operation conditions. Then to explore the lean capabilities of Ethanol, some data points were taken at leaner global air-excess ratios. Ethanol tested conditions and the corresponding Methane data reported by Hlaing et al. [20] are shown in Figure 3.1.

Figure 3.1 Ethanol and Methane operation conditions tested.

Twenty-two points were collected at an air intake temperature of 30°C; the intake pressure was kept constant at 1.5 bar. In the first part, typical results of Ethanol alone will be discussed and will be followed by 2D contour plots of pre-chamber air-excess ratios against the global air-excess ratios. Indicated mean effective pressure (IMEP), combustion characteristics, emissions, and efficiency will be presented. In the second part, a comparison between Ethanol and Methane for different criteria will be presented. Likewise, IMEP, combustion characteristics, emissions, and efficiency will be described.

3.1 Ethanol as a main chamber fuel

Figure 3.2 shows a typical Ethanol pressure trace, the high pre-chamber pressure raise is due to the narrow throat feature. The main chamber peak pressure depends on the main chamber air-excess ratio; the leaner the charge the lower the peak.

Figure 3.2 Typical Pressure traces in the main and pre chambers.

3.1.1 Rate of Heat release

Unlike the rate of heat release reported by Hlaing using methane only, with methane in the pre-chamber and ethanol in the main chamber, the rate of heat release only shows two stages of heat release instead of three stages. A typical rate of heat release of Ethanol combustion is shown in Figure 3.3. The two stages, as shown in Figure 3.11, are (a) initial heat release stage and (b) is the main heat release stage. The post-burning stage was not observable in all of the data points that were collected; this is

mainly due to the air-excess ratio of ethanol data being too lean. The point at $\lambda_{\text{Global}}=2$ and $\lambda_{\text{Pre-chamber}}=0.4$ can be chosen to represent the data set in terms of the shape of the rate of heat release.

Figure 3.3 Rate of heat release (top), the difference in pressure between the pre and main chambers (bottom). Stages of main chamber heat release: (a) Initial heat release stage, (b) bulk heat release stage, $\lambda_{\text{Main chamber}}=2.2$

The initial heat release rate in the main chamber starts when the pre-chamber jets penetrate the main chamber, the jet penetration can be detected when the difference in pressure between the pre and main chambers starts to increase. The initial heat release stage is designated by (a) in Figure 3.3. Shortly, the rate of heat release stops increasing, and this marks the end of the initial heat release stage. During this stage, the pre-chamber rate of heat release becomes negative, indicating an energy flow from the main chamber into the pre-chamber due to combustion in the main chamber. The bulk heat release stage -designated by (b) - follows soon after with a second, not as the rapid increase in the rate of heat release. This stage is where most

of the heat release occurs. A negative difference in pressure between the two chambers indicates that there is flow coming into the pre-chamber from the main chamber due to the main chamber combustion and hence pressure increase.

3.1.2 Combustion Characteristics

The combustion phasing (CA50) defined as CAD at 50% of the accumulated heat release, shown in Figure 3.4. CA50 is dependent primarily on the global air-excess ratio alone as increasing going leaner in the global air-excess ratio will retard CA50.

Figure 3.4 combustion phasing (CA50), CAD of 50% of accumulated heat release for different global and pre-chamber air-excess ratios.

The combustion duration identified as the difference in CAD between 10% of accumulated heat release and 90% of accumulated heat release is shown in Figure 3.5. It is found to be a strong function on the global air-excess ratio; increasing λ_{Global} showed to be slowing down the combustion process as the combustion duration

(CA10-90) is taking a long time to the completion of the combustion, mostly because of slowing down of flame speed.

Figure 3.5 Combustion Duration (CA10-90), the difference between the CAD of 10% of accumulated of heat release and the CAD 90% of accumulated heat release

3.1.3 $IMEP_g$

The gross indicated mean effective pressure was shown to be mostly dependent on the global air-excess ratio and not that much on the pre-chamber air-excess ratio. Going leaner in the overall air-excess ratio showed to be reducing IMEP as the overall energy added is reduced, as shown in Figure 3.6. Also, there is a weak dependence on the $\lambda_{Pre-chamber}$ as going leaner in the pre-chamber increased the gross indicated mean effective pressure.

Figure 3.6 gross indicated mean effective pressure ($IMEP_g$) for different values of pre and global air-excess ratios

3.1.4 Emissions

Unburned hydrocarbons (HC), and Nitrogen Oxides (NO_x) emission are shown in Figure 3.7, and Figure 3.8, respectively. The HC and Carbon monoxide (CO) emissions showed a strong dependence on the overall air-excess ratio. In both cases, the maximum amount of emissions occurs at the ultra-lean points. This can be attributed to temperatures being too low in the ultra-lean conditions. The pre-chamber air-excess ratio did not affect HC significantly, as shown in Figure 3.7. Similarly, it did not influence CO nearly as much as the overall air-excess ratio.

For hydrocarbons emissions, however, the pre-chamber air-excess ratio starts to influence HC as the global air-excess ratio is decreasing towards the richer region. The pre-chamber air-excess ratio decreased HC emissions as the $\lambda_{\text{Pre-chamber}}$ is going rich as observed in Figure 3.7.

Figure 3.7 Specific unburned hydrocarbons emissions for different global and pre-chamber air-excess ratios

The NO_x emission, however, was a function of both pre and global air-excess ratios, as shown in Figure 3.8. Rising the pre-chamber air-excess ratio appeared to be increasing the NO_x emissions until a $\lambda_{\text{Pre-chamber}}$ of one, which indicate that NO_x also forms in the pre-chamber. after about $\lambda_{\text{Pre-chamber}}$ of one it starts to have a decreasing effect; this effect is perhaps due to the peak in flame temperature at stoichiometry in the pre-chamber[18]. Going leaner in the overall air-excess ratio showed a decrease in NO_x emission since the combustion temperature is decreasing as we go leaner in λ_{global} .

Figure 3.8 Specific NOx Emissions for different global and pre-chamber air-excess ratios

3.1.5 Efficiency

3.1.5.1 The combustion efficiency

The combustion efficiency showed a strong dependence on the global air-excess ratio, increasing the global air-excess ratio will decrease the combustion efficiency, as shown in Figure 3.9. This effect is due to the increase in combustion losses in the form of unburned hydrocarbons and carbon monoxide, the leaner the global air-excess ratios. It should be noted that the leanest λ_{global} point recorded combustion efficiency is overestimated as the CO emissions were not recorded.

Figure 3.9 Combustion efficiency for different global and pre-chamber air-excess ratios

The overall relatively low combustion efficiency is perhaps due to a combination of the narrow throat feature, and the high latent heat of vaporization of Ethanol. As Ethanol is injected via the PFI, it reduces the intake air temperature due to its high latent heat of vaporization, which in turn slows down the reaction and possibly increases the emissions [23]. One point with $\lambda_{\text{Global}}=1.8$ and $\lambda_{\text{Pre-chamber}}=0.4$ was evaluated at three different temperatures to examine the effect of the heat of vaporization on combustion efficiency. It was found that increasing the temperature of the air-intake would indeed increase combustion efficiency, as shown in Figure 3.10.

Figure 3.10 Ethanol combustion efficiency as a function of air intake temperature

3.1.5.2 The thermodynamic efficiency

In Figure 3.11, it can be noticed that thermodynamic efficiency is increasing with the increase in the global air-excess ratio. This influence is a result of lowering temperatures as we increase λ_{Global} , which in turn decreases the heat losses. Moreover, the leaner mixture increases the ratio of specific heats.

The region with lowest thermodynamic efficiency in Figure 3.11 is due to increase in combustion efficiency alongside Fuel mean effective pressure (FuelMEP) as it contains the richest points in operation conditions tested in both chambers.

Figure 3.11 Thermodynamics efficiency as a function of global and pre-chamber air-excess ratios

3.1.5.3 The gross indicated efficiency

The gross indicated efficiency shown in Figure 3.12, demonstrated a dependency on both the global and pre-chamber air-excess ratios. Increasing the global excess ratio seems to decrease the gross indicated efficiency as it increases the combustion duration and hence decreases the indicated mean effective pressure. While increasing the pre-chamber air-excess ratio will decrease the gross indicated efficiency until a $\lambda_{\text{pre-chamber}}$ of 0.7, then it has a positive effect.

Figure 3.12 Gross indicated efficiency for different global and pre-chamber air-excess ratios

3.2 Ethanol vs. Methane

3.2.1 The effect of global air-excess ratio

The effect of the global lambda with a fixed $\lambda_{\text{Pre-chamber}}=0.4$ was studied with both Methane and Ethanol as the main fuel. Figure 3.13 shows the rate of heat release of Ethanol and Methane at $\lambda_{\text{Pre-chamber}}=0.4$ along with the pressure difference between the two chambers (ΔP) for several global air-excess ratios. Going richer in the overall mixture affected the heat release stage the same in both fuels, as it increased the initial heat release. Decreasing λ_{Global} showed to be increasing the peak in the main heat release, deterioration in main heat release in Methane is perhaps due to the lean limit being $\lambda_{\text{Global}}=2.4$. Variation in the global air-excess ratio did not seem to change the heat release of the pre-chamber using Ethanol in the main chamber and Methane in the pre-chamber. While increasing λ_{Global} showed to be advancing the pre-chamber

heat release using Methane in both chambers. Using Ethanol in the main chamber and Methane in the pre-chamber exhibited larger ΔP than that when using Methane in both chambers, it also experienced less variation.

Figure 3.13 Rate of heat release of Ethanol and Methane(top), and the difference in pressure between pre and main chambers for both fuels (bottom). $\lambda_{Pre-chamber}=0.4$, $T_{in}=30^{\circ}\text{C}$.

The effect of the overall air-excess ratio on the flame development angle is not that strong in the case of Methane. In contrast, the effect is almost nonexistent in the case of Ethanol, as observed in Figure 3.14. Generally, Ethanol takes less time to start combustion in the main chamber compared to Methane. Although the difference is not that large.

Figure 3.14 Flame development angle of Ethanol and Methane for different global air-excess ratios

The influence of the global air-excess ratio on CA50 is smaller in ethanol than in Methane, as observed in Figure 3.15. Meanwhile, the combustion takes a longer time as we go leaner in the global air-excess ratio for both fuels, as indicated in Figure 3.16. Nevertheless, the effect of λ_{global} is stronger on Methane than Ethanol. As shown in Figure 3.16, Methane combustion is faster than Ethanol for richer cases, but at around $\lambda_{\text{global}}=2.3$, it becomes slower than Ethanol.

Figure 3.15 combustion phasing (CA50) of Ethanol and Methane for different global air-excess ratios

Figure 3.16 combustion duration (CA10-90) of Ethanol and Methane for different global air-excess ratios

The combustion efficiency of Ethanol was drastically lower than that of Methane, and it continued to drop as the global air-excess ratio is increased. However, the decrease in combustion efficiency lowered the temperatures in the combustion chamber, which helped in gaining more thermodynamic efficiency in both fuels. The Thermodynamic efficiency follows the opposite trend as it increases with an increase in overall air-excess ratio. Meanwhile, the combustion losses are more for Ethanol, therefore the thermodynamic efficiency is higher. The combustion, Thermodynamic and gross indicated efficiencies for both fuels at different global air-excess ratios are shown in Figure 3.17, Figure 3.18 and Figure 3.19.

Figure 3.17 Combustion efficiency for both fuels at $\lambda_{\text{pre-chamber}}=0.4$

Figure 3.18 Thermodynamic efficiency for both fuels at $\lambda_{\text{pre-chamber}}=0.4$

Figure 3.19 Gross indicated efficiency for both fuels at $\lambda_{\text{pre-chamber}}=0.4$

Using Ethanol as the main chamber fuel increased the lean-limit from $\lambda_{\text{global}}=2.3$ using Methane to $\lambda_{\text{global}}=2.7$, as illustrated in Figure 3.20. This increase in the lean-limit is perhaps due to Ethanol higher flame speed in leaner mixtures [24].

Figure 3.20 The lean limit is set at CoV of IMEP_g of 5%, for both Methane and Ethanol

3.2.2 The effect of Pre-Chamber air-excess ratio

Two global air-excess ratios were selected, $\lambda_{\text{global}}=1.8$ and $\lambda_{\text{global}}=2$, to compare the effect of the Pre-chamber air-excess ratio with both fuels. Flame development angle and Combustion phasing (CA50) both are decreasing with increasing the pre-chamber air-excess ratio until $\lambda_{\text{Pre-chamber}}=0.7$, as shown in Figure 3.21, and Figure 3.22 respectively. Afterward, both the combustion phasing (CA50) angle and flame development angle start to rise. It could be noted that Ethanol starts the combustion earlier than Methane, as indicated in Figure 3.21. The difference in flame development angle between Ethane and Methane grows more significant as we increase the pre-

chamber air-excess ratio; this is more obvious in the case $\lambda_{\text{Global}}=2$. The combustion phasing (CA50) is affected by both pre-chamber and global air-excess ratios in the case of Methane, whereas it is mainly affected by the pre-chamber air-excess ratio in the case of Ethanol.

Figure 3.21 Flame development angle of Ethanol and Methane for different pre-chamber air-excess ratios

Figure 3.22 combustion phasing (CA50) of Ethanol and Methane for different pre-chamber air-excess ratios

Combustion duration, as a function of pre-chamber air-excess ratio, is shown in Figure 3.23. Ethanol is slower in both global air-excess ratios, as the global air-excess ratio is less than 2.3, as discussed above. For $\lambda_{\text{global}}=1.8$, there almost no effect of pre-chamber air-excess ratio on the combustion duration, but as we go leaner in overall air-excess ratio, the pre-chamber influence becomes evident. For the latter case, increasing the pre-chamber air-excess ratio will result in faster combustion until $\lambda_{\text{pre-chamber}}=0.4$ for Methane and $\lambda_{\text{pre-chamber}}=0.7$ for Ethanol; increasing the pre-chamber air-excess ratio further will slow down the combustion.

Figure 3.23 combustion duration (CA10-90) of Ethanol and Methane for different pre-chamber air-excess ratios

Chapter 4 Conclusions

This study investigated using Ethanol in the main chamber in a heavy-duty single-cylinder research engine equipped with Methane fueled narrow-throat pre-chamber, under different global and pre-chamber air-excess ratios, while keeping the intake conditions the same. Then it was compared to using Methane only in both chambers under the same operating conditions. In summary, the following conclusions can be drawn:

- 1- Two stages of combustion only were apparent in the Ethanol main chamber heat release rate, the initial and bulk heat release stages. The post-burning stage observed in Methane main chamber heat release was absent in Ethanol. It is possibly due to the nature of operation conditions tested being in the globally lean region.
- 2- Global air-excess ratios influenced the combustion duration for both fuels. Although Methane duration of combustion was more affected by the global air-excess ratio as it was faster than Ethanol for λ_{Global} less than 2.3. After that, Methane combustion starts to become slower. Combustion phasing (CA50) and flame development angle were dependent on pre-chamber enrichment. For both fuels, CA50 and the flame development angle were minima at $\lambda_{\text{Pre-chamber}}=0.7$.
- 3- Using Ethanol as the main chamber fuel extended the lean limit in the global air-excess ratio compared to using Methane. A lean limit of $\lambda_{\text{Global}}=2.7$ was reported in this study.
- 4- NOx emissions for Ethanol were dependent on the global air-excess ratio, similar to Methane, where the highest NOx recorded is in the richest global air-

excess ratio. For the richest λ_{Global} for Ethanol, NOx emissions were lower than that of Methane for the same point. For $\lambda_{\text{Global}} > 1.8$, Methane NOx emissions were slightly lower. As NOx emissions were low in the case of Ethanol, Pre-chamber air-excess ratio did influence NOx, $\lambda_{\text{pre-chamber}}$ did not seem to influence Methane fueled main chamber.

- 5- While still depending on the global air-excess ratio, HC and CO emissions of both fuels followed the opposite trend. HC and CO were excessive at the leanest λ_{Global} . Comparing HC and CO of Ethanol to Methane showed a significant difference between them, as Methane HC and CO emissions are much lower than Ethanol.
- 6- For Ethanol, Combustion efficiencies of less than 85% were reported significantly less than that of using methane as the main chamber fuel at the same operating conditions. It was attributed to the narrow-throat feature and the high heat of vaporization of the liquid fuel. Increasing the temperature of intake from 30°C to 70°C helped in gaining 3% in combustion efficiency. The global air-excess ratio negatively influenced both Methane and Ethanol Combustion efficiencies, where Ethanol was significantly lower than Methane. However, improvement in thermodynamic efficiency was reported for both fuels.
- 7- Thermodynamic efficiencies higher than 45% were reported when using Ethanol in the main chamber. It was shown decreasing air-fuel ratio in the pre, and main chambers showed to be reducing the thermodynamic efficiency as it increases the fuel mean effective pressure and the combustion efficiency. Likewise, a gross indicated efficiencies were between 34% and 39%, the

maximum the gross indicated efficiency were reported at the richest operation

condition tested of $\lambda_{\text{pre-chamber}}=0.4$ and $\lambda_{\text{global}}=1.8$.

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APPENDICES

Appendix A. Comparison between fuels

Figure 4.1 Specific HC for both fuels at $\lambda_{\text{pre-chamber}}=0.4$

Figure 4.3 Specific CO for both fuels at $\lambda_{\text{pre-chamber}}=0.4$

Figure 4.2 Specific NOx for both fuels at $\lambda_{\text{pre-chamber}}=0.4$

Figure 4.4 IMEP_g for both fuels at $\lambda_{\text{pre-chamber}}=0.4$

Figure 4.5 Specific CO for both fuels for two λ_{global}

Figure 4.6 Specific HC for both fuels for two λ_{global}

Figure 4.7 Specific NOx for both fuels for two λ_{global}

Figure 4.9 Combustion efficiency for both fuels for two λ_{global}

Figure 4.8 IMEPg for both fuels for two λ_{global}