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Optimization of Pre-Chamber Design and Operation for Marine Future Fuels

New Engine Concepts & Systems

Nathan Peters, Dumarey USA

Michael Bunce, Dumarey USA

ABSTRACT

To meet future emissions targets, a variety of low life-cycle carbon fuels are being considered to decarbonize the marine industry including hydrogen, methanol, and ammonia. Several factors will influence the choice of fuel for a particular vessel including the application, fuel cost, and regional availability of fueling infrastructure. The industry will likely adopt a diverse fuel mix given the uncertainty in both availability and demand of these future fuels, and the diversity of applications within the marine sector. Marine engine OEMs must therefore develop engine hardware capable of adapting to the future fuels of interest to the industry to protect for a wide variety of possible scenarios. Many modern marine engines, particularly high-speed and medium-speed four-stroke engines, are adapted to the use of CNG or LNG fuel, where combustion is initiated via pre-chambers. Understanding the degree to which this central combustion technology must be tailored to optimize its performance with a variety of fuels is critical to ensuring that this technology is a possible pathway for developing a fuel-adaptive engine platform.

The study is a combined numerical and experimental investigation of actively fueled pre-chamber operation with hydrogen, ammonia, and methanol fuels. Computational fluid dynamics (CFD) simulations are used to guide initial pre-chamber design. Hardware is then tested on-engine and the CFD model is correlated to experimental results, providing a high-fidelity numerical tool for investigating intra-pre-chamber effects. A heavy-duty off-road hydrogen engine and flexible light-duty research engine constitute the test platforms for this study. The smaller, flexible engine platform enables all fuels to be compared experimentally on a single engine architecture. Engine bore-related performance parameters, such as hydrogen end-gas knock, are captured through the additional use of the heavy-duty engine with similar, scaled pre-chamber hardware. This study will contrast the pre-chamber operational requirements with the different fuels to understand the opportunities and limits of component commonality amongst the fueling configurations considering the diverse range of fuel properties. Furthermore, the results provide a roadmap for system performance opportunities associated with pre-chamber-based combustion system optimization for each fuel. Finally, new experimentally validated pre-chamber-enabled combustion concepts are introduced which provide opportunities for further engine power and efficiency improvements over conventional spark-ignited and active pre-chamber engines using these fuels. The conclusions are directly applicable to medium and high-speed four-stroke pre-chamber engines being adapted for use with future fuels.

1 INTRODUCTION

To meet future emissions targets, a variety of low life-cycle carbon fuels are being considered to decarbonize the marine industry including hydrogen (H_2), methanol, and ammonia (NH_3). The industry is expected to adopt a diverse fuel mix given the uncertainty in both relative availability and relative demand of these future fuels, and the diversity of applications within the marine sector. The uncertainty in future fuels means engine original equipment manufacturers (OEMs) must protect for a wide variety of possible usage scenarios. Current and future engine development activities therefore require an understanding of how hardware must be adapted to accommodate each of these fuels.

Pre-chambers are one such technology that will require optimization for the range of future marine fuels currently under consideration. In use in many modern marine engines, particularly high-speed and medium-speed four-stroke engines, pre-chambers play a critical role in sustaining successful, stable combustion at lean operating conditions. Understanding the degree to which this central combustion technology must be tailored to optimize its performance with a variety of fuels is critical to ensuring that this technology is a possible pathway for developing a fuel-adaptive engine platform.

1.1 Abridged History of Pre-Chamber Combustion Systems

A pre-chamber is a chamber proportionally smaller than, and directly connected to, the combustion chamber. Its historical uses have spanned low pressure fuel delivery, spark plug protection, and use directly as an ignition system.

Its use as an ignition system was first applied to spark ignited (SI) engines through Sir Harry Ricardo's patent for the Ricardo Dolphin engine, on which he began development in 1903 [1]. The most prominent automotive production applications of the pre-chamber concept in the latter half of the 20th century were produced by Honda (the Compound Vortex Controlled Combustion, or CVCC, engine) [2], Volkswagen, and Toyota [3]. Images of the Ricardo Dolphin and Honda CVCC pre-chambers are shown in Fig. 1. These pre-chamber concepts initiate the combustion process in the pre-chamber, with the products of this combustion process then transferring to the main chamber and subsequently causing those main chamber contents to ignite.

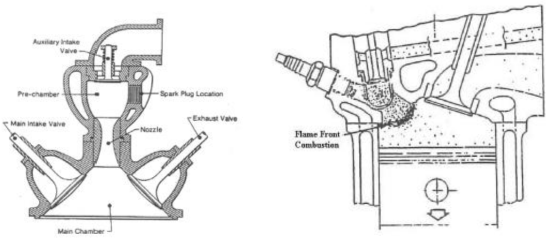

Figure 1. Two notable SI pre-chamber engine examples: Ricardo Dolphin [1] (left), and Honda CVCC [2] (right).

In the late 1970s, natural gas became a prevalent fuel in the large bore power generation sector. Many large bore stationary power engine manufacturers began adding natural gas variants to their existing diesel engine product line. Pre-chamber technology, long familiar to heavy-duty diesel engine manufacturers and researchers, and in parallel having been established as a lean combustion-enabler for SI engines, was therefore developed for this new class of large bore natural gas-fueled engines. In 1991, Caterpillar developed a natural gas-fueled variant of its 3600 diesel engine series. This variant incorporated a pre-chamber combustor to ignite lean natural gas mixtures. Other large bore engine manufacturers such as Wartsila, Waukesha, and Jenbacher also commercialized pre-chamber lean burn natural gas engines during this period.

Many of the pre-chamber applications that were commercialized in the industry were passive pre-chamber designs. In these designs, fuel is injected conventionally into the main chamber and piston motion during the compression stroke forces a volume of this air-fuel mixture proportional to the pre-chamber volume to enter the pre-chamber. Other pre-chambers, known as active pre-chambers, contain both the spark plug and an auxiliary fuelling source, allowing for a de-coupling of the chamber air-fuel ratios.

A chemical kinetically-controlled pre-chamber-based combustion mode known as jet ignition was first researched by Nicolai Semenov in the 1950s [4], followed shortly by pioneering research by Lev Gussak. While it retains the pre-chamber combustor, jet ignition differs primarily from its antecedent by the manner in which combustion translates from the pre-chamber to the main chamber. In the case of jet ignition, a high degree of quenching of the pre-chamber combustion products occurs as the products exit through a multi-orifice nozzle. This allows the products entering the main chamber to manifest as high velocity jets of partially combusted radical species. After a short delay, these jets thermo-chemically initiate combustion in the main chamber. A comprehensive history of pre-chamber concepts is provided in [5].

1.2 Jet Ignition

Jet Ignition pre-chambers have been in development by the authors for a variety of engine applications since 2009 [6]. The active jet ignition pre-chamber used in these studies incorporates a direct fuel injector (DI) and spark plug within the pre-chamber as shown in Fig. 2. Use of a direct injector enables precise targeting and metering of auxiliary fuel within the pre-chamber. This ensures that an optimal air-fuel ratio near the spark plug can be achieved at all operating conditions.

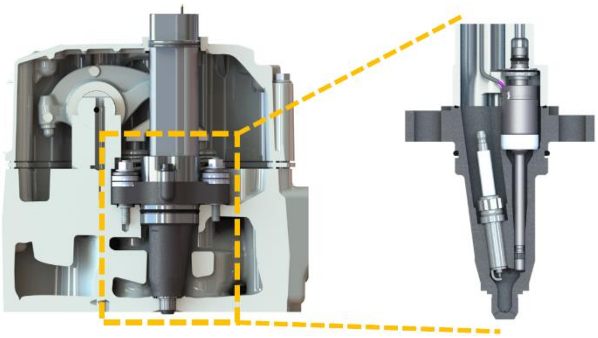

Figure 2. Rendering of jet ignition pre-chamber in a cutaway of a heavy-duty cylinder head.

Once an appropriate fuel quantity has been injected into the active jet ignition pre-chamber, it is ignited by a conventional spark plug. Combustion inside of the pre-chamber raises its pressure relative to that of the main combustion chamber. This pressure differential pushes the contents of the pre-chamber through a multi-orifice nozzle and into the main chamber at a high velocity. Unlike torch ignition, in which the orifices are large enough for a flame to travel through them, with jet ignition a high degree of quenching occurs as the flame travels through the nozzle. The jets entering the main combustion chamber contain combustion products and partially combusted radical species at a high temperature [7]. The charge within the main chamber is then thermo-chemically ignited by the jets after a short delay [8,9]. The jets create a distributed ignition event, reducing the main chamber combustion duration compared to a single point ignition source [10].

While active pre-chamber jet ignition was initially developed for use in single-fuel liquid gasoline passenger car applications by the authors, it has since been applied to several different end-use and alternative fuels applications. This includes variants for natural gas applications, funded in part by the U.S. Department of Energy (DOE) Advanced Research Projects Agency – Energy (ARPA-E)[1] and hydrogen applications funded in part by the U.S. DOE Vehicle Technologies Office[2].

1.3 Objectives

As will be demonstrated in this study, optimized pre-chamber geometry differs with fuel type. Hardware optimization through engine testing is expensive, particularly for heavy-duty engines due to the prototype part and fuel costs associated with large engines. Additionally, considering the number of unique pre-chamber geometries that could be considered, an analysis-led design approach helps to downselect to designs that are expected to perform favorably, further reducing testing costs.

This study explores both future fuels and more conventional ones, which help provide context for the unique properties of fuels such as H_2 and NH_3. Data from commonly used SI engine fuels, namely gasoline and compressed natural gas (CNG), are used in this study as a point of reference for liquid and gaseous fuels, respectively.

The aim of this work is to detail the complex relationship between pre-chamber geometry and performance, and how these can be leveraged to enable optimal system performance for each future marine fuel.

2 METHODOLOGY

The results presented in this study are a combination of computational fluid dynamics (CFD) simulation results and experimental data from multi-cylinder engine testing. The engine test results in Section 3 and Section 4 primarily come from a multi-cylinder light duty research engine, described in Section 2.2.1. This flexible engine platform has been used for understanding pre-chamber geometric and performance relationships when utilizing various fuels. Section 5 shows an example of how the pre-chamber development methodology has been employed to design pre-chamber geometries for a multi-cylinder heavy-duty pre-production engine described in Section 2.2.2. For all results and discussions, the pre-chamber and main chamber utilize the same fuel type unless otherwise stated.

2.1 Simulation

The study utilizes the CONVERGE software package [11] for open-cycle, full-geometry, three-dimensional CFD simulations. Liquid fuel injection is modeled with parcels, each representing a group

[1] DE-AR0000607

[2] DE-EE0011177

of identical droplets and used to statistically represent the entire flow field. Injection parameters (duration, pressure, flow rate) are based on engine test data and injector characterization provided by the injector supplier. Gaseous fuel injection is modeled by simulating gas flow through an orifice boundary featuring a trapezoidal flow rate shape profile. To accurately capture fuel diffusing and mixing in the chamber, a fixed embedded cone downstream to the injector is employed. The spark event is simulated using the Eulerian energy deposition model integrated into CONVERGE. This model releases energy from the spark into the fluid within a sphere situated between the electrodes, with the energy transfer modeled as an L-type distribution of the total spark energy. This representation aims to mimic the breakdown and arc/glow phases of the spark discharge and has demonstrated accuracy in depicting the ignition process for conventional spark-based systems.

CONVERGE facilitates a mesh generation process and accommodates the utilization of simple orthogonal grids, locally forced embedding, and the adaptive mesh refinement (AMR) algorithm. Notably, the AMR functionality enables the generation of small grid sizes in regions with high temperature and velocity gradients, without significantly escalating the overall number of computational cells. The combined use of AMR and embedding facilitates the attainment of high mesh resolution with a maximum cell count of less than 6 million cells. A description of the primary CONVERGE sub-models and grid size used in the study can be found in Table 1.

Table 1. CONVERGE CFD model description and sub-models.

Combustion and Emissions	
Combustion Solver	SAGE
Turbulence	RNC k-e
Wall heat transfer	O'Rourke and Amsden
Grid Size	
Base grid	4.0 mm
Smallest – AMR / Fixed Embedding	0.25 / 0.0625 mm

2.2 Test Engines

Measurements are taken in a test cell environment with motoring AC dynamometer, which is used to control the engine speed. Top dead center is measured using a probe and cylinder pressure is referenced to a high-speed intake pressure transducer. Critical measurement parameters including torque measurement are calibrated prior to commencement of testing. A development engine control unit (ECU) is utilized with additional provisions in the strategy for controlling port fuel injection (PFI) and DI fuel injectors and ignition system. Both main chamber PFI and pre-chamber auxiliary DI fuel pressure are provided externally in the test cell through pressure regulators.

Main chamber fuel is varied to achieve the commanded overall normalized air-fuel ratio (λ) in a closed loop mode employing feedback from a wide-band oxygen sensor located in the exhaust. This sensor reading is verified with a calculated λ from measured exhaust emissions. Power is controlled using an electronic throttle body and turbocharger wastegate position. As the engine is enleaned, fuel is injected into the pre-chamber. The pulse width of the pre-chamber DI is increased until the coefficient of variation (COV) of indicated mean effective pressure (IMEP) reduces to an acceptable level and abnormal combustion events are avoided. Main chamber and pre-chamber fuel flow rates are measured using a MicroMotion Coriolis flow meter and Bronkhorst M13 Coriolis flow meter, respectively. Dilution is introduced through de-throttling and increasing boost pressure via ECU demand. Measured data is averaged over 30 seconds after achieving a stable engine operating condition. Indicated data is provided by pressure transducers mounted in the cylinder head.

2.2.1 Multi-Cylinder Light Duty Research Engine

The jet ignition engine used as the research platform in the present study is an in-line turbocharged 3-cylinder. It is configurable to either DI or PFI for the main chamber fueling and is used in the PFI configuration for this study. Engine specifications are listed in Table 2.

Table 2. Specifications of the multi-cylinder light duty research engine.

Parameter	Value
Configuration	In-line engine
Number of cylinders	3
Displaced volume	1.5 l
Stroke	92.4 mm
Bore	83 mm
Compression Ratio	8:1 - 15:1
Number of Valves	4
Injection	PFI main chamber, DI pre-chamber
Piston Geometry	Flat-top with valve cutouts
Cylinder Head Geometry	Pent-roof
Boost System	Variable-geometry turbocharger

The engine incorporates an identical pre-chamber assembly into each cylinder. The pre-chamber assembly houses a fuel injector, spark plug, and high-speed pressure transducer (main chamber –

Kistler 6041, pre-chamber – AVL GH14). The pre-chamber body and nozzle are separate pieces to allow for the use of interchangeable designs.

2.2.2 Multi-Cylinder Heavy-Duty Pre-Production Engine

An example of pre-chamber sizing for future fuels is shown on a diesel engine modified for H_2 fuel. Specifications for the H_2 variant of the engine are provided in Table 3. H_2 is evacuated from the crankcase in the lab to prevent H_2 concentration build-up in the crankcase. Production-intent piston and ring materials and components are not used in this study but have been evaluated separately. The boost system for the H_2 engine is a 2-stage wastegate turbocharger with a high-pressure exhaust gas recirculation loop. The base engine is relatively low swirl, and the H_2 version of the engine tested in this study does not include any additional charge motion inducements. The engine is equipped to operate with either a spark insert providing an SI configuration or an active pre-chamber configuration.

Table 3. Specifications of the multi-cylinder heavy-duty pre-production engine.

Parameter	Value
Configuration	In-line engine
Number of cylinders	4
Displaced volume	9 l
Stroke	157 mm
Bore	135 mm
Compression Ratio	11:1
Number of Valves	4
Injection	PFI main chamber, DI pre-chamber
Piston Geometry	Bowl-in-piston
Cylinder Head Geometry	Flat
Boost System	2-stage wastegate turbocharger

3 PRE-CHAMBER DESIGN CONSIDERATIONS

Despite their relatively simple geometry, there are numerous features of a pre-chamber's design that can be modified, e.g., volume, aspect ratio, number of orifices, and orifice diameter. This makes optimization of a pre-chamber for a given engine a significant undertaking. However, several decades of pre-chamber research in both light duty and heavy-duty engines have elucidated geometric relationships for maximizing the performance of pre-chamber engines using conventional fuels.

In this section, three important variables are discussed which must be considered when optimizing pre-chamber design and performance: engine geometry, engine operating envelope, and pre-chamber fueling.

3.1 Engine Geometry

Engine combustion chamber geometry is the primary determinant for many aspects of pre-chamber design. To achieve an appropriate distribution of ignition sites in the main combustion chamber, pre-chamber geometry must be selected to achieve an appropriate jet velocity. Figure 3 shows the correlation between jet penetration prior to ignition and jet velocity from a previous optical engine study [6]. The jets emerge at a velocity that is presumed to be proportional to the pressure resulting from pre-chamber combustion (ΔP), considering a pressure/velocity relationship based on Bernoulli's principle. Pre-chamber geometries must therefore be scaled with engine size to achieve appropriate values for critical pre-chamber combustion metrics, e.g. combustion duration and ΔP as defined in Fig. 4. These metrics are affected in part by selection of the pre-chamber volume and nozzle geometry and in part by fuel properties as will be discussed later.

The pre-chamber volume for pre-chamber jet ignition is between 1% and 5% of the cylinder clearance volume. This range of volumes has been well cited in literature through the work of Gussack, *et al.* [7]. The relatively small pre-chamber volume minimizes crevice volume and avoids excessive heat loss through the pre-chamber walls. Additionally, a large pre-chamber can result in jet velocities that are too high, causing jet impingement on surfaces within the main combustion chamber (e.g., liner or piston crown, depending on combustion chamber design). Alternatively, too small of a pre-chamber volume can result in a higher incidence of misfires due to quenching of the spark kernel.

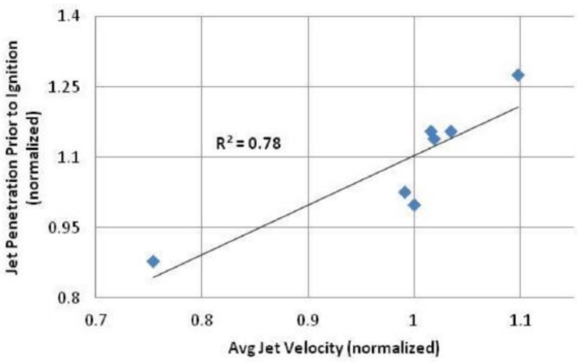

Figure 3. Jet penetration prior to ignition vs. average jet velocity from optical engine study with various pre-chamber designs [6].

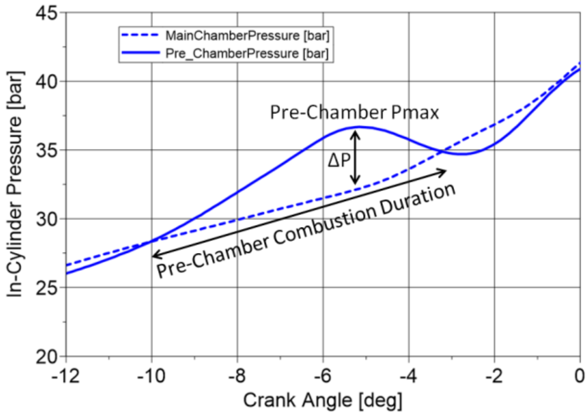

Figure 4. Pre-chamber (solid line) and main chamber (dashed line) pressure traces with pre-chamber combustion event isolated to show the definition of ΔP, pre-chamber local maximum pressure (Pmax), and pre-chamber combustion duration used in this study.

Nozzle geometric specifications for pre-chamber jet ignition are selected using patented relationships [12,13] developed through prior optical and metal engine studies [6] by the authors. These relationships are used to determine appropriate ranges for nozzle orifice area and diameter based on engine geometry. Figure 5 shows an example of orifice area and diameter ranges, sized for a light duty engine, highlighting the disadvantages of nozzle geometries falling outside of this range. This broad range of pre-chamber nozzle geometries is further downselected based on other combustion chamber geometric features and engine performance targets, as described in the subsequent section.

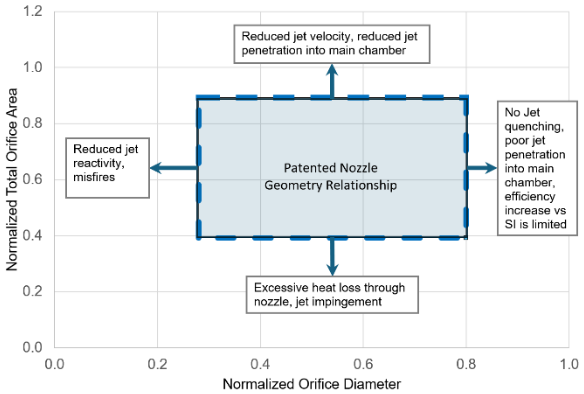

Figure 5. Example of optimal nozzle geometry for pre-chamber jet ignition. Text boxes summarize the disadvantages of geometries outside these bounds.

3.2 Operating Envelope and Performance Targets

Traditionally, jet ignition pre-chambers have had poor low load and cold start combustion stability. Pre-chamber design optimization has therefore focused on further bolstering one of the strengths of jet ignition, high load knock mitigation. Recent work by the authors has developed pre-chamber jet ignition designs and methods of operation that have significantly improved low load performance [14]. This is shown in Fig. 6 through a comparison of cold start spark retard capability for a pre-chamber optimized for low load performance, a non-optimized pre-chamber, and conventional spark ignition. While the optimized pre-chamber is able to match the spark retard capability of a conventional SI engine, the non-optimized pre-chamber has an extremely narrow window of stable operation at low load.

The pre-chamber geometry optimization for low load comes with concessions in high load performance. Figures 7 and 8 show a small reduction in brake thermal efficiency and enleanment capability, respectively, with the low-load optimized pre-chamber design. This performance trade-off makes it critical to have well defined engine operational requirements and performance targets as pre-chamber design can be tailored to enhance performance at different parts of the engine map. Typically, a more universal design is desired, requiring a relatively modest compromise between low load performance (combustion stability) and high load performance (efficiency, knock mitigation, and emissions).

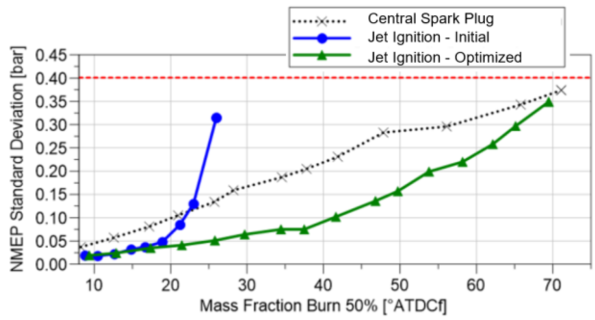

Figure 6. Steady-state cold start spark retard performance at 2 bar net indicated mean effective pressure (NMEP), comparing SI (black, dashed), initial pre-chamber design (blue), and a low-load optimized pre-chamber design (green).

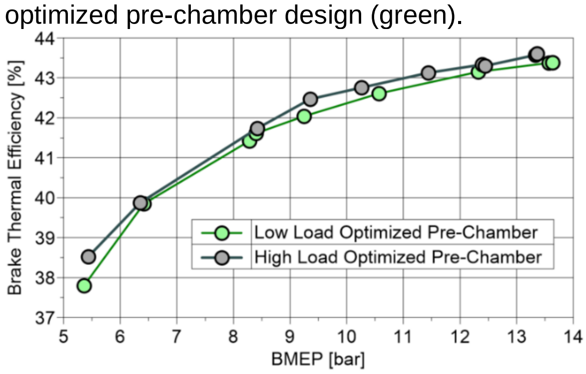

Figure 7. Brake thermal efficiency vs. brake mean effective pressure, comparing performance of low load (green) and high load (grey) optimized pre-chamber designs.

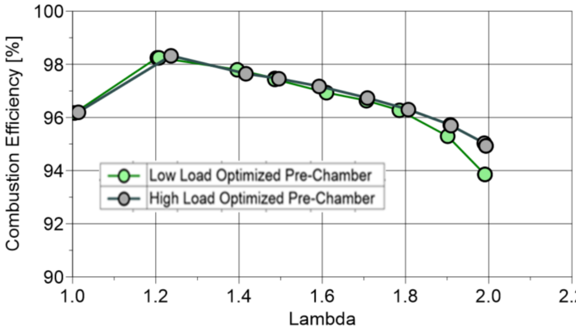

Figure 8. Combustion efficiency vs. main chamber lambda, comparing performance of low load (green) and high load (grey) optimized pre-chamber designs.

3.3 Pre-Chamber Fueling Optimization

The auxiliary fueling event has been identified in previous pre-chamber jet ignition studies as a key parameter affecting system performance. Auxiliary fuel is necessary to provide an ignitable mixture near the spark plug, thus ensuring appropriate pre-chamber combustion, jet formation, and, consequently, main chamber ignition. This “ignition fuel”, though entering from an auxiliary source, contributes to a relative increase in system fuel consumption, irrespective of any efficiency benefits provided by pre-chamber jet ignition combustion. Therefore, efficient delivery and utilization of this “ignition fuel” is important to minimize the auxiliary fuel requirement and the overall system fuel consumption.

Pre-chamber jet ignition incorporates a direct fuel injector in the pre-chamber. This enables precise control of fuel metering and spray targeting. The former is necessary for efficient, judicious use of the pre-chamber fuel in order to preserve the system efficiency, while the latter is critical for successful operation with a liquid fuel. Although liquid fuel impingement on the pre-chamber walls is unavoidable, deliberate spray targeting in the pre-chamber can minimize particulate emissions from the pre-chamber combustion event.

Another advantage of direct fuel injection for an active pre-chamber engine operating lean is the opportunity for injection late in the cycle. A fuel injection event that occurs too early can result in “over-mixing” as seen in Fig. 9, producing an overly dilute mixture near the spark plug, posing a risk of misfire. A fuel injection event late in the compression stroke is therefore desired in order to ensure an ignitable mixture near the spark plug and maximize the quantity of auxiliary injected fuel that participates in the pre-chamber combustion event.

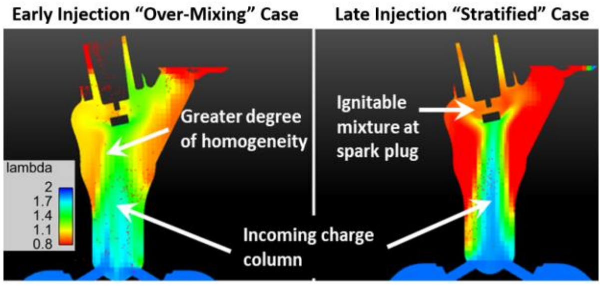

Figure 9. Mixture preparation with early (left) and late fuel injection (right) timing in the pre-chamber at time of spark with constant pre-chamber fuel quantity [15].

The impact that pre-chamber injection timing has on combustion and thermal efficiency has been investigated by the authors in a previous study [16]. Example pressure traces for various pre-chamber injection timings at similar operating conditions are shown in Fig. 10. The richer pre-chamber conditions from the late injection timing produces jets that contain a larger mass of incomplete combustion products such as partially combusted hydrocarbon species. These species, as the primary chemical trigger for main chamber ignition, are a key component of jet energy. They are discussed in greater detail in [17]. The larger mass of incomplete products during the jet expulsion stage is hypothesized to be partially responsible for the increased IMEP and efficiency seen in the late injection case.

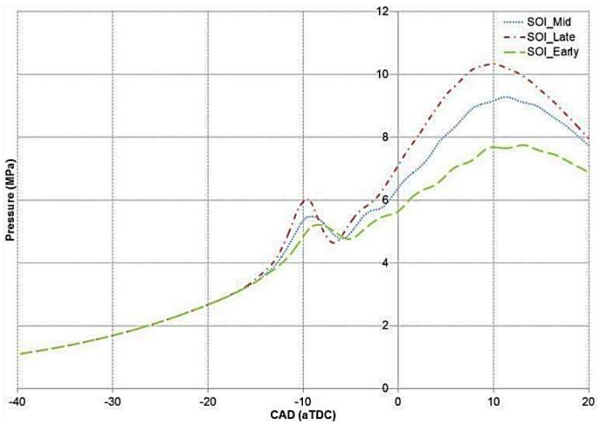

Figure 10. Cylinder pressure vs. crank angle degree (CAD), 2500 rpm, 11.7 bar IMEPg, compression ratio 14:1, λ=1.7, constant spark timing, constant fuel injection quantity, multiple start of injection (SOI) timings [16].

Table 4. Properties of selected fuels [18].

		Gasoline	Methanol	Methane	NH_3	H_2
Stoichiometric Ratio	-	14.7	6.47	17.23	6	34.3
Lower Heating Value	MJ/kg	44	20	50	18.6	120
Laminar Burning Velocity	cm/s	45	50	38	7	225
Autoignition Temperature	K	~600	738	813	923	858
Minimum Ignition Energy	mJ	0.25	0.14	0.28	>8	0.02
Mass Diffusivity in Air	cm^2/s	0.05	0.14	0.16	0.228	0.61
Quenching Distance at NTP	mm	2	1.85	2.03	7	0.64

4 FUEL-SPECIFIC PRE-CHAMBER DESIGN CONSIDERATIONS

This section explores how pre-chamber design and operation are adapted to optimize performance for a number of future marine fuels. Properties of these fuels are summarized in Table 4 and will be referenced throughout the section.

4.1 Pre-Chamber Fueling with Liquid and Gaseous Fuels

The strong sensitivity of liquid fuels to the injection timing of pre-chamber auxiliary fuel was described previously and the impact on engine performance was presented. Parametric engine studies by the authors with CNG have shown a lower sensitivity to injection timing of the pre-chamber auxiliary fuel. Figure 11 shows engine-out hydrocarbon emission data from a matrix of pre-chamber DI injection duration and end of injection timing for gasoline (top) and CNG (bottom) at the same operating condition. Hydrocarbon emissions are presented in order to highlight the impact that varying these parameters has on combustion efficiency.

While the liquid fuel in Fig. 11, gasoline, has a strong dependence on injection timing and weaker dependence on injection quantity, this relationship is reversed for the gaseous fuel. There are several reasons for the lower injection timing sensitivity with gaseous fuels. Early injection of the liquid fuel into the pre-chamber can cause some of the fuel to exit through the nozzle due to the relatively low background pressure [16]. This loss of pre-chamber auxiliary fuel lessens with gaseous injection due to the lower momentum of the injected fluid. Additionally, fuel mixing within the pre-chamber is promoted by the higher molecular diffusivity of gaseous fuels, as shown in Table 4, and a greater degree of mixing during the injection event, making charge stratification in the pre-chamber a less time-dependent process. Finally, as short-chain hydrocarbons more readily form reactive radical species, the rich pre-chamber conditions for seeding the chemically reactive jet efflux becomes less critical than with liquid, long-chain hydrocarbon fuels.

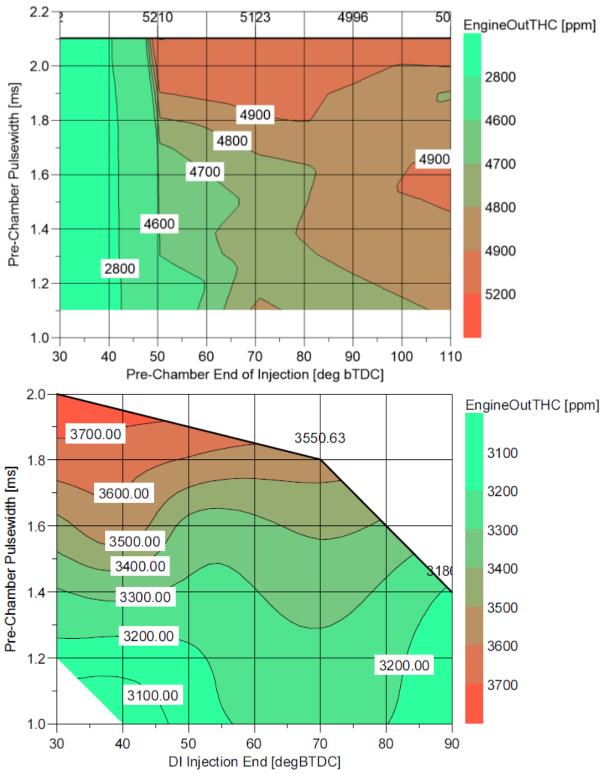

Figure 11. Contour plot of engine-out hydrocarbon emissions, highlighting the sensitivity to pre-chamber auxiliary fuel injection duration and injection timing for gasoline (top) and CNG (bottom).

There are exceptions to the general guidelines outlined above for certain gaseous fuels, most notably H_2 and NH_3 (gaseous under certain conditions). For H_2, injection timing is a critical parameter. This is due to the significant pre-ignition risk with H_2, making it more important to achieve a proper λ evolution in the pre-chamber prior to spark timing than with other fuels. For NH_3, conventional pre-chamber jet ignition operational strategies must be altered to sustain combustion, due to its unique ignition and combustion properties.

The left plot of Fig. 12 shows the combustion regime of NH_3 and gasoline during the early burn phase (ignition and flame kernel development) for a stoichiometric mixture on a Borghi-Peters' diagram. The right plot of Fig. 12 shows the flame thickness of 0.5-0.8 mm for NH_3 at 3 degrees after spark timing. For reference, the flame thickness for gasoline is typically on the order of 0.1-0.4 mm. NH_3 begins in the thickened flame regime on the left side of the plot, progressing to the corrugated flamelet region of the diagram. By contrast, gasoline is positioned in the thin reaction zone to the far right [19]. Because of NH_3's flame structure during flame kernel development, it is significantly more sensitive to local flow, thermodynamic, and fuel mixture perturbations when compared with most other fuels. As shown previously in Fig. 9, charge stratification is typically desired for pre-chamber jet ignition. However, for NH_3, a certain degree of homogeneity in terms of fuel and charge motion is desired within the pre-chamber. This can be achieved through internal pre-chamber geometry modifications to promote flow and charge uniformity, and through adjustments to the auxiliary fueling strategy.

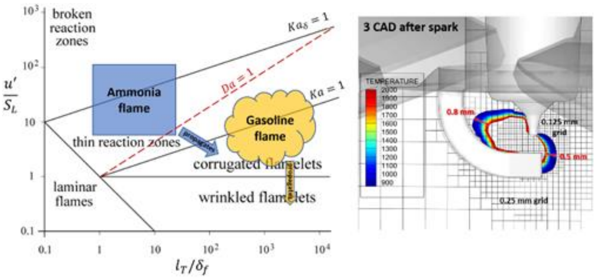

Figure 12. Left: NH_3 and gasoline combustion regime on Borghi-Peters' diagram for premixed turbulent combustion under engine relevant conditions. Right: NH_3 laminar flame thickness at 3 degrees after spark timing [19].

4.2 Pre-Chamber Volume

The engine and pre-chamber geometric relationships impacting pre-chamber combustion have been described in a previous section, in addition to the role of jet velocity as a critical metric for optimizing pre-chamber design. Here, it will be discussed how pre-chamber design is further impacted by fuel type.

Selection of the pre-chamber volume, within the bounds defined earlier of 1% to 5% of clearance volume, has a clear corollary to the amount of fuel energy contained within the pre-chamber at spark timing. For a fixed nozzle geometry and fuel type, this increase in fuel energy manifests as an increase in jet velocity, as shown in Fig. 13 for a modest 15% increase in pre-chamber volume. However, when transitioning between fuels, additional considerations for fuel density, stoichiometric ratio, and specific energy are required to maintain an appropriate pre-chamber ΔP and jet velocity.

Figure 14 shows how the usable fuel energy inside the pre-chamber varies with pre-chamber volume and pre-chamber λ for gasoline, methanol, CNG, H_2, and NH_3. The usable fuel energy calculation assumes pre-chamber conditions of 100 bar and 1000 K gas temperature. Additionally, the calculation accounts for variations in fuel properties, heat loss, and combustion efficiency. The green circle on each plot shows the pre-chamber volume requirement for 0.2 kJ of energy at stoichiometric conditions. The exception is H_2, which is marked at rich and lean pre-chamber λ's due to the pre-ignition risk of operating the pre-chamber near stoichiometric conditions.

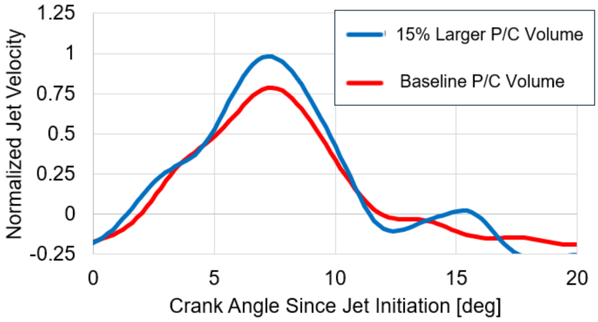

Figure 13. Normalized jet velocity vs. crank angle for two pre-chamber volumes with all other geometric parameters held constant.

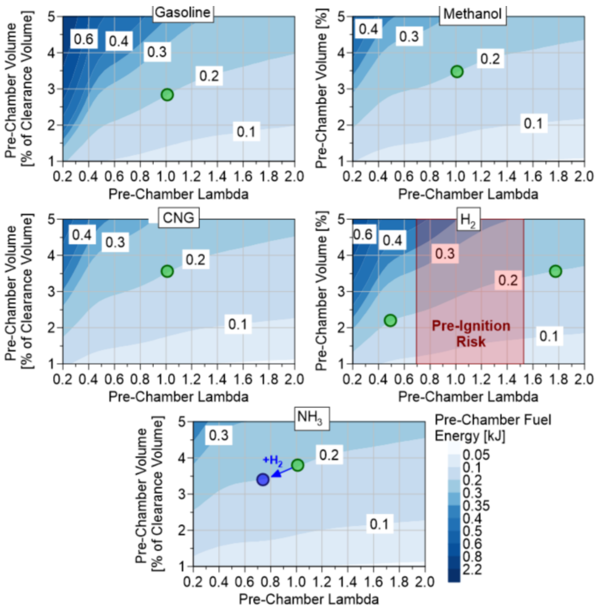

Figure 14. Contour plots of useable pre-chamber fuel energy for various fuels for a range of pre-chamber volumes and pre-chamber lambda. Green circles denote 0.2 kJ of energy at lambda 1 for all fuels except H_2, which shows both rich and lean operating points.

Compared to gasoline, the pre-chamber volume needed for equivalent fuel energy is higher for methanol due to the difference in lower heating value and higher for CNG due to the difference in density. For the H_2 pre-chamber, an interesting case is presented where sizing depends greatly on the intended mode of operation. While an H_2 pre-chamber operated lean would have equivalent fuel energy to CNG at a similar volume, one operated rich would require a significantly smaller volume.

Volume requirements for NH_3 are similar to CNG despite their vastly different lower heating values, in part due to their different stoichiometric ratios which compensates for that offset. Based only on the usable energy calculation in Fig. 14, pre-chambers designed for CNG and NH_3 should be the same volume. However, the combustion behavior after spark timing must also be accounted for in the pre-chamber design. Figure 15 shows the normalized jet velocity for CNG-fueled (blue) and NH_3-fueled (red) pre-chambers with the same geometry, at the same operating conditions. The pre-chamber jet event for the NH_3-fueled case is a longer duration and at a lower velocity than the CNG-fueled case. This is caused by the slow flame speed and lower combustion temperature of NH_3 (Table 4), leading to a pre-chamber combustion duration and ΔP, as defined in Fig. 4, that are less favorable than those of the CNG-fueled pre-chamber. The reduced pre-chamber ΔP with NH_3 can be counteracted with pre-chamber nozzle design changes that will be addressed in the subsequent section. H_2 auxiliary pre-chamber fueling for NH_3 pre-chamber engines is also often proposed as a way to mitigate this issue.

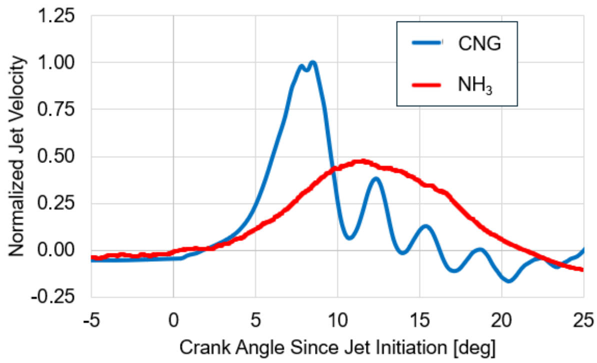

Figure 15. Normalized jet velocity vs. crank angle for CNG and NH_3 fueled pre-chambers with the same pre-chamber geometry and equivalent fuel energy.

All of the fuels included in Fig. 14 can tolerate a reasonably high degree of enleanment with the exception of NH_3. Its relatively narrow flammability limits leads to NH_3-fueled engines typically being operated with a main chamber λ closer to stoichiometry than the other fuels. As the main chamber and pre-chamber λ's are similar in the absence of auxiliary fueling, any H_2 auxiliary fueling will cause the NH_3 pre-chamber to be operated rich. This is shown by the blue circle in the bottom plot of Fig. 14. Studies on H_2-assisted jet ignition in NH_3-fueled engines have shown that the pre-chamber tolerates only a narrow band of H_2 auxiliary fueling before it becomes detrimental to engine performance [20].

4.3 Pre-Chamber Nozzle

A high degree of flame quenching is accomplished by limiting the diameters of the orifices in the nozzle. This quenching aspect of jet ignition is what differentiates it from a torch ignition system [21]. The flame quenching process is impacted by fuel type, as shown in Table 4 by the variation in quenching distances among the fuels. Gasoline, methanol, and CNG have similar quenching distances, meaning orifice diameter selection would generally be the same for these fuels.

The quenching distance for H_2 is significantly lower than the other fuels. In order to preserve the pre-chamber jet ignition quenching process, the nozzle diameter must be reduced compared to the other fuels. As shown in Fig. 16, a reduction in nozzle diameter with all other geometric aspects held constant (pre-chamber volume, number of orifices), results in an increase in jet velocity. The total orifice area must therefore be corrected to maintain an appropriate jet velocity for H_2.

In contrast to H_2, the quenching distance for NH_3 is significantly larger than the other fuels. Utilizing a nozzle with orifice diameters similar to the other fuels would result in an over-quenched flame. In this scenario, the thermal energy of the jet would not be significantly impacted since heat loss to the nozzle walls is more a function of nozzle geometry. However, the chemical energy of the over-quenched jet would be impacted through radical adsorption and recombination, reducing the number of reactive radical species to promote ignition of the main chamber charge.

Figure 17 is a pictorial representation of the general pre-chamber nozzle design changes for each fuel included in this study. Gasoline, methanol, and CNG are grouped together as their relatively similar combustion properties lead to small differences in optimized pre-chamber nozzle orifice area and orifice diameter. More significant changes to pre-chamber nozzle design are required for H_2 and NH_3 due to their more extreme combustion properties, highlighted in the rectangular boxes in Fig. 17.

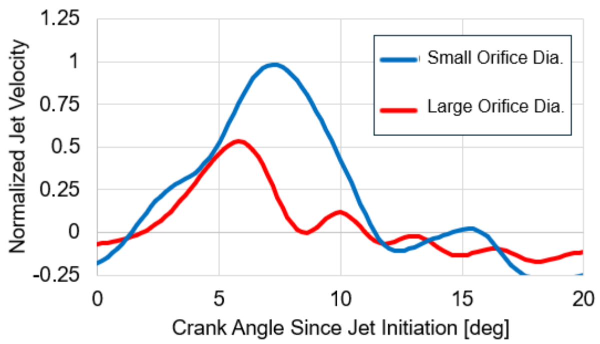

Figure 16. Normalized jet velocity vs. crank angle for two pre-chambers with different nozzle orifice diameters. All other geometric parameters are held constant.

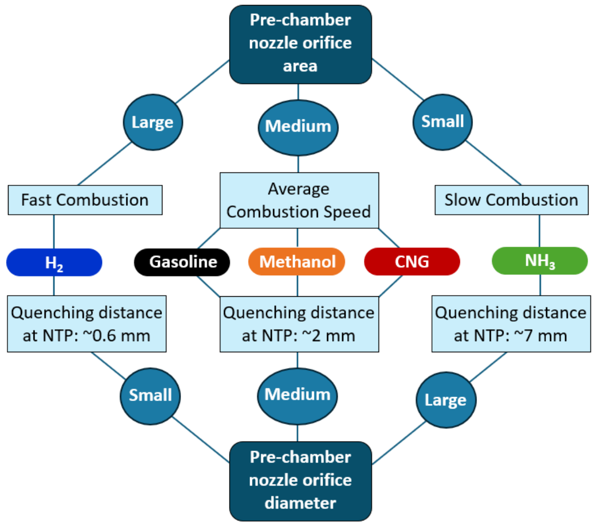

Figure 17. Summary of general pre-chamber nozzle design guidelines for fuels in this study.

5 ANALYSIS-LED DESIGN

As conveyed in the prior sections, there are many design considerations for pre-chamber jet ignition systems, further complicated by the wide range of future fuels and their unique properties (e.g., H_2 and NH_3). This makes optimization of a pre-chamber for a given engine a significant undertaking through experimental means alone. The use of 0D analysis tools as well as 1D and 3D simulations greatly reduces the cost and ambiguity of pre-chamber design optimization efforts. This approach requires a refined CFD methodology and correlated models in order to use the CFD tool in an exploratory manner. Correlated models are particularly important for H_2 due to its propensity for pre-ignition and for accurately capturing fuel slip with NH_3 due to the severe emissions implications.

The generalized pre-chamber design trends for each fuel discussed in the preceding sections are summarized in Table 5. All design changes are described with respect to gasoline, used here as the baseline. The information summarized in Table 5 constitutes a roadmap for initial pre-chamber specification, to be further optimized through analysis tools and experimental validation.

Here an example is provided of pre-chamber optimization for the H_2-fueled heavy-duty engine using simulation tools and the design methodology described in the previous sections. Figure 18 shows normalized pre-chamber sizing, using the geometric relationships described in Section 3.1. Iterations of the pre-chamber geometry within the design space are shown, starting from the CNG-spec pre-chamber design, A, and ending with the final design, C. Pre-chamber volume and nozzle geometry were adjusted in accordance with the guidelines discussed in the prior sections (A → B). The nozzle geometry was further modified to correct for the fast pre-chamber combustion and high ΔP observed with H_2 (B → C). The selection of pre-chamber geometry was guided by a correlated H_2 CFD model, used to validate the final design selection through simulations prior to hardware procurement.

Table 5. Summary of general pre-chamber design and operation changes for fuels in this study. All geometry changes are with respect to the gasoline pre-chamber.

	Gasoline	CNG	H_2	Methanol	NH_3
Main chamber λ	Lean	Lean	Lean	Lean	Near λ=1
Pre-chamber λ	Near λ=1	Near λ=1	Rich / lean	Near λ=1	Near λ=1
Pre-chamber volume	-	Increase	Decrease / increase	Increase	Increase
Orifice diameter	-	-	Decrease	-	Increase
Orifice area	-	Small Decrease	Increase	-	Decrease
Pre-chamber fuel timing	Late	Variable	Early	Late	Early
Pre-chamber fuel quantity	-	-	Increase / decrease	-	Decrease

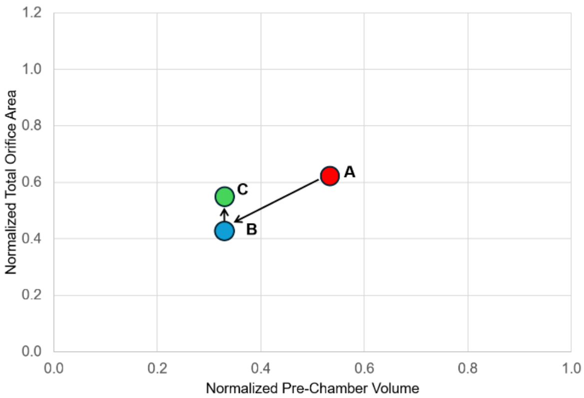

Figure 18. Geometry optimization of a pre-chamber jet ignition system for a heavy-duty H_2 engine, starting from a CNG-spec pre-chamber (A, red).

Performance of the pre-chamber was verified through testing on the multi-cylinder engine described in Section 2.2.2. Significant improvements in combustion stability, abnormal combustion mitigation, and H_2 slip reduction during enleanment were observed over the same engine operating in a SI configuration. Ultimately these improvements enabled the H_2 pre-chamber engine to increase in load to achieve the rated torque and rated power equivalent to those of the base diesel engine. The full load curves for the H_2 SI variant, H_2 pre-chamber variant, and base diesel engine at the conclusion of the test campaign are compared in Fig. 19.

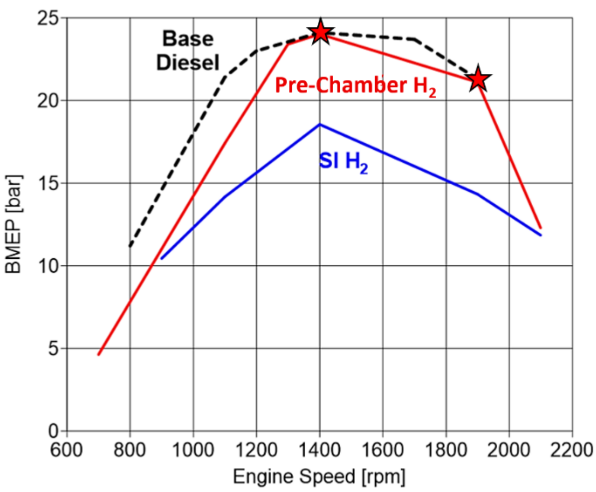

Figure 19. Full load curves for the SI and pre-chamber H_2 engines represented by blue and red lines, respectively. The full load curve for the base diesel engine is represented by the black dashed line. Red stars highlight the peak torque and peak power for the diesel and H_2 pre-chamber engines.

6 PRE-CHAMBER ENABLED ADVANCED COMBUSTION CONCEPTS

In pursuit of further improvements in engine efficiency and emissions, new combustion modes are continuously being evaluated. Here, two promising pre-chamber-based combustion systems that have been co-developed by the authors will be described, along with their associated benefits over current systems.

6.1 Pre-Chamber Main Injection

Hydrogen stands out among internal combustion engine fuels as having a number of unique properties including exceptionally fast flame speeds and wide flammability limits. These properties are often detrimental to engine performance, causing abnormal combustion events such as pre-ignition and knock. However, H_2's unique properties can also be leveraged to enable new combustion modes that would not be possible with other fuels.

Pre-Chamber Main Injection (PCMI) is an active pre-chamber technology whereby all system fuel is delivered through the pre-chamber injector. The concept is depicted in Fig. 20. Fuel introduction into the pre-chamber and through the nozzle into the main chamber is shown in Fig. 20a. Figure 20b shows the compression stroke which pushes oxygen into the pre-chamber to achieve an ignitable mixture. Figure 20c shows pre-chamber combustion and ignition of the main chamber charge which progresses as it would in a conventional active pre-chamber configuration. At spark timing, the pre-chamber charge is fuel-rich, falling outside of the ignition limits of conventional fuels, but within an acceptable range for H_2. This change in operation compared to a conventional active pre-chamber jet ignition system necessitates further optimization of the pre-chamber geometry.

PCMI simplifies the concept of active pre-chambers by utilizing one point of fuel introduction while addressing several common issues with H_2-fueled engines. Since PCMI is a form of DI fueling, backfire can be avoided which is prevalent with PFI-fueled H_2 engines. Typically, DI H_2 engines have a narrow injection window due to poor fuel mixing in the combustion chamber. Poor mixing can lead to both combustion stability issues and an increased risk of pre-ignition. With PCMI, fuel passes through the pre-chamber nozzle promoting uniform fuel distribution and thorough mixing within the main chamber. Initial tests of this concept have shown combustion stability in-line with active pre-chamber engines, low H_2 slip, and good authority over injection timing throughout the engine map.

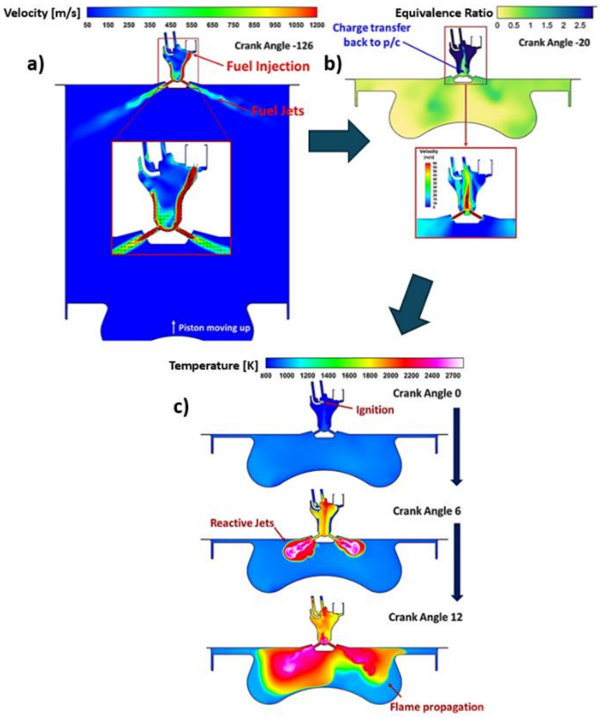

Figure 20. Depiction of the Pre-Chamber Main Injection concept showing critical parts of the cycle: (a) fuel introduction via the pre-chamber only, (b) compression, (c) pre-chamber combustion.

6.2 Pre-Chamber Enabled Mixing-Controlled Combustion

Dual fuel, utilizing a diesel pilot spray as an ignition source, is among the most prevalent combustion systems in marine engines using low carbon fuels. This extremely robust combustion mode enables efficient combustion of a wide variety of low and zero carbon containing fuels. Although diesel fuel utilization is often low at nominal medium and high load operating points, it can increase significantly in some regions of the engine map. As the industry continues to pursue dramatic reductions in its carbon footprint, single-fuel combustion systems may be desired, phasing out the use of diesel and similar heavy hydrocarbon fuels.

One such potential single-fuel solution for current dual fuel marine engines is Pre-chamber Enabled Mixing Controlled-Combustion (PC-MCC). The approach involves a pre-chamber ignition system to initiate combustion of a high-pressure direct injected fuel in the main combustion chamber. Although similar in principle to dual fuel systems, whereby hot gas generated by the pilot is used to quickly initiate combustion of the main fuel, PC-MCC replaces the diesel pilot injector with a pre-chamber utilizing auxiliary fuel that is common with the main fuel. A schematic of the concept is shown in Fig. 21.

Initial testing with methane and alcohols has shown promising results to-date [22,23]. Ongoing work with other fuels, such as H_2, will elucidate the potential of PC-MCC as an enabling technology for fuel agnostic engines.

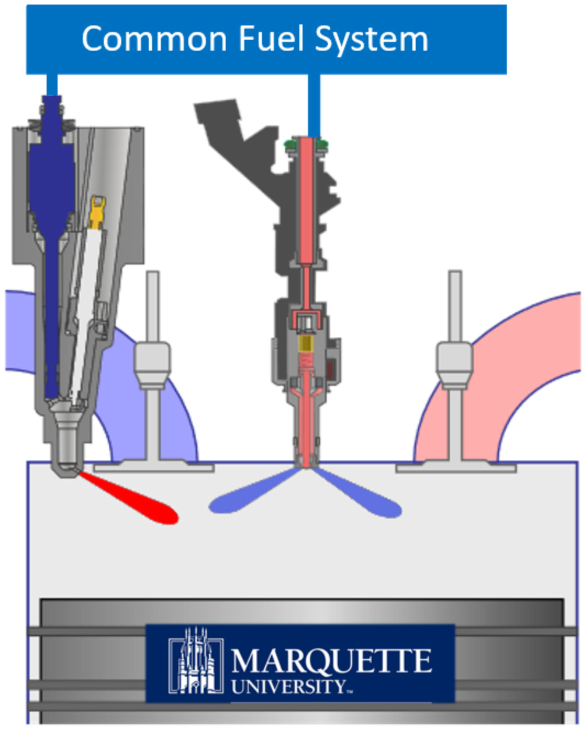

Figure 21. Schematic of the Pre-Chamber Enabled Mixing-Controlled Combustion concept showing the pre-chamber jets (red) and central high pressure injection (blue) using a common fuel source.

7 CONCLUSIONS

A significant amount of hardware optimization will be required to prepare engines for the wide range of fuels that will be available for use in the near future. This work extends to pre-chamber combustors, for which proper optimization is critical given their substantial impact on engine performance. In this paper, pre-chamber jet ignition optimization was explored both at a high level for any fuel, and through a discussion of fuel-specific pre-chamber design considerations.

General pre-chamber design starts with engine-based scaling relationships, creating a design space of possible pre-chamber geometries. Downselection of the geometry within that design space depends on a number of factors. A trade-off between high load and low load performance is required, often leading to a compromise if a high degree of transients is required. Pre-chamber optimization is also significantly impacted by fuel type as volume, nozzle geometry, and fueling strategy are all highly affected by fuel properties. It has been shown how this is particularly true of H_2 and NH_3 due to their unique fuel properties when compared with other future marine fuels. Considerations for fuel density, heating value, and combustion properties are required for appropriately sizing the pre-chamber volume. Nozzle geometry for pre-chamber jet ignition must account for quenching distance of the fuel and further modifications to maintain pre-chamber ΔP, and therefore jet velocity, within an acceptable window.

With an understanding of the critical pre-chamber geometric relationships, simulation tools can be utilized to confirm appropriate selection of hardware designs, reducing the cost of experimental validation. This process has been shown for a heavy-duty H_2 engine, achieving diesel-equivalent performance in terms of power and torque. These simulation tools also aid in the development of new combustion modes, as shown for two examples utilizing pre-chamber jet ignition systems as enabling technologies.

8 ACKNOWLEDGMENTS

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10 CONTACT

Nathan Peters
Technical Expert – Research
Dumarey USA, LLC
nathan.peters@dumarey.com