Estimates of the Air-fuel Ratio at the Time of Ignition in a Pre-chamber using a Narrow Throat Geometry

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ABSTRACT: The benefits of pre-chamber combustion (PCC), such as improved engine efficiency and reduced NO_x emissions, are primarily observed when operating at lean conditions with an active pre-chamber, where auxiliary fuel is supplied directly to the pre-chamber. Estimating the pre-chamber excess air ratio (λ) is important in the active pre-chamber concept to gain insights into the pre-chamber combustion phenomenon. Experimental investigations were performed using a narrow-throat pre-chamber at global- λ 1.6, 1.8, and 2.0. The fraction of fuel energy injected in the pre-chamber over the total fuel energy was fixed at 3%, 7%, and 13% for each global- λ . The mixture formation process inside the pre-chamber is first simulated using the 1-D simulation software GT-Power to analyze the pre-chamber λ at the ignition timing. However, the 1-D results were unable to reproduce the experimental

observations on the pre-chamber pressure buildup accurately. Upon simulating the same conditions using the 3-D CFD software CONVERGE, the pre-chamber λ estimated from the CFD model is well-correlated to the experimental data. The CFD results indicate that the amount of fuel trapped in the pre-chamber at the inlet valve closing timing is over-predicted by the 1-D simulations. A correlation between the injected and the trapped fuel in the pre-chamber is proposed by theoretical scavenging models and applied to the 1-D simulation results to improve pre-chamber λ prediction accuracy.

Introduction

Pre-chamber combustion (PCC) is a promising concept that has been shown to improve the thermal efficiency of an internal combustion (IC) engine by enabling combustion at leaner conditions than with conventional spark-ignition engines, leading to improved combustion stability, reduced knock tendency¹⁻³, and lower engine-out NO_x emissions⁴⁻⁶. The pre-chambers are suited for a wide array of alternative fuels, including hydrogen⁷⁻⁹, natural gas¹⁰⁻¹², propane¹³, alcohol fuels¹⁴, and conventional gasoline.

A pre-chamber is a small volume attached to the main combustion chamber of an IC engine in order to create an enriched mixture for enhanced reactivity. The combustion process is initiated by a spark plug installed in the pre-chamber. The pressure rise due to combustion allows the mass transfer between the two chambers through a series of nozzle holes. The combustion products from the pre-chamber are introduced into the main chamber as turbulent jets, which serve as ignition sites for combusting the main chamber air-fuel mixture.

There are two distinct versions of the PCC concept found in the literature. A *passive* pre-chamber, also known as a pre-chamber spark plug, does not have a separate fueling system and relies on air-fuel mixture supplied from the main chamber during the compression stroke. As a result, the level of mixture stratification and thus the lean operation limit extension compared to the conventional SI engines is limited¹⁵. Yu et al.¹⁶ reported that a well-designed spark ignition concept could compete with a passive pre-chamber with poor scavenging characteristics. Alternatively, an *active* pre-chamber is equipped with a separate fuel supply system to enrich the mixture in the pre-chamber, allowing a better control to extend the lean limit of the engine effectively (global- $\lambda > 2$) while reducing NO_x formations

substantially¹⁷⁻¹⁹. Provided an effective design and fabrication of the hardware, the active PCC concept is the favored option for a wider range of applications.

The history of PCC dates back to 1979 when Gussak et al.^{20,21} investigated the effect of pre-chamber fuel-enrichment and reported that a pre-chamber excess air ratio (pre-chamber λ) at 0.4-0.7 could further extend the lean operation limit compared to near-stoichiometric pre-chamber operation, by promoting the chain-branching radicals in the pre-chamber jets, which facilitates the ignition of the lean main chamber charge. Shah et al.²² conducted a similar set of experiments by varying the pre-chamber fueling rate. They found that enriching the pre-chamber air-fuel mixture could improve the combustion efficiency of the engine. Attard et al.²³ found that injecting around 2% of the total fuel via the pre-chamber yielded the highest net indicated mean effective pressure (IMEP_n) and optimal combustion stability. Silva et al.²⁴ suggested that the highest pressure buildup in the pre-chamber and the shortest ignition delay for main chamber combustion were achieved when the pre-chamber air-fuel ratio is at the stoichiometric condition at the spark timing.

On the other hand, the complexity associated with the installation of a reliable fuel supply system into the pre-chamber is not a simple task. The auxiliary fuel supply system implies a greater space requirement in the engine cylinder head for the pre-chamber assembly. The issue is not a major barrier in the case of new pre-chamber engine development, as in the case of Mahle Powertrain's turbulent jet ignition engine¹⁷ and IAV's pre-chamber engines²⁵, where dedicated pre-chamber fuel injectors are incorporated into the cylinder head design. However, reducing emissions from existing vehicles on the road is equally important, if not more, than ensuring that the new engines and vehicles can achieve regulated emissions standards. Retrofitting a new promising technology, such as the PCC, in conjunction with a low carbon fuel, such as methane, would serve as a stop-gap measure and would assist in realizing net carbon neutrality in the upcoming decades^{26,27}.

The efficiency benefit of the PCC concept would be significantly more appealing to the engine manufacturers if the pre-chamber assembly could be packaged into the existing cylinder head without substantial modifications. The pre-chamber design presented in the current study can be installed into the diesel injector pocket of a heavy-duty truck engine without cylinder head modifications. Since the pre-chamber design is adapted to the outside dimensions of the stock diesel injector, a long and narrow

throat is by necessity present in the geometry. Hence, the pre-chamber design is classified as a narrow-throat geometry, the cross-section view of which can be found in **Figure 1**.

Figure 1. Cross-section view of the pre-chamber

The main objective of the present study is to identify a correlation between the injected and the trapped fuel in the pre-chamber to accurately predict the mixture condition inside the pre-chamber at the spark timing. Toward this goal, a 1-D gas dynamic model using GT-Power is developed and compared with detailed 3-D CFD simulations in CONVERGE. Upon identifying critical factors in the model simplification, a simple correlation between the injected and trapped fuels was derived, adapted from a traditional two-stroke gas exchange model. The correlation is further validated against CFD results from additional data points at similar operating conditions but different airflow rates. The experimental results and the comparison between the 1-D and 3-D simulations are also undertaken to demonstrate the robustness of the correlation.

Pre-chamber Design

Gussak et al.²⁰ suggested that the emission formations, combustion variability, and ignition delay for the MC are minimum if the PC volume is within 2 – 3% of the engine clearance volume, and when the total nozzle area normalized by PC volume is between 0.03 – 0.04 cm²/cm³. The volume of the pre-chamber and the total nozzle opening area were selected according to the recommendation by Gussak

and the dimensions are detailed in **Table 1**. Due to the narrow pre-chamber tip diameter, it was impossible to allocate the designed nozzle area in one row. Therefore, the nozzles were arranged in two rows with a 4mm vertical offset along the principal axis of the pre-chamber. Each row of nozzles contains six nozzle holes, and the two nozzle planes had an angular offset of 30 degrees. The intention here was to maximize the spatial coverage of the main chamber with as many nozzles as possible within the design constraints.

Table 1. Geometric parameters of the pre-chamber

Parameter	Value	Unit
Pre-chamber Volume	5.07	[cc]
Pre-chamber Vol./Clearance Vol.	2.5	[%]
Number of Nozzles	12	[-]
Nozzle Diameter	1.40	[mm]
Nozzle Area/Pre-chamber Volume	0.036	[cm ⁻¹]
Nozzle Included Angle	134	[deg]
Throat Diameter	3.30	[mm]
Throat Length	22	[mm]
Throat Area/Nozzle Area	0.46	[-]

The pre-chamber assembly consisted of the pre-chamber body, the adaptor, and the spacer. The pre-chamber body, where the combustion takes place, was screwed into the adaptor, which houses the piezoelectric pressure sensor (AVL GH15DK), the spark plug (NGK ER9EH), and the check valve (The Lee Company 558 Series). A fuel supply pipe delivered the fuel to the pre-chamber via the check valve. The latter prevented the backflow of combustion gases into the fuel line during the period of pre-chamber combustion. The assembly was fitted inside the diesel injector pocket using a standard clamp arrangement on the spacer top. The pre-chamber components are presented in **Figure 2** (a), while **Figure 2** (b) illustrates how the assembly fits inside the standard diesel injector pocket.

Figure 2. Pre-chamber assembly: (a) Exploded view showing pre-chamber components, and (b) Cross-section view of the cylinder head

The fuel delivery system with a check valve was used due to its lower space requirement. By employing such an arrangement, an off-the-shelf M8 sparkplug and a M5 piezoelectric pressure sensor can be installed. OEMs such as Mahle and IAV employs electronically controlled fuel injectors in the PC assembly. The latter may operate with higher accuracy in the injector opening duration and injection timing. The check valve arrangement is rather passive since the injection relies on having a pressure difference across the check valve. However, since early cycle injection timing during the intake stroke is adopted for PC fueling, the performance difference between an electronically controlled injector and the check valve is expected to be minimal for a similar injection timing.

Experimental Engine Setup

The experimental engine was based on a Volvo D13 heavy-duty diesel engine modified for the active PCC experiments. The engine was configured to operate as a single-cylinder engine. For the main chamber fuel injection, an in-house fabricated intake manifold with pockets for port fuel injectors was fitted to the engine. The geometric compression ratio of the engine was reduced to 11.5 by changing

the standard piston for a modified one. The piston had an ω -shape with a diesel re-entrant bowl design. The specifications of the engine are listed in **Table 2**.

Table 2. Engine specifications

Engine Model	Volvo D13C500	
Control System	NI LabVIEW	
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]
Comp. Ratio	11.5	[-]
Displacement	2.1	[L]
Intake Valve Timing	Open Close	-360 CAD aTDC -170 CAD aTDC
Exhaust Valve Timing	Open Close	145 CAD aTDC -360 CAD aTDC

Control and Data-acquisition System

The hardware of the engine control and data-acquisition system was built using a CompactRIO from National Instruments. The CompactRIO was fitted with multiple C-Series modules, each performing a dedicated function such as reading thermocouple measurements or providing control inputs to the engine. The CompactRIO communicated with a host PC where the user read the sensor readings and provided control actions. The software of the control and data-acquisition system was written in NI LabVIEW. The exhaust emissions were measured by an exhaust gas analyzer (HORIBA MEXA-1700).

During the experiments, the engine speed was kept constant by a motor which also serves as a dynamometer when the engine generated positive work. A crank encoder with a 0.2 CAD resolution synchronized the engine rotation with the engine control system. An AVL GH15DK pressure sensor was installed in the engine cylinder head to measure the main chamber pressure. The engine sub-systems, such as the coolant water, air intake, and exhaust, were equipped with multiple pressure sensors and thermocouples to monitor and record the conditions. The inlet and exhaust lines were also equipped

with piezoresistive pressure sensors (AVL LPD12A) to measure the pressure profiles with 0.2 CAD resolution.

Fuel Injection System

The schematic diagram of the experimental setup can be observed in **Figure 3**. The gaseous fuel line, after the main gas admission valve, was divided into two parts: one for the main chamber and another for the pre-chamber. Both fuel lines were fitted with damping vessels to reduce pressure fluctuations. After the damping vessel, a Brooks SLA5853 mass flow meter was installed for the main chamber fuel line. The mass flow meter outlet was fed to two Bosch NGI-2 gaseous fuel injectors installed onto the engine air intake. Two injectors were necessary due to the low flow rate of the injector.

A Brooks mass flow controller SLA8580 was installed in the pre-chamber fuel line after the damping vessel. A Clean Air Power SP-021 solenoid valve was installed downstream of the mass flow controller to determine the injection timing. A piezoresistive pressure sensor (AVL LPD12A) was installed immediately downstream of the solenoid valve to measure the fuel line pressure with 0.2 CAD resolution. The outlet of the solenoid valve was connected to the fuel channel inside the pre-chamber adaptor. The length between the solenoid valve and the fuel channel is around 800 mm.

In contrast to conventional injectors with accurate control of the start and end of injection, the mass flow controller for the pre-chamber had a variable valve opening depending on the flow rate set-point. Therefore, the pressure level in the pre-chamber fuel line varied for different fuel flow rates, and the fuel continued to flow into the pre-chamber as long as a positive pressure difference was maintained between the fuel line and the pre-chamber. Accordingly, the start and end of pre-chamber injection could not be actively controlled.

Figure 3. Schematic diagram of the experimental setup: 1. Engine, 2 Intake manifold, 3. Port fuel injectors, 4. Pre-chamber, 5. Dynamometer, 6. Backpressure valve, 7. EGR Cooler, 8. EGR valve, 9. Exhaust gas analyzer, 10. Lab air valve, 11. Airflow meter, 12. Air heater, 13. Main gas admission valve, 14. Mass flow meter, 15. Mass flow controller, 16. Pre-chamber solenoid valve

Experimental Conditions and Methodology

Three global- λ (1.6, 1.8, and 2.0) conditions were investigated with three levels of pre-chamber FuelMEP ratio (PCFR) at 3%, 7%, and 13%, apart from the PCFR 3% case at global- λ 2.0, where the engine misfired. The data points were stable conditions with the coefficient of variation of gross IMEP (CoV of IMEPg < 3%). The PCFR represents the fraction of fuel energy injected into the pre-chamber over the total fuel energy supplied to the engine. The expressions for the FuelMEP and PCFR are provided below.

$$FuelMEP_{PC,MC} = \frac{\text{mass of fuel}_{PC,MC} \times LHV \text{ of Fuel}}{\text{Engine Displacement Volume}}$$

$$PCFR = \frac{FuelMEP_{PC}}{FuelMEP_{PC} + FuelMEP_{MC}} \times 100 \%$$

The initial and boundary conditions for the 1-D GT-Power and 3-D CONVERGE models were derived from the experimental data. The mixture formation process in the pre-chamber was studied using the two simulation models. From the analysis, a correlation between the injected and the trapped

fuel in the pre-chamber was identified employing a simplified gas exchange model adapted from a two-stroke gas exchange model. The application of the correlation was further proven by simulating two additional experimental data points at 1.7 bar and 1.9 bar intake pressure, respectively. These additional data points were with 13% PCFR and at the global- λ of 2.0. The experimental data points can be summarized in **Table 3**.

Table 3. Data matrix for parametric conditions under study.

	Intake Pressure = 1.5 bar			Intake Pressure = 1.7 bar	Intake Pressure = 1.9 bar
	Global- λ = 1.6	Global- λ = 1.8	Global- λ = 2.0	Global- λ = 2.0	Global- λ = 2.0
PCFR 3%	o	o	x		
PCFR 7%	o	o	o		
PCFR 13%	o	o	o	Δ	Δ
o - Stable operating points (CoV of IMEPg < 3%) x - Unstable operating point (CoV of IMEPg > 3%) Δ - Additional stable operating points at different airflow rates (CoV of IMEPg < 3%)					

Throughout the experiments, the engine speed was fixed at 1200 rpm. A pressure difference of 0.2 ± 0.05 bar was maintained between the intake and the exhaust pressure. The reason was to mimic real engine operation where there existed a pressure difference between the intake and exhaust due to the operation of a turbocharger. Both the pre-chamber and the main chamber were fueled with methane of 99.5% purity. Fuel was injected at -360 CAD aTDC (crank angle degree after the firing top-dead-center) for both the pre-chamber and main chamber fuel injections. The spark timing was fixed at -15 CAD. The experimental parameters are presented in **Table 4**.

Table 4. Engine operating conditions

	Units	Values
Engine Speed	RPM	1200
Coolant Temperature	°C	83 ± 5
Lubricating Oil Temperature	°C	85 ± 5
Intake Temperature	°C	30 ± 5

Intake Pressure	bar	1.5 ± 0.05
Exhaust ΔP	bar	0.2 ± 0.05
Spark Timing	CAD aTDC	-15
PC/MC Fuel Start of Injection	CAD aTDC	-360
PC/MC Fuel	CH ₄ (99.5% by volume)	

1-D GT-Power Modeling

The 1-D simulation model mimicked the single-cylinder operation of the testbed. The “PreChamber” object was configured with the detailed geometry of the pre-chamber cavity shown in **Figure 1**. Other input parameters directly measured include the intake and exhaust valve lift profiles, instantaneous intake, exhaust and pre-chamber fuel lines pressure traces from the experiments averaged over 500 cycles. The intake and exhaust valve discharge coefficients and the check valve flow rates were provided by the manufacturers.

The pre-chamber template employed the SI turbulence flame combustion model, while the main chamber combustion process was simulated using the jet ignition combustion model following Wenig et al.^{28,29}. Both the pre-chamber and main chamber templates were simulated in the “heat release based calibration (HRBC)” mode, where the apparent heat release rates were calculated based on the measured pressure traces of the two chambers. The apparent heat release rates were utilized in the prediction runs of the pre-chamber and main chamber combustion processes. To ensure that the trapped mass in the main chamber was matched with the experiments, the pumping loop of the pressure vs. volume diagram from the simulations and the experiments were checked for individual data points, as shown in **Figure 4**.

Figure 4. Log Pressure vs. Log Volume Diagram for experimental and simulated pressure traces

The combustion and turbulent parameters in the combustion model were tuned to match both the pre-chamber and the main chamber pressure traces to the experimental data. The discharge coefficients in the forward and reverse direction of the pre-chamber nozzle were calibrated to match the pressure measured from the experiments, especially the pre-chamber pressure buildup. The discharge coefficients used in the pre-chamber nozzle object were low, typically around 0.4, as the narrow-throat pre-chamber design restricted the flow substantially³⁰. In previous simulation studies on the narrow-throat pre-chamber, Sanal et al.³¹ discovered that the six nozzles in the upper row are choked and do not discharge powerful jets. The six lower nozzles dominated the hot jets, which was later verified by optical engine experiments^{32–35}.

Heat Release Analysis

From the measured pre-chamber and the main chamber pressure traces, the heat release analysis was conducted, following the procedure proposed by Duong et al.³⁶. The analysis is based on the first law of thermodynamics. The methodology is explained in detail in the previous work of Hlaing et al.^{36,37}. The mass transfer between the pre-chamber and the main chamber was calculated using the isentropic flow equation⁴⁰.

$$\frac{dm}{dt} = \frac{C_d A_T P_{PC}}{\sqrt{RT_{PC}}} \left(\frac{P_{MC}}{P_{PC}} \right)^{\frac{1}{\gamma}} \left[\frac{2\gamma}{\gamma-1} \left(1 - \left(\frac{P_{MC}}{P_{PC}} \right)^{\frac{\gamma-1}{\gamma}} \right) \right]^{\frac{1}{2}} \quad (1)$$

where C_d is the discharge coefficient of the pre-chamber nozzle, A_T is the geometric nozzle opening area of the pre-chamber nozzle, R is the gas constant of the mixture, and γ is the specific heat capacity ratio.

The expression for the main chamber rate of heat release is shown below.

$$\frac{dQ_{MC}}{d\theta} = \frac{\gamma}{\gamma-1} P_{MC} \frac{dV_{MC}}{d\theta} + \frac{1}{\gamma-1} V_{MC} \frac{dP_{MC}}{d\theta} - h_{PC,MC} \frac{dm}{d\theta} \quad (2)$$

The third term refers to the heat release rate due to enthalpy transfer between the main chamber and the pre-chamber. If the flow direction is from the pre-chamber to the main chamber, the enthalpy of the pre-chamber mixture (h_{PC}) is used and vice versa.

For the pre-chamber rate of heat release, the equation is further simplified since the pre-chamber volume is fixed.

$$\frac{dQ_{PC}}{d\theta} = \frac{1}{\gamma-1} V_{PC} \frac{dP_{PC}}{d\theta} + h_{PC,MC} \frac{dm}{d\theta} \quad (3)$$

The accuracy of the heat release calculations depends on the prediction of the specific heat ratio (γ) and the enthalpy term (h), which depends on the mixture composition. From experimental data alone, it is not possible to gather the species composition. Therefore, the mixture fractions estimated by the GT-Power model were employed in the heat release calculations.

3-D CONVERGE Simulations

The full 3-D computational fluid dynamics (CFD) simulations were conducted using CONVERGE™ in order to compare with the 1-D flow predictions. For an objective comparison, the 3-D simulations employed the initial and boundary conditions extracted from the 1-D GT-Power. These were imposed at the exhaust valve opening (EVO) of the previous cycle, and the CFD simulations were run for a full cycle until the exhaust valve opened in the next cycle.

Figure 5. Computational Fluid Domain in CONVERGE

The fluid domain used for this study is shown in **Figure 5**, with the pre-chamber in the middle of the main combustion chamber. The pre-chamber fuel inlet was modeled as a circumference with the same diameter as the fueling channel. A base grid size of 4 mm was initially adopted. Both uniform grid and adaptive mesh refinement (AMR) were used to combine the accuracy of the solution and reasonable run time. The pre-chamber, more sensitive, was kept at 0.5 mm base grid size all the time, while the nozzles were kept at 0.25 mm mesh size. During the combustion event, this value was halved, with the pre-chamber grid having 0.25 mm and the nozzles region 0.125 mm. Three levels of refinement were adopted for the main chamber based on velocity and temperature, achieving a minimum grid size of 0.5 mm.

The coupling between pressure and velocity was done by the pressure-implicit with the splitting of operator (PISO) algorithm, with an absolute tolerance of 1/1000. The unsteady Reynolds-averaged Navier-Stokes (URANS) model with the RNG k- ϵ turbulence closure was used. The O'Rourke and Amsden model was adopted for the wall heat transfer, with the Von Karman constant of 0.42. The Redlich-Kwong equation of state was used, along with real gas properties as a function of temperature with mixture-averaged diffusion coefficients, turbulent Prandtl number of 0.9, and turbulent Schmidt number of 0.78. For run time optimization, a variable time step algorithm was utilized, with time steps ranging from 1e-8 to 1e-6 seconds (corresponding to 0.00072 CAD and 0.072 CAD at the engine speed of 1200 rpm), according to the Courant-Friedrichs-Lewy (CFL) stability conditions for convection, diffusion, and acoustics at 1.0, 2.0, and 50.0, respectively.

Results and Discussion

The heat release rates in the pre-chamber and the main chamber, calculated from the experimentally measured pressure traces, along with the pressure difference between the two chambers, are shown in **Figure 6**. The pre-chamber heat release rate is multiplied five times for illustration purposes. The profile of the pre-chamber rate of heat release is in Gaussian shape, indicating that the pre-chamber combustion is mainly in the flame propagation mode. For all the global- λ cases, the pressure buildup in the pre-chamber decreases with increasing PCFR. The phasing of pre-chamber heat release also becomes retarded as the PCFR increases, causing corresponding retardation of the main chamber heat.

Figure 6. Rate of heat release calculated from experimentally measured pressure traces for the main chamber and pre-chamber, together with pressure buildup in the pre-chamber: (a) Global- λ 1.6, (b) Global- λ 1.8, and (c) Global- λ 2.0

Note that the PCFR 3% and 7% cases of global- λ 1.8 (**Figure 6** (b)) show nearly identical pre-chamber pressure buildup. According to the GT-Power estimation, represented by solid lines in **Figure 7** (a), the pre-chamber λ for the PCFR 3% and 7% cases is around 0.88 and 0.67, respectively. The laminar flame speed calculated in Chemkin at the conditions relevant to the spark timing (temperature = 740 K and pressure = 28.5 bar) is shown in **Figure 7** (b). The chemical mechanism employed is a reduced chemical mechanism developed by Li et al. ⁴¹. The mean pre-chamber pressure from the

experiments and temperature of the unburned gas inside the pre-chamber up to the spark timing for the three global- λ conditions estimate from 1-D GT-Power are presented in **Figure 7** (c) and (d).

Figure 7. Pre-chamber conditions at the ignition timing: (a) Pre-chamber λ predicted by 1-D GT-Power and 3-D CONVERGE simulations, (b) Calculated flame speed at conditions relevant to pre-chamber temperature and pressure at the spark timing from Chemkin simulations, (c) Mean pre-chamber pressure from experiments, and (d) Pre-chamber unburned gas temperature estimated by 1-D GT-Power at the intake pressure of 1.5 bar for Global- λ 1.6, 1.8, and 2.0 conditions

Figure 8 (a) depicts the fuel injection process for the three PCFR cases at global- λ 1.8. Although the solenoid valve in the pre-chamber fuel line is opened at -360 CAD aTDC, due to the considerable length of the fuel line, the pre-chamber fuel line pressure rises only at -340 CAD aTDC for all the cases. As

the PCFR is increased, the fuel line pressure increases while prolonging the duration of fuel flow to the pre-chamber. The pre-chamber pressure is hardly disturbed by the increase in pre-chamber fuel line pressure. The turbulent kinetic energy (TKE) of the pre-chamber, despite the differences in pre-chamber fuel pressure, is almost identical for different PCFR, as shown in **Figure 8** (b). Given that the pre-chamber combustion is mainly in the flame propagation mode, the PC pressure buildup is mainly affected by the pre-chamber λ and the turbulence levels in the pre-chamber at the spark timing. Since turbulent kinetic energy (TKE) levels are almost identical between the three PCFR cases, the difference in PC pressure buildup can be attributed to the difference in pre-chamber λ .

Figure 8. Pre-chamber fuel injection process: (a) Pre-chamber and fuel line pressure and (b) Turbulent kinetic energy for PCFR 3%, 7%, and 13% cases at global- λ 1.8

If the pre-chamber λ estimated by the 1-D GT-Power is assumed to be correct, the laminar flame speeds at PCFR 3% and 7% for the global- λ 1.8 differ by approximately 50%. This would not properly explain the nearly identical pre-chamber pressure rise for the two cases. On the other hand, the pre-chamber λ predictions from the 3-D CFD model are 1.08 and 0.87, respectively, which according to **Figure 7** (b), would yield a similar laminar flame speed. This observation is consistent with the measured pressure buildup in the pre-chamber, confirming that the pre-chamber mixing process is more accurately described by the 3-D CFD simulation.

In **Figure 9** (a), the pre-chamber λ predictions from the CFD model are correlated with the pre-chamber combustion duration (the crank angle duration between 10-90% heat released in the pre-chamber), and the observed maximum burn rate in the pre-chamber, calculated from experimental pre-chamber pressure traces. The pre-chamber λ closest to stoichiometry shows the shortest combustion duration and the higher peak burn rate in the pre-chamber. In contrast, richer pre-chamber λ shows longer pre-chamber combustion durations and lower peak burn rates. Around the stoichiometry, the flame speed is less sensitive to changes of λ . The slightly improved burn rate at pre-chamber λ of 1.1 may be attributed to the small variation in the experimental conditions.

Figure 9 (b) shows that the coefficient of variation of maximum pressure during pre-chamber combustion is affected by the pre-chamber λ . As shown in **Figure 9** (c), the pre-chamber pressure buildup becomes less stable with increasing PCFR. The phasing of pre-chamber combustion shows high cycle-to-cycle variations when the pre-chamber is rich, compared to the near-stoichiometric pre-chamber fueling.

Figure 9. Pre-chamber combustion characteristics: (a) Correlation between predicted pre-chamber λ from CFD and pre-chamber combustion duration (10-90% of heat release in pre-chamber) and maximum burn rate of pre-chamber, (b) Coefficient of variation of maximum pressure during pre-chamber combustion calculated from experimental pre-chamber pressure, and Pre-chamber pressure buildup for 100 cycles at global- λ 1.8: (c) PCFR 3%, (d) PCFR 7%, and (e) PCFR 13%

Mixture Formation Process in 1-D and 3-D Simulations

Given the importance of the pre-chamber λ , the engineering goal is to develop a semi-empirical correlation to accurately predict the pre-chamber λ over a wide range of operating conditions. To do this, a detailed analysis is conducted to identify the key physical processes that lead to the discrepancies

between the 1-D and 3-D model results. First, the mixture formation process into the pre-chamber is investigated.

Figure 10 compares the pre-chamber λ and trapped fuel predicted by the 1-D and 3-D simulations at the global- λ of 1.8. Note that the pre-chamber cannot maintain all the mass injected into it – some of the fuel leaks into the main chamber during the intake stroke, which is captured by both modeling approaches. The 1-D model consistently over-predicts the fuel retained in the pre-chamber. Hence, the pre-chamber λ predicted by the 1-D model is always lower than the 3-D prediction.

Figure 10. Pre-chamber λ estimation and mixture formation analysis for global- λ 1.8 cases: (a) PCFR 3%, (b) PCFR 7%, and (c) PCFR13%

To illustrate the mixture formation process in the pre-chamber, a series of CFD images depicting the mole fraction of methane for the PCFR 7% case at global- λ 1.8 are presented in **Figure 11**. The cross-section plane passing through the centerline of the check valve is shown here. The first instant of the injected fuel escaping into the main chamber was detected at -314 CAD aTDC. A considerable portion of the injected fuel bypasses the pre-chamber and becomes diluted with the main chamber lean air-fuel mixture. The fuel gradually fills the remaining volume of the pre-chamber, and at IVC, the pre-chamber contains a significantly high fraction of methane. During the compression stroke, the pre-chamber

mixture is diluted with the lean main chamber charge, causing the pre-chamber λ to fall within the burnable limits, which can also be detected in **Figure 10**.

Figure 11. Mole fraction of methane in pre-chamber – PCFR7%, Global- λ 1.8 (Cross-section plane through the centerline of the check valve)

In **Figure 12** (a), the fuel retained by the pre-chamber up to the spark timing is depicted for the global- λ 1.8 cases. For the PCFR 3% case, the fuel content in the pre-chamber at spark timing is higher than the injected fuel due to a higher proportion of main chamber fuel ingested during the compression stroke. A progressively smaller fraction of the injected fuel remains in the pre-chamber with increasing PCFR. Note that the mass of the main chamber fuel that enters the pre-chamber due to compression is comparable for both the 1-D and 3-D simulations. The latter values are slightly higher because, in the 3-D simulations, the main chamber contains a relatively higher mass of fuel, which is the fraction of fuel discharged from the pre-chamber. On the other hand, the trapped fuel at IVC varies considerably between the 1-D and 3-D simulations, ranging from 27 to 34%.

Figure 12 (b) shows that the amount of air trapped in the pre-chamber at the spark timing and the amount of air entering the pre-chamber during the compression stroke is similar between the 1-D and 3-D simulation results with variations of less than 3%. Therefore, the disagreement in pre-chamber λ estimation between the two models can be attributed to the difference in the amount of fuel captured by the pre-chamber at IVC. Since 1-D simulation can model the gas flow between the two chambers during the compression stroke with sufficient accuracy, the reliance on CFD simulations to accurately estimate

the pre-chamber λ can be avoided if a correlation can be established between the injected and the trapped fuel in the pre-chamber before compression begins in the main chamber.

Figure 12. Analysis of mixture formation process in the pre-chamber (a) fuel, and (b) air for global- λ 1.8 condition

Correlation between the Fuel Trapped in Pre-chamber at IVC to the Injected Fuel

The pre-chamber scavenging process is analogous to that of a two-stroke engine, assuming the inflow boundary at the fuel inlet and the outlet flow at the pre-chamber nozzle. The injected fuel scavenges the pre-chamber in a manner similar to the scavenging air entering the two-stroke. Some terms are inherited from the two-stroke gas exchange process to correlate the trapped fuel in pre-chamber at IVC to the injected fuel mass. The first term is the delivery ratio (DR) which correlates the volume of the injected fuel (V_{inj}) to the volume of the pre-chamber (V_{PC}).

$$DR = \frac{V_{inj}}{V_{pc}} \quad (4)$$

The injected fuel volume is calculated at the pre-chamber gas temperature and pressure at IVC. The second is the scavenging efficiency (SE) which is denoted as a ratio between the volume of trapped fuel (V_t) at IVC to the pre-chamber volume.

$$SE = \frac{V_t}{V_{pc}} \quad (5)$$

The trapping efficiency (TE) is the ratio between the volume of trapped fuel to the injected volume.

$$TE = \frac{V_t}{V_{inj}} = \frac{SE}{DR} \quad (6)$$

In this section, only the 3-D CFD results are considered since the 1-D results disagree with experimental findings.

For two-stroke gas exchange, Hopkinson ⁴² mentioned two ideal scavenging processes: the “perfect displacement” and the “perfect mixing”. In the perfect displacement process, all the injected fuel is retained in the pre-chamber and “perfectly displaces” the existing gases. It is also assumed that there is no short-circuit of injected fuel. Benson and Bradham ⁴³ developed a two-part model, where the scavenging air perfectly displace the residuals in the first part up to a certain delivery ratio (DR_{pd}) and then perfectly mixing with the residuals in the second. Benson ⁴⁴ further extended the combined model to include short-circuiting, where the incoming air bypass the engine without partaking in the scavenging process. The trapping efficiency derivations for the aforementioned models are given in **Appendix A**.

As depicted in **Figure 11**, the injected fuel clearly short-circuits the pre-chamber before filling up the whole volume coinciding with Benson’s conceptual model. Blair ⁴⁵ illustrated Benson’s combined model with short-circuiting in his 1996 book, which can be adapted to explain the pre-chamber fueling process as shown in **Figure 13**.

Figure 13. Pre-chamber scavenging model

In this model, a fraction of the injected fuel, σ , is considered to be directly discharged out of the pre-chamber nozzle without displacing or mixing with the pre-chamber gases. The remaining effective delivery ratio conducts the pre-chamber scavenging $(1 - \sigma)DR$. The injected fuel shows perfect displacement characteristics when the delivery ratio is less than $(1 - \sigma)DR_{pd}$. Beyond this limit, the incoming fuel “perfectly” mixes with the existing gases in the pre-chamber. The scavenging and trapping efficiency equations, adapted from Benson’s combined model with short-circuiting are given in **Table 5**.

Table 5. Combined Perfect Displacement and Perfect Mixing Model with Short-circuiting for Pre-chamber Scavenging Process

Condition	Scavenging efficiency (SE)	Trapping Efficiency (TE)
$DR < (1 - \sigma)DR_{pd}$	$SE = (1 - \sigma)DR$	$TE = \frac{(1 - \sigma)DR}{DR} = 1 - \sigma$
$DR \geq (1 - \sigma)DR_{pd}$	$SE = 1 - (1 - DR_{pd})e^{(DR_{pd} - (1 - \sigma)DR)}$	$TE = \frac{1 - (1 - DR_{pd})e^{(DR_{pd} - (1 - \sigma)DR)}}{DR}$

DR – Delivery ratio

DR_{pd} – Delivery ratio until which scavenging gases show perfect displacement characteristics

σ – Fraction of scavenging gas that short-circuits the pre-chamber

A curve fit, employing Benson's trapping efficiency equation with short-circuiting, is applied to the trapping efficiency predicted by the CFD model is plotted in **Figure 14**, along with theoretical perfect displacement and perfect mixing assumptions. The trapping efficiency of the pre-chamber falls below the perfect mixing model, indicating short-circuiting of injected fuel. The curve fit estimates the perfect displacement delivery ratio (DR_{pd}) to be 0.62 and the short-circuit fraction (σ) to be 0.72. It implies that around 72% of the injected fuel volume short-circuits the pre-chamber, and 62% of the remaining 28%, i.e., around 17% of the total injected fuel, shows perfect displacement characteristics.

Figure 14. Correlation between trapping efficiency and the delivery ratio of fuel injected into the pre-chamber

Using the trapping efficiency correlation, the amount of fuel trapped in the PC at IVC can be estimated from the delivery ratio, calculated by the injected fuel using **Equation (4)**. The remaining geometric volume of the PC is assumed to be composed of air in order to simplify the calculations. From the 1-D GT-Power simulation, the amount of air and fuel supplied from the main chamber during the compression stroke can be estimated. The total air and fuel quantity inside the pre-chamber at the spark timing can then be calculated and by normalizing the actual air-fuel ratio at the spark timing with the

stoichiometric air-fuel ratio of the fuel (17.0 for methane), the pre-chamber λ can be estimated. A simple flow diagram summarizing the steps is provided in **Figure 15**.

Figure 15. Pre-chamber λ estimation process

To test the applicability of the trapping efficiency equation further, two additional experimental data points were investigated, both having PCFR 13% and global- λ 2.0 but the intake pressure 1.7 bar and 1.9 bar, respectively. The trapping efficiency for the additional data points from 3-D CFD results is shown as the green star symbol in **Figure 14**. The mass of air and fuel inducted into the pre-chamber during the compression stroke is taken from 1-D simulations since the value is similar between the two simulation results.

Table 6. Comparison between the Pre-chamber λ estimated from the correlation and the 3D-CFD results

	Unit	Intake Press. = 1.7 bar	Intake Press. = 1.9 bar
Delivery ratio	[-]	2.91	2.96
Trapping Efficiency from Correlation	[-]	0.23	0.23
Trapped Fuel @ IVC	[mg]	2.98	3.31
Fuel from MC dur. Comp. Stroke	[mg]	1.95	2.18
Air from MC dur. Comp. Stroke	[mg]	63.24	70.71
Predicted Pre-chamber λ	[-]	0.75	0.75
3-D Pre-chamber λ (CONVERGE)	[-]	0.75	0.73
1-D Pre-chamber λ (GT-Power)	[-]	0.59	0.60
Variation from 3-D Pre-chamber λ	[%]	0.95	2.67

From **Table 6**, for the intake pressure at the 1.7 bar, the pre-chamber λ predicted from Benson's equation is almost identical to the pre-chamber λ predicted from 3-D simulation, with less than 1% variation. For the intake pressure at 1.9 bar, the estimated pre-chamber λ only shows around 3% variation from the 3-D CFD prediction.

Conclusions

The pre-chamber λ values for different operating conditions are predicted from the 1-D GT-Power and the 3-D CFD simulations. The pre-chamber λ results from the CFD model are well-correlated to the measured pressure buildup in the pre-chamber. In contrast, the 1-D model predictions of pre-chamber λ fail to match the experimental observations. The difference in pre-chamber λ between the two models is mainly due to the difference in predicting the injected fuel trapped at IVC. From CFD simulations, a substantial proportion of the injected fuel is found to be leaking into the main chamber during the fuel injection window. Hence the idealized 1-D model needed to incorporate these effects.

The trapped fuel in the pre-chamber is estimated using Benson's trapping efficiency equation with short-circuiting. The coefficients required for Benson's equation are acquired by a curve fit of trapping efficiency calculated from the CFD results. The applicability of Benson's equation is tested further by additional data points, and the pre-chamber λ predicted from the equation is almost identical to the CFD results. Instead of expensive 3-D CFD simulations, pre-chamber λ can be estimated through a combination of trapped fuel at IVC estimation using Benson's equation with short-circuiting and the compression stroke modeling from 1-D simulations, as demonstrated by this study.

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ABBREVIATIONS

PC pre-chamber; MC main chamber; CAD crank angle degree; CAD aTDC crank angle degree after the combustion top-dead-center; CFD computational fluid dynamics; DR delivery ratio, IVC inlet valve closing; PPC pre-chamber combustion concept; PCC pre-chamber combustion concept; PD perfect displacement, SE scavenging efficiency, SI spark ignition; TE trapping efficiency.

REFERENCES

- (1) Toulson, E.; Schock, H. J.; Attard, W. P. A Review of Pre-Chamber Initiated Jet Ignition Combustion Systems; 2010. <https://doi.org/10.4271/2010-01-2263>.
- (2) Alvarez, C. E. C.; Couto, G. E.; Roso, V. R.; Thiriet, A. B.; Valle, R. M. A Review of Prechamber Ignition Systems as Lean Combustion Technology for SI Engines. *Appl. Therm. Eng.* **2018**, 128, 107–120. <https://doi.org/10.1016/j.applthermaleng.2017.08.118>.
- (3) Attard, W.; Blaxill, H.; Anderson, E. K.; Litke, P. Knock Limit Extension with a Gasoline Fueled Pre-Chamber Jet Igniter in a Modern Vehicle Powertrain. *SAE Int. J. Engines* **2012**, 5 (3), 2012-01–1143. <https://doi.org/10.4271/2012-01-1143>.
- (4) Attard, W.; Blaxill, H. A Single Fuel Pre-Chamber Jet Ignition Powertrain Achieving High Load, High Efficiency and Near Zero NO_x Emissions. *SAE Int. J. Engines* **2011**, 5 (3), 2011-01–2023. <https://doi.org/10.4271/2011-01-2023>.
- (5) Olsen, D. B.; Lisowski, J. M. Prechamber NO_x Formation in Low BMEP 2-Stroke Cycle Natural Gas Engines. *Appl. Therm. Eng.* **2009**, 29 (4), 687–694.

<https://doi.org/10.1016/j.applthermaleng.2008.03.049>.

- (6) Reitz, R. D.; Ogawa, H.; Payri, R.; Fansler, T.; Kokjohn, S.; Moriyoshi, Y.; Agarwal, A.; Arcoumanis, D.; Assanis, D.; Bae, C.; Boulouchos, K.; Canakci, M.; Curran, S.; Denbratt, I.; Gavaises, M.; Guenther, M.; Hasse, C.; Huang, Z.; Ishiyama, T.; Johansson, B.; Johnson, T.; Kalghatgi, G.; Koike, M.; Kong, S.; Leipertz, A.; Miles, P.; Novella, R.; Onorati, A.; Richter, M.; Shuai, S.; Siebers, D.; Su, W.; Trujillo, M.; Uchida, N.; Vaglieco, B. M.; Wagner, R.; Zhao, H. IJER Editorial: The Future of the Internal Combustion Engine. *Int. J. Engine Res.* **2020**, *21* (1), 3–10. <https://doi.org/10.1177/1468087419877990>.
- (7) Kyaw, Z. H.; Watson, H. C. Hydrogen Assisted Jet Ignition for near Elimination of NO_x and Cyclic Variability in the S.I. Engine. *Symp. Combust.* **1992**, *24* (1), 1449–1455. [https://doi.org/10.1016/S0082-0784\(06\)80169-4](https://doi.org/10.1016/S0082-0784(06)80169-4).
- (8) Toulson, E.; Watson, H. C.; Attard, W. P. The Effects of Hot and Cool EGR with Hydrogen Assisted Jet Ignition. *SAE Tech. Pap.* **2007**, No. 724. <https://doi.org/10.4271/2007-01-3627>.
- (9) Biswas, S.; Qiao, L. Ignition of Ultra-Lean Premixed H₂/Air Using Multiple Hot Turbulent Jets Generated by Pre-Chamber Combustion. *Appl. Therm. Eng.* **2018**, *132*, 102–114. <https://doi.org/10.1016/j.applthermaleng.2017.11.073>.
- (10) Gholamisheeri, M.; Wichman, I. S.; Toulson, E. A Study of the Turbulent Jet Flow Field in a Methane Fueled Turbulent Jet Ignition (TJI) System. *Combust. Flame* **2017**, *183*, 194–206. <https://doi.org/10.1016/j.combustflame.2017.05.008>.
- (11) Almansour, B.; Vasu, S.; Gupta, S. B.; Wang, Q.; Van Leeuwen, R.; Ghosh, C. Prechamber Equipped Laser Ignition for Improved Performance in Natural Gas Engines. *J. Eng. Gas Turbines Power* **2017**, *139* (10), 101501. <https://doi.org/10.1115/1.4036291>.
- (12) Biswas, S.; Tanvir, S.; Wang, H.; Qiao, L. On Ignition Mechanisms of Premixed CH₄/Air and H₂/Air Using a Hot Turbulent Jet Generated by Pre-Chamber Combustion. *Appl. Therm. Eng.* **2016**, *106* (January 2018), 925–937. <https://doi.org/10.1016/j.applthermaleng.2016.06.070>.
- (13) Toulson, E.; Watson, H. C.; Attard, W. P. Gas Assisted Jet Ignition of Ultra-Lean LPG in a Spark Ignition Engine. *SAE Tech. Pap.* **2009**. <https://doi.org/10.4271/2009-01-0506>.
- (14) Bureshaid, K.; Shimura, R.; Feng, D.; Zhao, H.; Bunce, M. Experimental Studies of the Effect

- of Ethanol Auxiliary Fueled Turbulent Jet Ignition in an Optical Engine. July 2019.
- (15) Shah, A.; Tunestal, P.; Johansson, B. Investigation of Performance and Emission Characteristics of a Heavy Duty Natural Gas Engine Operated with Pre-Chamber Spark Plug and Dilution with Excess Air and EGR. *SAE Int. J. Engines* **2012**, 5 (4), 2012-01–1980. <https://doi.org/10.4271/2012-01-1980>.
 - (16) Yu, X.; Zhang, A.; Baur, A.; Engineer, N. The Impact of Pre-Chamber Design on Part Load Efficiency and Emissions of a Miller Cycle Light Duty Gasoline Engine. **2021**, 1–16. <https://doi.org/10.4271/2021-01-0479>.Abstract.
 - (17) Attard, W.; Fraser, N.; Parsons, P.; Toulson, E. A Turbulent Jet Ignition Pre-Chamber Combustion System for Large Fuel Economy Improvements in a Modern Vehicle Powertrain. *SAE Int. J. Engines* **2010**, 3 (2), 2010-01–1457. <https://doi.org/10.4271/2010-01-1457>.
 - (18) Vedula, R. T.; Toulson, E.; Schock, H.; Gentz, G.; Stuecken, T. Lean Burn Combustion of Iso-Octane in a Rapid Compression Machine Using Dual Mode Turbulent Jet Ignition System. *SAE Int. J. Engines* **2018**, 11 (1), 95–107. <https://doi.org/10.4271/03-11-01-0007>.
 - (19) Gentz, G. R.; Gholamisheeri, M.; Toulson, E. A Study of a Turbulent Jet Ignition System Fueled with Iso-Octane: Pressure Trace Analysis and Combustion Visualization. *Appl. Energy* **2017**, 189, 385–394. <https://doi.org/10.1016/j.apenergy.2016.12.055>.
 - (20) Gussak, L. A.; Karpov, V. P.; Tikhonov, Y. V. The Application of Lag-Process in Prechamber Engines. In *SAE Technical Paper Series*; 1979; Vol. 1. <https://doi.org/10.4271/790692>.
 - (21) Gussak, L. A. The Role of Chemical Activity and Turbulence Intensity in Prechamber-Torch Organization of Combustion of a Stationary Flow of a Fuel-Air Mixture. In *SAE Technical Paper Series*; 1983; Vol. 1. <https://doi.org/10.4271/830592>.
 - (22) Shah, A.; Tunestal, P.; Johansson, B. Effect of Relative Mixture Strength on Performance of Divided Chamber “Avalanche Activated Combustion” Ignition Technique in a Heavy Duty Natural Gas Engine. *SAE Tech. Pap.* **2014**, 1. <https://doi.org/10.4271/2014-01-1327>.
 - (23) Attard, W.; Blaxill, H. A Lean Burn Gasoline Fueled Pre-Chamber Jet Ignition Combustion System Achieving High Efficiency and Low NO_x at Part Load. **2012**. <https://doi.org/10.4271/2012-01-1146>.

- (24) Silva, M.; Sanal, S.; Hlaing, P.; Cenker, E.; Johansson, B.; Im, H. G. A Computational Investigation on The Influence of Fuel Enrichment In A Pre-Chamber on The Ignition of Main Chamber Mixture; SAE International, 2021.
- (25) Sens, M.; Günther, M.; Medicke, M.; Walther, U. Developing a Spark-Ignition Engine with 45 % Efficiency. *MTZ Worldw.* **2020**, *81* (4), 46–51. <https://doi.org/10.1007/s38313-020-0194-x>.
- (26) UK Department of Transport. *The Road to Zero: Next Steps towards Cleaner Road Transport and Delivering Our Industrial Strategy*; 2018.
- (27) U.S. Environmental Protection Agency (EPA). Diesel Retrofit Technology: An Analysis of the Cost-Effectiveness of Reducing Particulate Matter Emissions from Heavy-Duty Diesel Engines through Retrofits. *Environ. Prot.* **2006**, No. March, 1–32.
- (28) Wenig, M.; Roggendorf, K. Development of a Predictive Dual-Fuel Combustion and Prechamber Model for Large Two-Stroke Engines within a Fast 0D / 1D-Simulation Environment. **2019**, 386–397.
- (29) Wenig, M.; Fogla, N.; Roggendorf, K. *Towards Predictive Dual-Fuel Combustion and Pre-Chamber Modeling for Large Two-Stroke Engines in the Scope of 0D/1D Simulation*; 2019.
- (30) Silva, M.; Sanal, S.; Hlaing, P.; Cenker, E.; Johansson, B.; Im, H. G. Effects of Geometry on Passive Pre-Chamber Combustion Characteristics. *SAE Tech. Pap.* **2020**, 2020-April (April), 1–11. <https://doi.org/10.4271/2020-01-0821>.
- (31) Sanal, S.; Silva, M.; Hlaing, P.; Cenker, E.; Johansson, B.; Im, H. G. A Numerical Study on the Ignition of Lean CH₄/Air Mixture by a Pre-Chamber-Initiated Turbulent Jet. In *SAE Technical Papers*; 2020; Vol. 2020-April, pp 1–11. <https://doi.org/10.4271/2020-01-0820>.
- (32) Tang, Q.; Sampath, R.; Echeverri Marquez, M.; Hlaing, P.; Sharma, P.; Ben Houidi, M.; Cenker, E.; Chang, J.; Magnotti, G.; Johansson, B. Simultaneous Negative PLIF and OH* Chemiluminescence Imaging of the Gas Exchange and Flame Jet from a Narrow Throat Pre-Chamber; 2020; pp 1–14. <https://doi.org/10.4271/2020-01-2080>.
- (33) Echeverri Marquez, M.; Hlaing, P.; Tang, Q.; Sampath, R.; Cenker, E.; Ben Houidi, M.; Magnotti, G.; Johansson, B. High-Speed Imaging of Main-Chamber Combustion of a Narrow Throat Pre-Chamber under Lean Conditions; 2020; pp 1–13. <https://doi.org/10.4271/2020-01->

2081.

- (34) Sampath, R.; Tang, Q.; Echeverri Marquez, M.; Sharma, P.; Hlaing, P.; Ben Houidi, M.; Cenker, E.; Chang, J.; Johansson, B.; Magnotti, G. Study on the Pre-Chamber Fueling Ratio Effect on the Main Chamber Combustion Using Simultaneous PLIF and OH* Chemiluminescence Imaging; 2020; pp 1–13. <https://doi.org/10.4271/2020-01-2024>.
- (35) Tang, Q.; Sampath, R.; Marquez, M. E.; Sharma, P.; Hlaing, P.; Houidi, M. Ben; Cenker, E.; Chang, J.; Magnotti, G.; Johansson, B. Optical Diagnostics on the Pre-Chamber Jet and Main Chamber Ignition in the Active Pre-Chamber Combustion (PCC). *Combust. Flame* **2021**, *228*, 218–235. <https://doi.org/10.1016/j.combustflame.2021.02.001>.
- (36) Duong, J.; Andersson, Ö.; Hyvönen, J.; Álden, M.; Wellander, R.; Johansson, B.; Richter, M. High Speed Combustion Imaging in a Large Bore Gas Engine: The Relationship Between Pre- and Main Chamber Heat Release. In *Volume 8A: Heat Transfer and Thermal Engineering*; ASME, 2014; p V08AT09A022. <https://doi.org/10.1115/imece2013-64286>.
- (37) Hlaing, P.; Marquez, M. E.; Singh, E.; Almatrafi, F.; Cenker, E.; Houidi, M. Ben; Johansson, B. Effect of Pre-Chamber Enrichment on Lean Burn Pre-Chamber Spark Ignition Combustion Concept with a Narrow-Throat Geometry Effect of Pre-Chamber Enrichment on Lean Burn Pre-Chamber Spark Ignition Combustion Concept with a Narrow-Throat Geometry. *SAE Tech. Pap.* **2020**, 1–21. <https://doi.org/10.4271/2020-01-0825>.
- (38) Hlaing, P.; Echeverri Marquez, M.; Burgos, P.; Cenker, E.; Ben Houidi, M.; Johansson, B. Analysis of Fuel Properties on Combustion Characteristics in a Narrow-Throat Pre-Chamber Engine. SAE International 2021.
- (39) Hlaing, P.; Echeverri Marquez, M.; Bhavani Shankar, V. S.; Cenker, E.; Ben Houidi, M.; Johansson, B. A Study of Lean Burn Pre-Chamber Concept in a Heavy Duty Engine. *SAE Tech. Pap. Ser.* **2019**, *1*, 1–13. <https://doi.org/10.4271/2019-24-0107>.
- (40) Heywood, J. B. *Internal Combustion Engine Fundamentals*; 1998.
- (41) Li, Y.; Alfazazi, A.; Mohan, B.; Alexandros Tingas, E.; Badra, J.; Im, H. G.; Mani Sarathy, S. Development of a Reduced Four-Component (Toluene/n-Heptane/Iso-Octane/Ethanol) Gasoline Surrogate Model. *Fuel* **2019**, *247* (March), 164–178.

<https://doi.org/10.1016/j.fuel.2019.03.052>.

- (42) Hopkinson, B. *The Charging of Two-Cycle Internal Combustion Engines*; . n., 1914.
- (43) Benson, R. S.; Brandham, P. T. A Method for Obtaining a Quantitative Assessment of the Influence of Charging Efficiency on Two-Stroke Engine Performance. *Int. J. Mech. Sci.* **1969**, *11* (3), 303–312. [https://doi.org/10.1016/0020-7403\(69\)90048-4](https://doi.org/10.1016/0020-7403(69)90048-4).
- (44) Benson, R. S. Internal Combustion Engines. In *Internal Combustion Engine Volume 1 and 2*; Elsevier, 1979. <https://doi.org/https://doi.org/10.1016/C2013-0-03019-8>.
- (45) Blair, G. *Design and Simulation of Two-Stroke Engines*; 1996.

Appendix A – Trapping Efficiency from Two-stroke Scavenging Models

Perfect Displacement Model

In the perfect displacement process, all the injected fuel is retained in the pre-chamber and “perfectly displaces” the existing gases. There is no short-circuit of injected fuel. If the volume of injected fuel is less than the pre-chamber volume, i.e., $DR < 1$:

then

$$V_t = V_{inj}$$

consequently,

$$SE = SR$$

and

$$TE = 1$$

If the volume of injected fuel exceeds the pre-chamber volume, i.e., $DR \geq 1$
then

$$V_t = V_{PC}$$

consequently,

$$SE = 1$$

and

$$TE = \frac{1}{DR}$$

Perfect Mixing Model

In perfect mixing scavenging, the injected fuel mixes “perfectly” with the existing gases in the pre-chamber and does not exhibit perfect displacement characteristics. Consider the scavenging process at an instance where an increment in injected fuel volume (dV_{inj}), induces an equal increment in the outflow of gases (dV_{out}) from the pre-chamber. Part of the outflow gases contains an increment of fuel

(dV_e). The volume of fuel retained in the pre-chamber after this increment in injected fuel (dV_t) is given by:

$$dV_t = dV_{inj} - dV_e$$

or

$$dV_t = dV_{inj} - dV_{out} * \frac{V_t}{V_{PC}}$$

$$dV_t = dV_{inj} - dV_{out} * SE$$

Differentiating the scavenging efficiency gives:

$$dSE = \frac{dV_t}{V_{PC}}$$

Substituting dV_t and taking into account that dV_{out} is numerically equivalent to dV_{inj} ,

$$dSE * V_{pc} = dV_{inj} - dV_{inj} * SE$$

Rearranging the equation yields,

$$\frac{dSE}{1 - SE} = \frac{dV_{inj}}{V_{pc}}$$

Differentiating the delivery ratio gives,

$$dDR = \frac{dV_{inj}}{V_{pc}}$$

Therefore,

$$\frac{dSE}{1 - SE} = dDR$$

Integrating this equation gives,

$$\log_e(1 - SE) = -DR$$

Hence, the scavenging efficiency of the perfect mixing process is:

$$SE = 1 - e^{-DR}$$

The trapping efficiency for this process is

$$TE = \frac{1 - e^{-DR}}{DR}$$

Combined Perfect Displacement and Perfect Mixing Model

The combined model suggests, until a certain delivery ratio (DR_{pd}) is reached, the pre-chamber would exhibit perfect displacement characteristics. After DR_{pd} , a perfect mixing process takes place where the incoming fuel mixes with the existing gases in the pre-chamber. The combined model can be summarized as follows:

When

$$DR < DR_{pd}$$

then

$$SE = DR$$

and

$$TE = 1$$

and if

$$DR \geq DR_{pd}$$

then,

$$SE = 1 - (1 - DR_{pd})e^{(DR_{pd}-DR)}$$

and

$$TE = \frac{1 - (1 - DR_{pd})e^{(DR_{pd}-DR)}}{DR}$$

A summary of trapping efficiency expressions from the above gas exchange models and the Benson's combined model with short-circuiting are provided in **Table A1**.

Table A1. Trapping Efficiency from Theoretical Models

Model	Trapping Efficiency (TE)		Reference
Perfect Displacement	$DR < 1$	$DR \geq 1$	Hopkinson ⁴²
	1	$\frac{1}{DR}$	
Perfect Mixing	$\frac{1 - e^{-DR}}{DR}$		Hopkinson ⁴²

Combined	$DR < DR_{pd}$	$DR \geq DR_{pd}$	Benson and Bradham ⁴³
	1	$\frac{1 - (1 - DR_{pd})e^{(DR_{pd}-DR)}}{DR}$	
Combined with Short-circuiting	$DR < DR_{pd}(1 - \sigma)$	$DR \geq DR_{pd}(1 - \sigma)$	Benson ⁴⁴
	$1 - \sigma$	$\frac{1 - (1 - DR_{pd})e^{(DR_{pd}-(1-\sigma)DR)}}{DR}$	

DR – Delivery ratio

DR_{pd} – Delivery ratio until which scavenging gases show perfect displacement characteristics

σ – Fraction of scavenging gas that short-circuits the pre-chamber

Appendix B - Sensitivity Analysis

A simple sensitivity analysis was performed for all the data points at intake pressure 1.5 bar to investigate the sensitivity of the coefficients ' DR_{pd} ' and ' σ ' on the trapping efficiency correlation. The base values of the two coefficients were increased and decreased by 10%, and the variation in pre-chamber λ was observed. The results are plotted in **Figure B1**. From the sensitivity analysis, 10% variations in ' DR_{pd} ' and ' σ ' caused less than 7% variation in the pre-chamber λ for all the data points. Therefore, the pre-chamber λ values are not very sensitive to the perturbations of the coefficients.

Figure B1. Sensitivity analysis of correlation coefficients on pre-chamber λ at intake pressure of 1.5 bar