

Experimental and Numerical Investigation of the Engine Operational Conditions' Influences on a Small Un-Scavenged Pre-Chamber's Behavior

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ABSTRACT

Despite significant benefits in terms of the ignition enhancement, the strength and timing of the turbulent flame jets subsequently issuing into the main chamber strongly depend on the pre-chamber combustion process and, thus, are sensitive to the specific engine operating conditions it experienced. This poses considerable difficulties in optimizing engine operating conditions as well as controlling engine performance. This paper investigates the influence of engine operating conditions on the pre-chamber combustion event using both experimental and numerical methods. A miniaturized piezo-electric pressure transducer was designed to be placed inside the engine cylinder head to record the pre-chamber inner volume pressure, in addition to conventional pressure indication inside the main chamber. The pressure difference between the main chamber and the pre-chamber ($\Delta p = p_{pre-chamber} - p_{main\ chamber}$) served as an indicator of the pre-chamber combustion event, through the study of the crank angle resolved Δp under different engine operating conditions. The variations include spark timing, air-fuel ratio and engine speed, as well as engine intake air temperature and boost pressure, for which a single-parameter sensitivity approach was carried out. 3D-CFD simulation in the RANS context and 0D modeling was further employed to estimate the turbulence level and thermo-physical conditions inside the pre- and main chambers in dependence of the engine operating conditions. The results indicate that the Δp peak value and its timing show the highest sensitivity to the air-fuel ratio variation, followed by the intake air temperature, while the sensitivity of Δp peak value and its timing to the pressure difference and the absolute pressure level at spark timing were observed to be considerably lower. Based on the observations, two correlations were proposed to predict the Δp peak value and its timing for which successful validation against a large number of engine operating conditions was demonstrated.

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INTRODUCTION

The demand for higher engine efficiency and lower NO_x emissions drives stationary gas engines to be operated under ever-leaner conditions. Nevertheless, lean burn technology presents several tradeoffs: the lower fuel concentration inside the combustible mixture reduces its reactivity and, as a result, the ignitability and flame propagation speed. Consequently, unburnt hydrocarbon (UHC) emissions, engine combustion stability as well as the risk of misfire are increased. To overcome these disadvantages induced by lean burn technology, ignition enhancement measures, such as pre-chamber ignition [1, 2], diesel pilot injection ignition [3, 4, 5] as well as laser

ignition [6] have been introduced. The present study focuses on an un-scavenged pre-chamber, which is chosen due to its good performance in addressing the aforementioned lean burn issues and also for its engineering simplicity [7, 8, 9]. The phrase "un-scavenged" means that there is no auxiliary fueling inside the pre-chamber. The unburnt mixture is fed into the un-scavenged pre-chamber through the compression motion of the engine piston. On the one hand, compared to the conventional open-chamber spark plug, the un-scavenged pre-chamber increases the ignition energy and the flame surface area, and shortens the flame transport distance thanks to the flame jets which penetrate the combustion chamber. On

the other hand, compared to more advanced scavenged pre-chamber or diesel pilot ignition techniques, the un-scavenged pre-chamber doesn't require an additional fuel supply system, which greatly reduces the system complexity and avoids additional emission problems (e.g. soot from diesel pilot injection ignition).

A large body of research devoted to the study of pre-chamber' performance as well as the fundamental physics involved can be found in literature. Oppenheim et al. [10] reported a study of jets of active radicals for igniting lean mixtures from a fundamental perspective. The Schlieren photography employed in his study indicates that the jet initial velocity plays a dominant role in jet's gas-dynamic properties. A new firing concept for internal combustion engines, so-called 'Auto-inflammation Pilotée par Injection de Radicaux' (APIR), was introduced by Robinet et al [11]. The extension of lean operating limit as well as a great reduction of the cycle variability were observed by using this system. A follow-up study conducted by Couet et al. [12] indicates that the APIR device leads to a better knock resistance which allows for higher compression ratio operation. Roethlisberger et al. [7, 13] studied the topic of engine operating condition and geometrical parameters' influence on the pre-chamber combustion by focusing on the final engine emissions and overall efficiency achieved. The publication claims that the pre-chamber ignition requires a significant delay of the spark timing to generate substantial gas jets. Minimum emissions (NO_x, CO and UHC) are obtained at a spark timing of about 8 °CA BTDC. In addition to the engine measurements, cold flow 3D-CFD simulations were employed to assist the investigation. The numerical results illustrate that the pre-chamber convective flow velocities are comparable to open chamber levels at the spark location, whereas the turbulence intensity was enhanced inside the pre-chamber. Kawabata et al. [14] investigated the flame jets optically inside a large bore gas engine. Fast and long penetrating jets were observed, and jet-wall impingement was evidenced by the 2D images. The author also concluded that by increasing the number of pre-chamber nozzles, the main chamber ignition and combustion can be accelerated. Attard et al. [15] investigated a so-call Turbulent Jet Ignition (TJI) system. The benefit on net indicated thermal efficiency of 42% was highlighted in a standard modern engine platform and the advantage in lean operating limit was further confirmed as well as extended by altering the flame kernel development inside the small but rich pre-chamber. The improvements on the combustion duration and the brake specific fuel consumption in comparison with a baseline spark plug ignited engine was further reported on a similar pre-chamber system over a wide range of lambda by Bunce et al. [16]. Shah et al. [17, 18] presented studies of the influence of the pre-chamber volume and the nozzle diameter on the resulting ignition characteristics. It was observed that a larger pre-chamber provides higher ignition energy, which results in shortened combustion duration while the nozzle diameter also affects the combustion duration, which may happen due to the difference in penetration of the jets issuing from the pre-chamber. The results of numerical simulation included in the investigation show that larger pre-chamber volume or smaller nozzle

area lead to longer penetration and shorten the combustion duration. Toulson et al. [19] studied a so-called gas assisted jet ignition system with several alternative fuels, namely gasoline, LPG, and hydrogen. The numerical simulation was conducted by using STAR-CD solver with CHEMKIN Kinetics and the outcome revealed that chemical properties of the rich pre-chamber jets were essential in promoting lean main chamber combustion in the jet ignition process. Gholamisheeri et al. [20, 21] investigated a turbulent jet ignition system on a rapid compression machine by means of both numerical and experimental methods. A Well Stirred Reactor (WSR) combustion model in conjunction with a detailed chemistry solver were applied for the combustion simulation, and the numerical results indicated reasonably good agreement between the predicted jet shape and penetration compared to the optical images. Experimentally, Gholamisheeri studied the effect of orifice size and mixture equivalence ratio. Both nozzle area reduction and equivalence ratio increment led to a reduction of the hot jet penetration speed. Thelen et al. [22, 23] focused on the dependence of overall combustion duration on spark location away from the pre-chamber nozzle and nozzle radius by using a similar numerical setup as Gholamisheeri. Biswas et al. [24, 25] studied the turbulent torch ignition mechanism from a more fundamental perspective with the help of simultaneous high-speed Schlieren and OH*-chemiluminescence imaging. The findings allowed for differentiation between the so-called flame ignition and jet ignition regimes according to the ignition delay in the main chamber. In addition, based on a proposed global Damköhler number and the Borghi diagram, he tried to explain the non-ignition cases by using the turbulence-flame interaction theory, and found out all non-ignition cases located at the broken reaction zone regime. Most recently, Vedula et al. [26] studied a single-cylinder dual-mode engine ignited by a scavenge pre-chamber. Very early spark timings (> 33 °CA BTDC) were reported to reduce the peak HRR and to slow down the main chamber combustion. In addition, the engine net thermal efficiency was discussed on a basis of an engine speed-load map.

Even though a large number of publications can be found concerning pre-chamber combustion, to the authors' best knowledge, very few studies focus specifically on the engine operational parameters' influence on pre-chamber performance. At spark timing, due to changes in engine operating conditions, the variations in pressure, temperature as well as turbulence quantities affect the pre-chamber combustion and later on govern the main chamber combustion process. Therefore, instead of investigating the influence of pre-chamber geometrical parameters (e.g. pre-chamber volume, nozzle sizes, nozzle orientation, etc), the present work attempts to quantitatively characterize the effect of changes in engine operational parameters' on pre-chamber combustion. To achieve this goal, both pre- and main chambers were monitored using pressure measurement under different engine operating conditions. The measurements allowed the calculation of the pressure difference Δp between the pre- and main chamber. The influence of mixture air-fuel ratio, engine speed, boost pressure, intake temperature and spark timing on the

pressure difference were analyzed and discussed individually. In addition, 0D and 3D CFD simulations were employed to precisely compute Δp at spark timing as well as to provide insights on pre-chamber internal thermochemical and turbulence conditions. Based on the engine dataset analysis, two correlations regarding the peak Δp and its timing were proposed and validated.

METHODOLOGY

In-Pre-Chamber Pressure Measurement

In order to obtain the information about the pre-chamber combustion, a miniature type Kistler sensor (type 6055C) was used. The sensor was mounted inside the cylinder head and connected to the pre-chamber volume through a small nozzle, which has a radius and a length of 1.5 mm and 3 mm respectively, as it is shown in Figure 1. Restricted by the space, no water cooling was available for this sensor. Therefore, the sensor was designed to be mounted on the intake side and close to the water jacket to minimize the influence of combustion heat. A water cooled Kistler piezoelectric pressure transducer (type 6067) was flush mounted in the main chamber for the cylinder pressure measurement. The pressure difference, Δp , which is defined in equation (1) was introduced, to account for the open system properties of the pre-chamber combustion process.

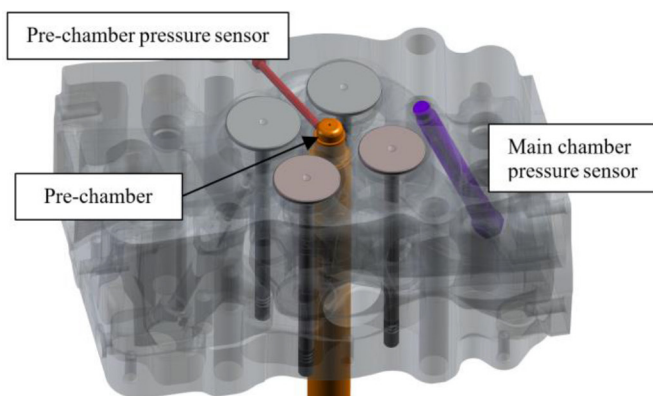


Figure 1. Designs of pre-chamber pressure sensor (red) and main chamber pressure sensor (blue) accesses in the Liebherr cylinder head.

$$\Delta p = p_{PC} - p_{MC} \quad (1)$$

p_{PC} and p_{MC} are pressure inside the pre- and main chamber respectively.

Figure 2 shows a typical plot of Δp shortly before and after the ignition together with the corresponding the pre- and main chamber pressures. The absolute Δp values have been normalized by dividing the trace with the peak Δp value. During the compression stroke and before ignition, due to the cylinder pressure increase, the existence of negative Δp and a flow from the main chamber to the pre-chamber is expected. Once the spark occurs inside the pre-chamber, the pre-chamber pressure increases as a result of the deposited spark

energy as well as the energy released from the flame propagation. This eventually leads to the pre-chamber pressure equaling the main chamber pressure ($\Delta p=0$), followed by a continued increase in Δp . Then, along with the depletion of the chemical energy inside the pre-chamber, Δp starts to decrease and finally approaches a quasi-steady condition where the pressure in the pre-chamber equals the pressure in the main chamber.

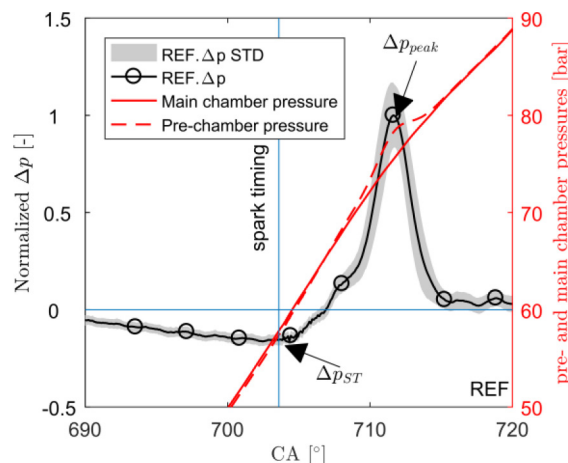


Figure 2. Normalized pressure difference Δp over crank angle with indications of its standard deviation and the spark timing. Δp_{ST} (Δp at spark timing) and Δp_{peak} (peak value of Δp) are marked by arrows. Δp is normalized using Δp_{peak} . The same normalization treatment is applied to all related values from experiments.

One thing to be noted here is that the measured Δp_{ST} , which ranges from -0.2 bar to -0.6 bar, is comparable to the measurement uncertainties, which is about ± 0.3 bar for the pre-chamber pressure sensor. Therefore, to determine accurate Δp_{ST} values, 0D calculation results were used, which had been properly validated using the 3D CFD simulations. All the measured Δp curves were then offset to the values calculated using the 0D simulations.

Interaction between Experiment, 0D Calculation and 3D CFD Simulation

The 3D CFD and 0D simulations used in this study have been validated using the experimental measurements. The relationship between them, together with the final correlation validation, are summarized in Figure 3. Measurements from the engine test bench provide the main chamber pressure profile as well as intake/exhaust pressure and temperature information which are used as initial/boundary conditions for the 3D CFD and 0D simulations. Outcomes from 3D CFD simulation provide in-pre-chamber physical and turbulence conditions which assist in deepening the understanding of Δp curves from experiments and the corresponding combustion processes. In addition, the 3D CFD-calculated Δp_{ST} and mass flow rate profile can serve as sources of verification for the 0D simulation, particularly for the nozzle flow model. Once the 0D model is verified, it can be used to deliver the values of Δp at spark timing for a large number of measurement conditions. It is worth noting that even though 3D CFD could also be used to obtain the values of Δp at spark timing, the computational costs are exorbitant for the large number of measurement points used in the correlation validation process.

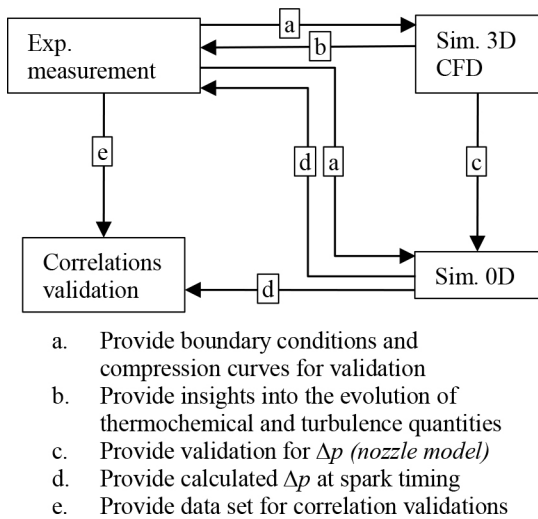


Figure 3. Workflow of the interactions between experiment, 0D calculation and 3D CFD simulation.

EXPERIMENTAL AND NUMERICAL SETUP

Experimental Setup

Engine Test Bench

The measurements were carried out on an instrumented research engine version of the in-line six cylinder Liebherr gas engine. The specifications of the engine configuration are listed in Table 1.

Table 1. Liebherr G946 Engine specifications

Engine specifications	
Displacement Volume	1.99 L/Cyl
NO. of Cylinder	6
Stroke	150 mm
Bore	130 mm
Number of Valves	4
Rated Speed	1500 rpm (50Hz) / 1800 rpm(60Hz)
Rated Power	246 kW (50Hz)

A gas flow mixer was used to ensure homogenous mixtures in the combustion chamber. Both the pressure sensor signals were recorded at a resolution of 0.1° CA in order to properly resolve pre-chamber combustion, which has a duration of the order of 10° CA. For each engine operating point, 250 consecutive cycles were recorded and the ensemble average was taken for the analysis to ensure the stabilization of the engine operating conditions. Intake temperature was well regulated with an intercooler while exhaust temperature was measured by a steady state thermocouple at the outlet of the exhaust port. Air mass flow rate was measured using Sensyflow N200 sensor. The air-fuel ratio was calculated using the measurement of the exhaust gas composition, utilizing the Brettschneider method [27].

Given that there is about 1 % uncertainty in measuring the species volume fraction, the calculated lambda values were reduced to two decimal points.

Operating Condition Matrix

The present study focuses on the engine operating conditions' effect on the pre-chamber performance. For this purpose, five groups of single parameter variations, resulting in a total number of 17 points, were involved in the testing matrix as shown in Table 2. The reference condition as well as the varying parameter in each sweep are given in bold. Within each sweep, a constant reference point is used to coordinate the range of variation and provide a reference for comparisons. Given the restrictions of engine knocking, cycle-to-cycle variation (CCV) as well as hardware limitations, the span of parameter variations and the reference condition were carefully chosen so that single parameter sensitivity analyses are feasible.

Numerical Setup

3D CFD Simulation for Spatially Resolved in-Pre-Chamber Turbulence

Cold flow 3D CFD simulations presented in this paper were conducted in an unsteady RANS (URANS) framework by using the STAR-CD v4.24 CFD [28] solver in conjunction with the es-ICE module [29] for mesh generation and controlling piston and valve motion. A well-known eddy-viscosity based two-equation turbulence model, namely Re-Normalization Group variant of the k-epsilon model (*RNG k-ε model*) [30], was used in this research. The sensitivity with respect to the choice of turbulence models for lean burn gas engine is reported in [31]. All related model constants were taken as default. The near-wall region was treated using a fine prism layer and the standard Angelberger correlation [28] was employed for calculating wall heat losses.

The full cylinder including the intake and exhaust runners were taken into consideration for the compression stroke calculation. In Figure 4, the computational domain is shown with the intake/exhaust ports and the pre-chamber marked. The cylinder including the moving piston, the valves and the intake/exhaust ports used trimmed mesh and one prism layer was applied on the wall. The pre-chamber was discretized independently in order to have better cell size control. The complete geometry of the pre-chamber was resolved, which was meshed with polyhedral cells and attached to the cylinder by means of coupling. The cells inside the pre-chamber were reduced to 0.5 mm in order to better resolve the flow and turbulence field. The cells at the nozzles were further scaled down to 0.12 mm due to the high velocity gradient from the jets. Three additional layers on the intake port inlet boundary and five layers on the exhaust port outlet boundary were applied to ensure the stability of the pressure boundaries. Concerning the rest of the computational domain, a grid independence analysis was conducted in [32] and an averaged cell size of 1.5 mm was used in this study. Overall, at BDC, the entire computational domain including intake and exhaust runners contained 1.4 million cells. At TDC, the combustion chamber domain consisted of roughly 0.5 million cells.

Table 2. Engine operating condition matrix

	Operating points	λ	ST(°BTDC)	Engine speed [rpm]	Boost pressure [bar]	T_{in} [K]
Lambda variation	LBD1	1.74	16.3	1500	2.49	318
	LBD2	1.71	16.3	1500	2.49	318
	LBD3 (REF)	1.69	16.3	1500	2.49	318
	LBD4	1.66	16.3	1500	2.49	318
	LBD5	1.63	16.3	1500	2.49	318
Speed variation	ES1 (REF)	1.69	16.3	1500	2.49	318
	ES2	1.69	16.2	1800	2.50	318
Boost pressure variation	BP1	1.69	16.3	1500	1.49	318
	BP2	1.69	16.3	1500	1.74	318
	BP3	1.69	16.3	1500	1.99	318
	BP4	1.69	16.3	1500	2.24	318
	BP5 (REF)	1.69	16.3	1500	2.49	318
	BP6	1.69	16.3	1500	2.69	318
Intake temperature variation	IntT1	1.69	16.3	1500	2.49	314
	IntT2 (REF)	1.69	16.3	1500	2.49	318
	IntT3	1.69	16.3	1500	2.49	323
	IntT4	1.69	16.3	1500	2.49	328
Spark timing variation	ST1	1.69	13.8	1500	2.49	318
	ST2 (REF)	1.69	16.3	1500	2.49	318
	ST3	1.69	18.8	1500	2.49	318
	ST4	1.69	21.2	1500	2.49	318

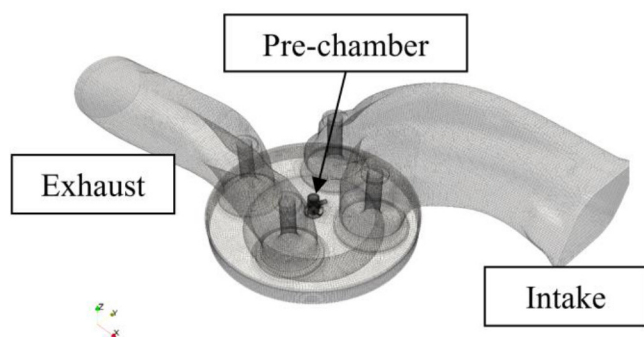


Figure 4. Computational domain including cylinder, pre-chamber and intake/exhaust ports

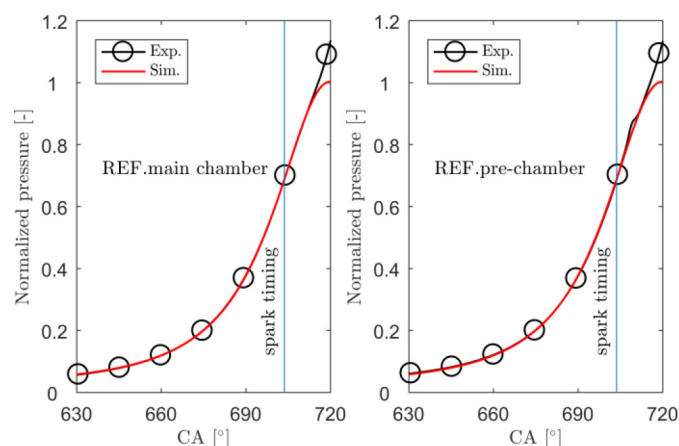


Figure 5. Comparison between measured and CFD-simulated main chamber (left) and pre-chamber (right) compression pressure normalized with the peak motored cylinder pressure.