6.3 Pre-chamber Ignition System for Homogeneous Lean Combustion Processes with Active Fuelling by Volatile Fuel Components

Moritz Schumacher, Tim Russwurm, Michael Wensing

Abstract

The combustion of homogeneous lean or diluted mixtures would significantly increase the efficiency of SI engines, but common spark ignitions systems are incapable to ignite these mixtures.
Pre-chamber ignition systems burn a small portion of the charge in a separated chamber, which is connected to the main chamber by multiple small orifices. The combustion in the pre-chamber generates hot gases, which penetrate the main chamber, increase the turbulence and ignite the mixture on multiple sites. This leads to an increased turbulent flame speed and an extended lean and dilution limit, if the mixture in the pre-chamber is kept stoichiometric.
Pre-chamber ignition systems have been investigated since the 1970s for passenger cars and are today commonly used in large gas engines. The adaption of the pre-chamber fuelling system to passenger car engines is not trivial, due to the problematic mixture preparation in the pre-chamber. Injection of a gaseous fuel in the pre-chamber would require a second fuel system with high pressure storage tank. Liquid gasoline direct injection in the pre-chamber is difficult due to the small space available for mixture preparation and the high surface to volume ratio, resulting in insufficient evaporation especially during cold start conditions.
To overcome this problem, we developed a pre-chamber ignition system with active fuelling by volatile fuel components, which facilitates the integration in passenger cars. The system uses a mixture of air saturated with gasoline vapour for the pre-chamber fuelling. This gaseous mixture is typically found in the fuel tank above the liquid level and hence available in passenger cars.
Former publications by the authors already proved the ability to enrich the pre-chamber and stabilize the combustion at homogeneous lean operation. Recent work focused on the optimization of the pre-chamber fuelling system and the pre-chamber geometry. To simulate the fuel tank atmosphere under different environment settings, a system was built, which saturates air with volatile gasoline components. This mixture gets compressed and dosed to the pre-chamber by a solenoid valve. Multiple prototypes of the pre-chamber with different volumes and geometry were investigated in a full engine at characteristic operating points regarding thermal efficiency, combustion process and emissions. These prototypes incorporate a spark plug, fuelling valve, thermocouple and pressure transducer. The results show the ability to ignite homogeneous lean mixtures with $\lambda \approx 2.0$. Optimum operation was achieved with $\lambda = 1.85$ at 4.5 bar IMEP and 1500 rpm. This operating point showed an efficiency gain of 15 % compared to stoichiometric spark plug operation and NO_x emissions below 20 ppm.
The technology enables the usage of actively fuelled pre-chambers in passenger cars. The volatile fuel components for the pre-chamber fuelling are available in the fuel tanks atmosphere and thus allow a single fuel solution with inexpensive components.

Kurzfassung

Homogen magere oder durch Restgas verdünnte Zylinderladungen können die Effizienz von Otto-Motoren signifikant erhöhen. Jedoch sind gewöhnliche Zündsysteme nicht in der Lage, homogen magere Gemische zu entflammen.
Vorkammer-Zündsysteme entflammen eine geringe Ladungsmenge in einer separaten Kammer, die durch mehrere kleine Bohrungen mit dem Hauptbrennraum verbunden ist. Die Verbrennung in der Vorkammer generiert heiße Gase, die in den Brennraum eindringen und dort die Turbolenz erhöhen und die Ladung an mehreren Stellen gleichzeitig entzünden. Damit wird die turbulente Flammengeschwindigkeit erhöht und die Magergrenze erweitert, wenn das Gemisch in der Vorkammer stöchiometrisch bzw. zündfähig bleibt.
Vorkammer-Zündsysteme wurden bereits seit den 1970er Jahren für PKW untersucht und werden heute gewöhnlich in großen Gasmotoren eingesetzt. Die Adaption des Vorkammer-Kraftstoffsystems auf einen PKW ist auf Grund der problematischen Gemischaufbereitung in der Vorkammer komplex. Das Einblasen von gasförmigem Kraftstoff würde ein zweites Kraftstoffsystem mit einem Hochdruck-Gastank benötigen. Flüssige Kraftstoffe direkt in die Vorkammer einzuspritzen ist kritisch, da nur wenig Platz zur Gemischaufbereitung vorhanden ist und durch das hohe Oberflächen- zu Volumen Verhältnis besonders im Kaltstart schlechte Verdampfungsbedingungen vorliegen.
Um dieses Problem zu lösen haben wir ein Vorkammer-Zündsystem mit aktiver Spülung durch leichtsiedende Komponenten des Otto-Kraftstoffes entwickelt, welches die Fahrzeugintegration vereinfacht. Dieses System verwendet ein Gemisch aus Luft und Kraftstoffdämpfen zur Kraftstoffanreicherung der Vorkammer. Dieses Gemisch kann typischerweise in Kraftstofftanks oberhalb des Flüssigkeitsspiegels gefunden werden und ist somit in PKW verfügbar.
Frühere Publikationen der Autoren haben bereits gezeigt, dass die Anreicherung der Vorkammer so erfolgen kann und eine Stabilisierung der Verbrennung bei mageren Betriebspunkten möglich ist. Diese Arbeit konzentriert sich auf die Optimierung des Vorkammer-Kraftstoffsystems und die Verbesserung der Geometrie der Vorkammer. Um die Tankatmosphäre bei verschiedenen Umgebungsbedingungen zu simulieren, wurde ein System zur Anreicherung von Luft mit Kraftstoffdämpfen aufgebaut. Dieses Gemisch wird verdichtet und über ein Magnetventil getaktet der Vorkammer zugeführt. Mehrere Prototypen der Vorkammer mit unterschiedlichen Volumen und Geometrien wurden im Vollmotor bei charakteristischen Betriebspunkten hinsichtlich der thermischen Effizienz, des Verbrennungsprozesses und der Emissionen untersucht. Diese Prototypen enthalten eine Zündkerze, ein Spülungsventil, ein Thermoelement und einen Drucksensor. Die Ergebnisse zeigen die Fähigkeit des Systems Gemische bis $\lambda \approx 2{,}0$ zu entflammen, wobei ein optimaler Betrieb bei $\lambda = 1{,}85$, 4,5 bar IMEP und 1500 1/min erzielt wurde. An diesem Betriebspunkt konnte die Effizienz um 15 % gesteigert werden bei NO_x-Rohemissionen unter 20 ppm.
Diese Technologie ermöglicht den Einsatz einer aktiv gespülten Vorkammer in PKW. Die leichtsiedenden Kraftstoffanteile für die Vorkammer sind in der Tankatmosphäre vorhanden und erlauben eine Ein-Kraftstoff-Lösung mit günstigen Komponenten.

1 Introduction

Facing stricter emission limits and imminent bans on diesel vehicles entering city centres, the gasoline engine remains an important bridging technology to a fully electric mobility sector. In principle, gasoline engines offer much better raw emission characteristics in real operation than diesel engines. However, the efficiency of diesel engines is not achieved with conventional stoichiometric, homogeneously operated gasoline engines. The thermal efficiency of gasoline engines can be increased significantly above current levels through high compression ratios and the combustion of lean mixtures or mixtures diluted with exhaust gas. The compression ratio has a direct effect on the efficiency of the gasoline process, while the charge dilution has an indirect effect via a higher isentropic exponent of the diluted and therefore colder charge. Further advantages result from reduced wall heat losses due to lower combustion temperatures and reduced charge exchange losses due to dethrottling.
With conventional ignition systems, however, the poor flammability of highly diluted air-fuel mixtures and the increased tendency to knock at high compression ratios represent a hurdle. The direct injection with stratified charge enables the thermodynamic benefits of lean mixtures but it leads to high NO_x emissions and is only possible in a limited operating range due to cyclic fluctuations in the mixture formation.
To meet the high requirements for ignition systems in modern engines many concepts have already been discussed in the technical literature. The aim is usually to extend the electrical discharge in time and space to increase the probability of stable flame core formation. A distinction must be made between direct current systems and alternating current systems. In the former case, several conventional ignition coils can be connected in parallel and operated alternately or simultaneously, thus producing a strong spark with virtually no time limit [1, 2]. In spatial terms however, the spark remains limited to the gap between the electrodes. In contrast, a corona ignition using high-voltage alternating current with frequencies in the MHz range can achieve a wide spatial expansion of the discharge. The entire combustion chamber is used as a counter electrode and a strong inhomogeneous electric field is generated which locally exceeds the ionization threshold, resulting in a corona discharge [10]. In contrast, laser ignition works without electrical discharge, whereby a pulsed laser beam is focused in the combustion chamber to locally ionize the mixture. The resulting plasma strongly absorbs the laser radiation, resulting in a rapid increase in temperature and pressure, which ignites the charge [3].
These systems have in common, that the ignition energy is taken from the vehicle's electrical system, limiting the possible ignition energy to a few 100 mJ per event. Furthermore, the systems mentioned are limited to ignition alone; an additional function, such as increasing the charge motion, is not possible.
Pre-chamber ignition systems burn a small portion of the charge in a separate chamber, which is connected to the main chamber by multiple transfer ports. The hot gases generated by the pre-chamber combustion penetrate the main chamber, increase the charge motion and ignite the mixture on multiple sites. The energy for ignition is drawn from a chemical reaction, whereby the possible ignition energy of electrical systems is far exceeded. The ignition via a spark plug in the pre-chamber is possible if the mixture in the pre-chamber is kept near stoichiometric. With lean or diluted mixture in the main chamber a separate fuelling system for the pre-chamber is necessary.
Such systems have been developed since the 1970s and are today commonly used in large natural gas engines. The early investigations until 1975 where summarised by Roessler and Muraszew [4]. Dale and Oppenheim [5] later reviewed advanced ignition

systems in 1981 and categorised the pre-chamber systems based on the pre-chamber size and the specific size of the transfer ports. Toulson, Schock and Attard wrote a more recent review on pre-chamber ignition systems in 2010 [6].
The pre-chamber ignition systems can be categorised based on the pre-chamber volume and the specific size of the transfer ports. Gussak et al. showed in their work, that a pre-chamber volume of 2…3 % of the combustion chamber volume and a transfer port cross-section area of 0.03… 0.04 cm^2/cm^3 (referenced to the pre-chamber volume) lead to the most favourable results [7, 8].
Gussak et al. used a fuelled pre-chamber, which was designed in such a way that the burning gases from the pre-chamber are strongly cooled in the transfer channels, whereby the combustion of the gases stops. According to Gussak's explanation, the resulting gas jets contain partially burnt, active radicals that ignite the mixture in the cylinder at many points simultaneously. In further research on this topic, Yamaguchi et al. were able to identify four possible ignition processes. These range, depending on the specific cross-sectional area of the transfer channels, from a pure chemical chain reaction via a combined active radical and thermal reaction to a pure flame propagation through the transfer channels [9]. The work of Gussak and Yamaguchi shows the most important design parameters of a pre-chamber ignition system:

- A large volume of the pre-chamber provides more energy for ignition and charge motion generation in the combustion chamber but requires more fuel and increases wall heat losses.
- The geometry of the transition channels and the wall temperature of the pre-chamber influence the chemical reactivity and temperature of the escaping gas jets and thus the ignition process in the combustion chamber.

An essential requirement of pre-chamber ignition systems is that the mixture inside the pre-chamber must be ignitable. In the case of a lean mixture in the combustion chamber, the pre-chamber must therefore be enriched with fuel. The fuel used to enrich the pre-chamber is another important design parameter. The smaller the pre-chamber, the more difficult it is to meter the required quantity of fuel. Due to the small dimensions and charge motion in the pre-chamber, mixture formation with liquid fuels is very demanding and gaseous fuels are preferred [10–12].
The use of gaseous fuels instead of liquid fuels additionally facilitates the metering of the fuel quantity and mixture formation in the pre-chamber, because the volumes are considerably larger due to the lower density. Pre-chamber ignition systems that are purged or fuelled with gaseous fuels have been intensively investigated over the past 20 years. Very high lean running limits of $\lambda > 2$ could be achieved with hydrogen as pre-chamber fuel by Watson et al. [13, 14]. Their so-called "Hydrogen Assisted Jet Ignition" (HAJI) was further developed by Watson, Boretti and Toulson and also investigated with propane and methane as pre-chamber fuel [15, 16]. The system uses a gasoline direct injector for the pre-chamber, other concepts use gas valves in combination with a check valve, such as the APIR system by Couet et al. [10]. Similar systems using methane for pre-chamber flushing have been investigated by Geiger et al. [17] and Getzlaff et al. [11].
The "Turbulent Jet Ignition" (TJI) by Attard et al. from Mahle Powertrain LLC [18] marks the state of the art in the field of combustion processes with pre-chamber ignition for passenger cars. Their concept uses a gasoline direct injector positioned directly in the pre-chamber. First experiments were carried out with gaseous fuels for pre-chamber

enrichment, followed by measurements with vaporized gasoline [19] and direct injection of liquid gasoline into the pre-chamber with the aid of a modified injector [12]. With vaporized gasoline as fuel for the pre-chamber, an indicated efficiency of 41.9 % with NO_x emissions below 10 ppm could be achieved at medium engine load [19]. This system was further developed at Mahler Powertrain LLC by Bunce and Blaxill et al., whereby detailed combustion analyses, investigations in optically accessible engines [20, 21], as well as RANS-based simulations were carried out [22]. With Mahle's TJI system a very high lean operating limit of $\lambda > 2$ and very fast combustion speeds could be achieved. However, recurring problems occurred due to deposits in the direct injector of the pre-chamber. This injector was operated with low fuel pressure and extremely small injection holes to dose the required low mass flows. This operation is expected to cause considerable coking problems and the self-cleaning of the injector tip observed with conventional direct injectors does not seem to work [21]. Furthermore, due to the positioning of the direct injector in the pre-chamber, the system requires a lot of installation space which is not available in modern gasoline engine cylinder heads without massive design changes.

Pre-chamber ignition systems are currently used in large gas engines, whereby the described advantages can be fully exploited [23, 24]. On a passenger car scale, equally high lean limits with associated increased efficiency at lowest NO_x emissions are achieved [19]. The use in passenger car engines has so far failed due to the availability of a gaseous fuel for enriching the mixture in the pre-chamber.

The core idea of the concept in this work is therefore to generate the required mixture of gaseous fuel and air for the pre-chamber in the vehicle. The gas atmosphere in the fuel tank above the liquid level is used for this purpose. This gasoline-vapour-air-mixture mainly contains volatile fuel components such as butane and pentane with a concentration of approx. 60 wt. % at room temperature. The use of this gases eliminates the need for additional fuel for the pre-chamber and the tank ventilation system common in passenger cars already provides some of the required components.

The authors showed in pervious publications that the volatile components of common gasoline fuels can be used to enrich the pre-chamber sufficiently for stable combustion processes up to $\lambda = 1.6$. This work shows results of an advanced fuelling system and pre-chamber, as wells as the influence of different vapour conditions and different geometrical characteristics on the combustion process.

2 Methodology

2.1 Test engine setup

As test engine a common 1.8 L four-cylinder DI-engine with turbocharger and variable intake and exhaust valve phasing was used. The engine was operated with 100 bar injection pressure and 95°C ± 1°C coolant temperature. The oil temperature is limited to 95°C ± 2°C by the internal oil-water heat exchanger of the engine. To reduce the samples of prototypes only one cylinder is fired while the other three are closed by a sealing plate in the intake. The main geometrical characteristics of the engine are shown in Table 1.

The operation point is kept steady over all measurements to 1500 rpm and 4.5 bar indicated mean effective pressure IMAP. This point was identified as a medium load point with high relevance to WLTP fuel consumption earlier [25].

The engine is equipped with a low-pressure sensor in the intake and tempered high pressure sensors Kistler 6041B in the cylinders. Both the pressure in the pre-chamber and in the cylinder were measured with a resolution of 0.36°CA and averaged over 100 cycles. All energetic calculations include the fuel mass flow of the combustion chamber and the pre-chamber, whereby both are measured by coriolis mass flow meters.

Table 1: geometrical characteristics of the test engine

Displacement volume per cylinder	450 ccm
Stroke	84.1 mm
Bore	82.5 mm
Rod length	148 mm
Compression ratio	9.6
Valves per cylinder	4
Intake valve diameter	33.9 mm
Exhaust valve diameter	28.0 mm
Injector position	side

2.2 Pre-chamber fuelling system

The focus of this work is the analysis of the behaviour of a pre-chamber ignition system with active fuelling by volatile fuel components. As already mentioned, these volatile fuel components are available in the gas atmosphere in typical gasoline tanks. The concentration of the volatile fuel or HC components in the tank atmosphere depends on the vapour pressure of the gasoline and the pressure and the temperature inside the tank. The amount of fuel injected into the pre-chamber must be metered so that a stochiometric mixture is reached inside the pre-chamber at spark timing. In case of a lean mixture in the main chamber an under-stochiometric air-gasoline-vapour mixture must be injected in the pre-chamber. The gasoline-vapour-air-mixture in the fuel tank is very fuel-rich and therefore suitable to enrich the mixture in the pre-chamber. As the gasoline-vapour or HC-concentration in this mixture depends on the surrounding conditions, the injected mass into the pre-chamber must be corrected to the HC-concentration. It is also possible to control the HC-concentration of the injected mixture by additional devices to increase the scavenging volume and the oxygen content in the pre-chamber.
To investigate the influence of these parameters, different fuel vapour concentrations and fuel mass flows into the pre-chamber were investigated. These measurements require a flexible fuel system in both volume and HC-concentration. At 20°C the fuel vapour in the fuel tank contains around 60 wt. % HC, mainly propane and butane. A typical detailed composition of the fuel vapour is shown in [26].

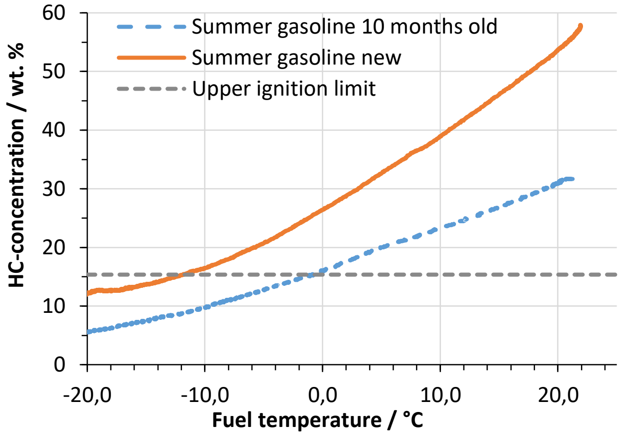

Figure 1: HC-concentration in the vapour versus fuel temperature

The main idea behind the fuelling system used for this study is the dependence of vapour pressure on temperature. Figure 1 shows the HC-concentration in the air-gasoline-vapour-mixture versus the gasoline temperature. Thus, different HC-concentrations can be achieved through variant fuel temperatures. Additionally, different environmental conditions like engine start in the winter can be simulated.

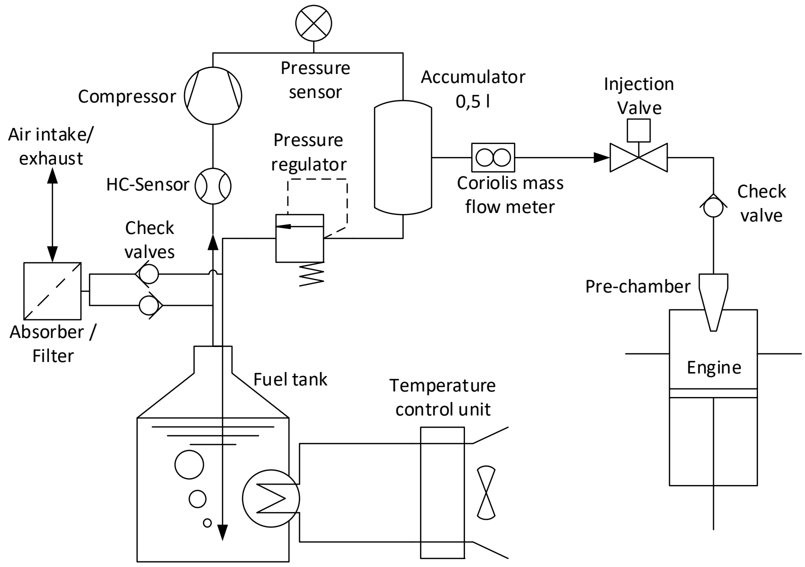

Figure 2: flow chart of the pre-chamber fuelling system

Figure 2 shows the flow chart of the fuelling system. The system contains a 20-litre fuel canister with integrated coolant pipes to temper the gasoline according to the desired HC-concentration in the area above the fuel level. To keep enough area for gasoline vapour, the canister is filled with no more than 10 litre gasoline. The connected external cooling device can be set between -20°C and +60°C. At the top of the fuel tank, a fuel resistant vacuum pump sucks off the gasoline-vapour-air-mixture. The sensor for the HC-concentration is positioned directly before the pump. After the compressor the pressure is measured, followed by a small pressure accumulator tank. This accumulator is needed to dampen pressure pulsations from the pump. Additionally, the line to the engine is positioned in the middle of the tank.
Thus, condensed gasoline components drop of and are feed back to the fuel tank. Condensation occurs if the partial pressure of one component exceeds its vapour pressure for the actual temperature. Condensation can be prevented by increasing the temperature or reducing the absolute or partial pressure of the relevant component.
At the bottom of the accumulator tank a second connection leads back to the bottom of the fuel tank after passing the pressure controller. The compressor's discharge flow exceeds the necessary value for fuelling the pre-chamber, so there is always a certain amount of gasoline-vapour-air-mixture that circulates back into the tank. In that way, the fuel pressure can be kept constant for different consumptions of the pre-chamber.
A check valve between the pressure controller and the fuel tank supplies fresh air to the system when mixture is delivered to the pre-chamber. As the air intake line ends at the bottom of the fuel tank, there is always a circulation of gas through the liquid gasoline leading to a constant HC-concentration in the gasoline-vapour-air-mixture.

2.3 Pre-Chamber system

The entire pre-chamber setup contains the pre-chamber fuelling system, the solenoid valve and the pre-chamber. In the pre-chamber body a check valve is included to seal the injection line against combustion pressure. This enables a very compact design of the pre-chamber and its injection line, so the entire system fits into a common cylinder head by replacing the spark plug. The pre-chamber itself is made from a high-strength copper alloy, that was already used earlier as pre-chamber material [25]. Figure 3 shows the measurement setup.

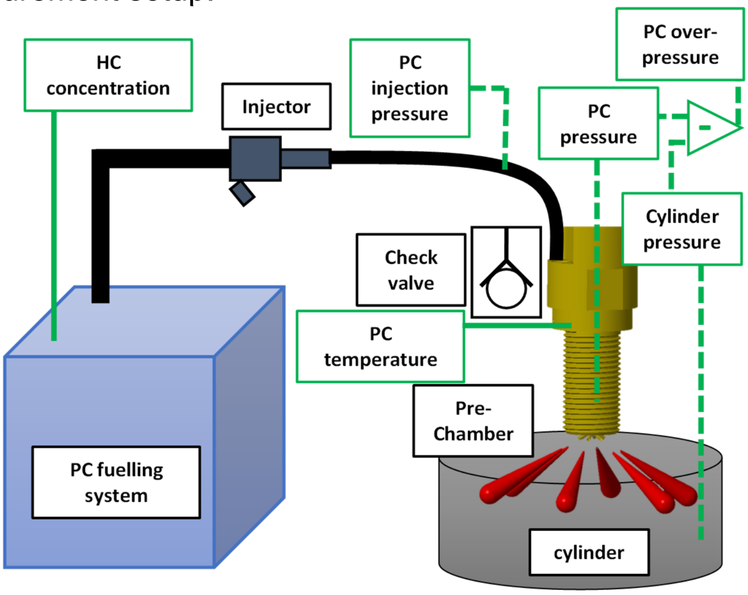

Figure 3: Pre-chamber system

To investigate the pre-chamber pressure, a Kistler 6113B pressure measuring spark plug was installed. The pressure in the pre-chamber injection line between injector and check valve is also measured. To characterize the performance of the pre-chamber, the signal pre-chamber overpressure was created artificially by subtracting the cylinder pressure value from the pre-chamber pressure value. Thus, the pre-chamber ignition can be evaluated online during the measurements regarding timing and intensity.
Apart from the pressure measurements, the setup also records the temperature of the pre-chamber body with a thermocouple and the HC-concentration as shown in 2.2.

2.4 Pre-chamber geometry

In this work, four different pre-chambers were analysed. Thereby the following three parameters were variated:

- Pre-chamber volume
- Diameter of the transfer holes
- Length of the transfer holes

All pre-chambers have six orifices for the connection between pre-chamber and cylinder. Table 2 summarizes the main geometrical properties of the used pre-chambers.

Table 2: Pre-chamber geometry

Pre-chamber	Volume / cm^3	Hole diameter / mm	Hole length / mm
VK 1	1.34	0.9	1.9
VK 1-1	1.34	1.0	1.9
VK 3	1.70	1.0	1.9
VK 4	1.70	1.0	4.5

The pre-chamber volume corresponds to 2.5 % or 3.25 % of the compression volume, the transfer port cross-section/volume ratio has been constant to ≈ 0.028 cm^2/cm^3 for VK 1, VK 3 and VK 4 whereas VK 1-1 has a larger ratio of 0.035 cm^2/cm^3.

3 Results

3.1 Pre-Chamber geometry

The various pre-chamber geometries were tested at a variation of lambda. Figure 4 shows the combustion duration from 5 % to 90 % mass fraction burned (MFB in crank angle, the 50 % MFB (center of combustion and the combustion stability (covariance of IMEP for the three tested geometries. The start of combustion in the main chamber was defined as timing with the highest pressure rise rate of the pre-chamber overpressure. Mathematically, this is the inflection point of the rising edge in the pre-chamber overpressure signal.

All pre-chambers were fuelled with a gasoline-vapour-air-mixture that contains an HC-concentration of around 45 % in the pre-chamber fuelling system. The pre-chamber fuelling mass was optimized in a previous step.

It is visible, that the geometry has a major impact on the performance of the pre-chamber ignition system. In general, the small orifice diameter of 0.9 mm shows the least promising behaviour. It shows the longest combustion duration and the latest centre of combustion. This was controlled to 8°CA after TDC if possible, VK 1 was set to 20°CA due to poor general running stability and long burning durations.

The longer orifices of VK 4 do not lead to a better stability of the combustion. On the contrary, the combustion duration with VK 4 was increased compared to VK 3 with the shorter holes. In the rich section, both the long holes and the smaller diameter show stability problems, running the engine at stochiometric mixture was not possible.

Both VK 3 and VK 1-1 show a good overall performance. The lean limit for VK 1-1 can be set to λ ≈ 1.85 with a COV of less than 4 %. The smaller pre-chamber volume shows a slightly shorter combustion duration at lean mixtures whereas the lager pre-chamber can reach a very good stability and short burning durations around λ = 1. The indicated efficiency raises along with higher fuel-air-ratio as long as the combustion stability remains under 5 % COV. VK 1 and VK 4 show lower efficiency due to later centre of combustion. At λ 1,85 VK 1-1 reaches an indicated efficiency of 37 %.

In order to achieve low NO_x raw emissions, a high fuel-air ratio is required. Figure 5 shows the NO_x raw emissions logarithmically plotted over lambda during operation with VK 1-1. This clearly indicates that a NO_x concentration of less than 20 ppm can only be achieved with λ > 1.8. At this operating point, the load on the exhaust aftertreatment system can be significantly reduced.

To get a closer impression on the pre-chamber behaviour, the pressure curves can be analysed. As an example, the values for cylinder pressure and pre-chamber pressure for VK 3 at λ = 1.65 are shown in Figure 6. Additional to these two measured values, the pre-chamber overpressure is drawn in Figure 6. It is defined as the difference between pre-chamber and cylinder pressure. This value can be used to compare different pre-chamber geometries at a predefined operating point – while for each pre-chamber design the timing showing the optimum efficiency is applied.

The pre-chamber overpressure at λ = 1.65 for the four different geometries is shown in Figure 7. In this comparison VK 4 shows the earliest ignition timing, but the ignition delay between pre-chamber and main cylinder is so long, that the MFB 50 % point cannot be earlier than 11° after TDC. With both VK 1-1 and VK 3 the desired MFB

50 % of 8° after TDC is possible. The pre-chamber VK 1 ignites very slowly but develops the highest peak pressure of around 15 bar. Nevertheless, VK 1 shows the poorest running stability due to longest burning duration and the late pressure peak.

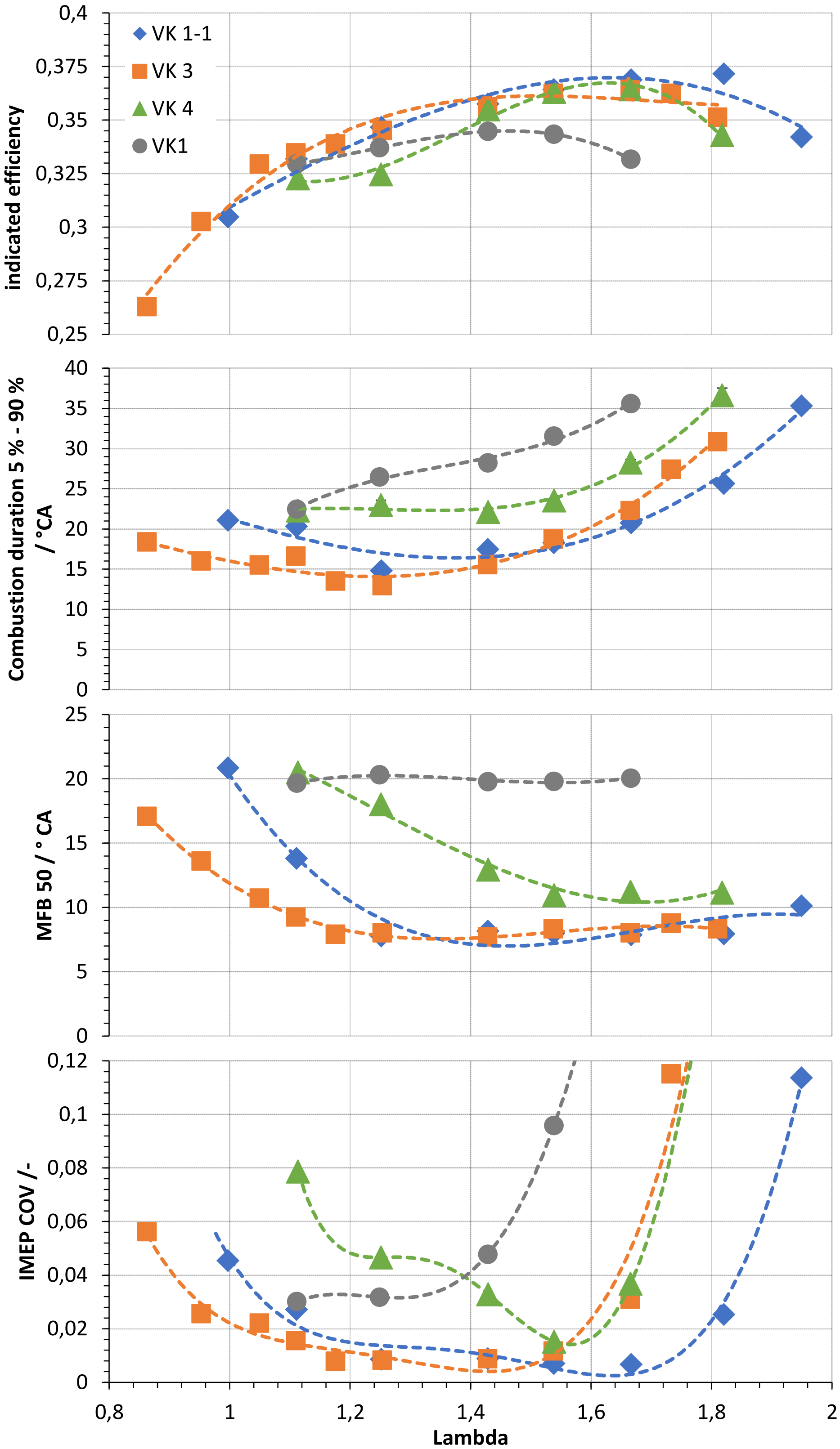

Figure 4: Variation of lambda with four different pre-chamber geometries at 4.5 bar IMEP and 1500 rpm

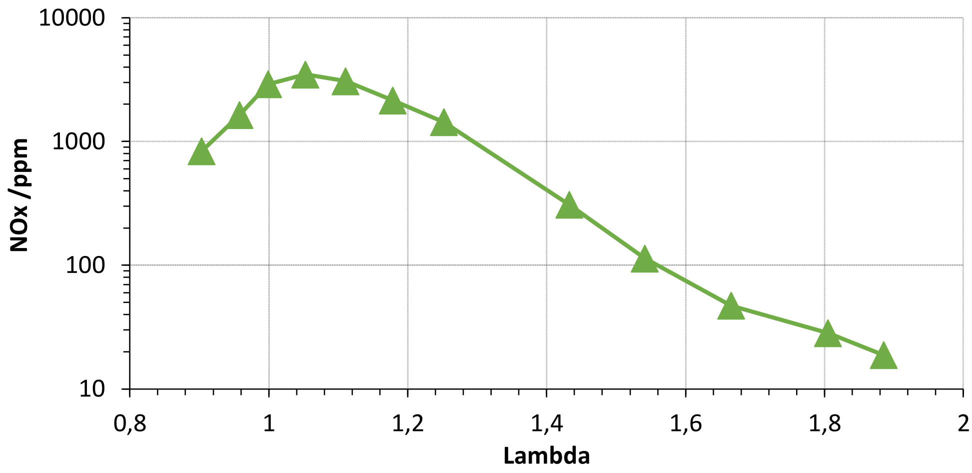

Figure 5: NO_x raw emissions for VK 1-1 at 4.5 bar IMEP and 1500 rpm

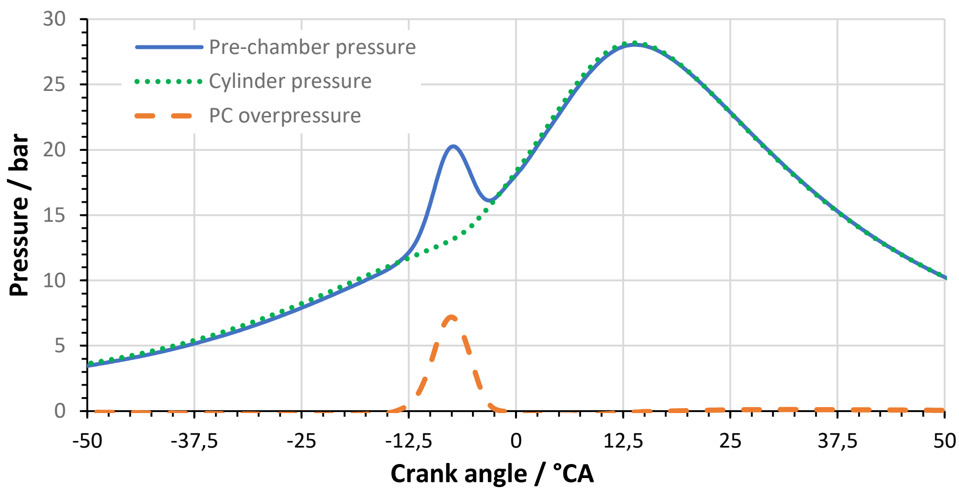

Figure 6: Pressure in pre-chamber and cylinder with PC overpressure for VK 3 at λ = 1.65, 4.5 bar IMEP and 1500rpm

In Figure 7 a clear insight into the physical conditions of the pre-chamber can be gained. The observation of VK 4 shows that the longer connection orifices between pre-chamber and cylinder directly generate a higher ignition delay between the pre-chamber and main combustion chamber. The desired centre of combustion at 8° cannot be achieved, although a sufficient overpressure can be generated in the pre-chamber at an early timing. This indicates that the combustion reaction is more delayed by the higher heat dissipation in the longer boreholes, so that there is a longer time until combustion starts in the main combustion chamber.

The comparison between VK 1 and VK 1-1 clearly shows the influence of the borehole diameter. VK 1 delivers a significantly slower reaction in the main cylinder and thus also a lower combustion stability, although the highest overpressure is generated in the pre-chamber. It was also not possible to achieve the desired combustion centre with this geometry. The combustion in the pre-chamber could not be started at a sufficiently early point in time. The comparison with VK 3 indicates that the worsening is related to the absolute bore diameter of 0.9 mm, not to the ratio of the bore cross-section to the pre-chamber volume. The smaller diameter leads both to a poorer mixture preparation in the pre-chamber and to a long ignition delay from the pre-chamber into the main combustion chamber.

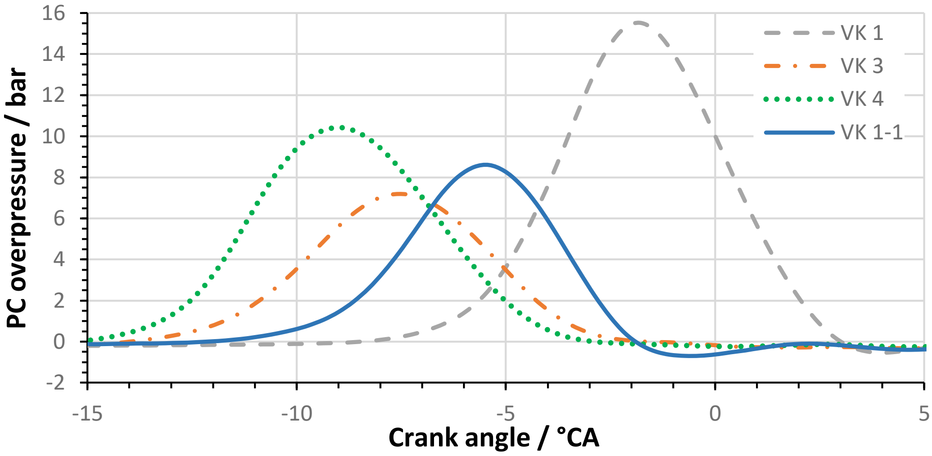

Figure 7: Pre-chamber overpressure at λ = 1.65, 4.5 bar IMPEP and 1500 rpm

3.2 Pre-Chamber fuel mass flow

One focus of this work was evaluating the pre-chamber performance at different fuelling conditions. To investigate this, both operation point and pre-chamber geometry was kept constant. The fuel-air-ratio was set to λ = 1.65 and all investigations were made with geometry VK 3, which showed a good performance in this operation point previously. Figure 8 shows a variation of the fuel mass flow to the pre-chamber at different HC-concentrations in the vapour. The different HC-concentrations in the vapour were achieved by alternating the temperature of the fuel in the pre-chamber fuelling system, as described earlier.

The first diagram shows that the system can supply the pre-chamber with the desired HC-concentration during the entire measurement. The temperature was set to -4°C for 28 %, 7.5°C for 45 % and 20°C for 63 % HC in the vapour. On the horizontal axis the fuel mass flow into the pre-chamber is plotted. This is calculated from the mass flow into the pre-chamber that is measured via a coriolis flow sensor and the HC-concentration of the gasoline-vapour-air-mixture. The second plot shows the opening time of the used solenoid injector to dose the flow. The timing for this injector fixed for all operating points to start of injection at 270 °CA before TDC. The running stability is again shown in the covariance of the IMEP. With the medium HC-concentration a wide band of fuel mass flows lead to a stable combustion. Only the smallest amount of fuel (≈ 19 g/h shows a slight increase of incomplete combustion.

The scavenging with the leaner mixture with 28 % HC shows some issues in the low mass flow region. However, when 40 g/h is reached, all HC-concentrations can provide a suitable amount of fuel-air-mixture in the pre-chamber.

Very rich mixtures beyond 60 g/h can only be reached with the rich vapour with the used injector. An injector with higher flow rate would be necessary to reach higher mass flow rates. A good firing condition with all HC-concentrations is reached with a supply of 40 g/h in this operating point. The used pre-chamber injector shows a suitable flow rate for medium HC-concentrations. The optimal injector size depends on the HC-concentration that is used in the system.

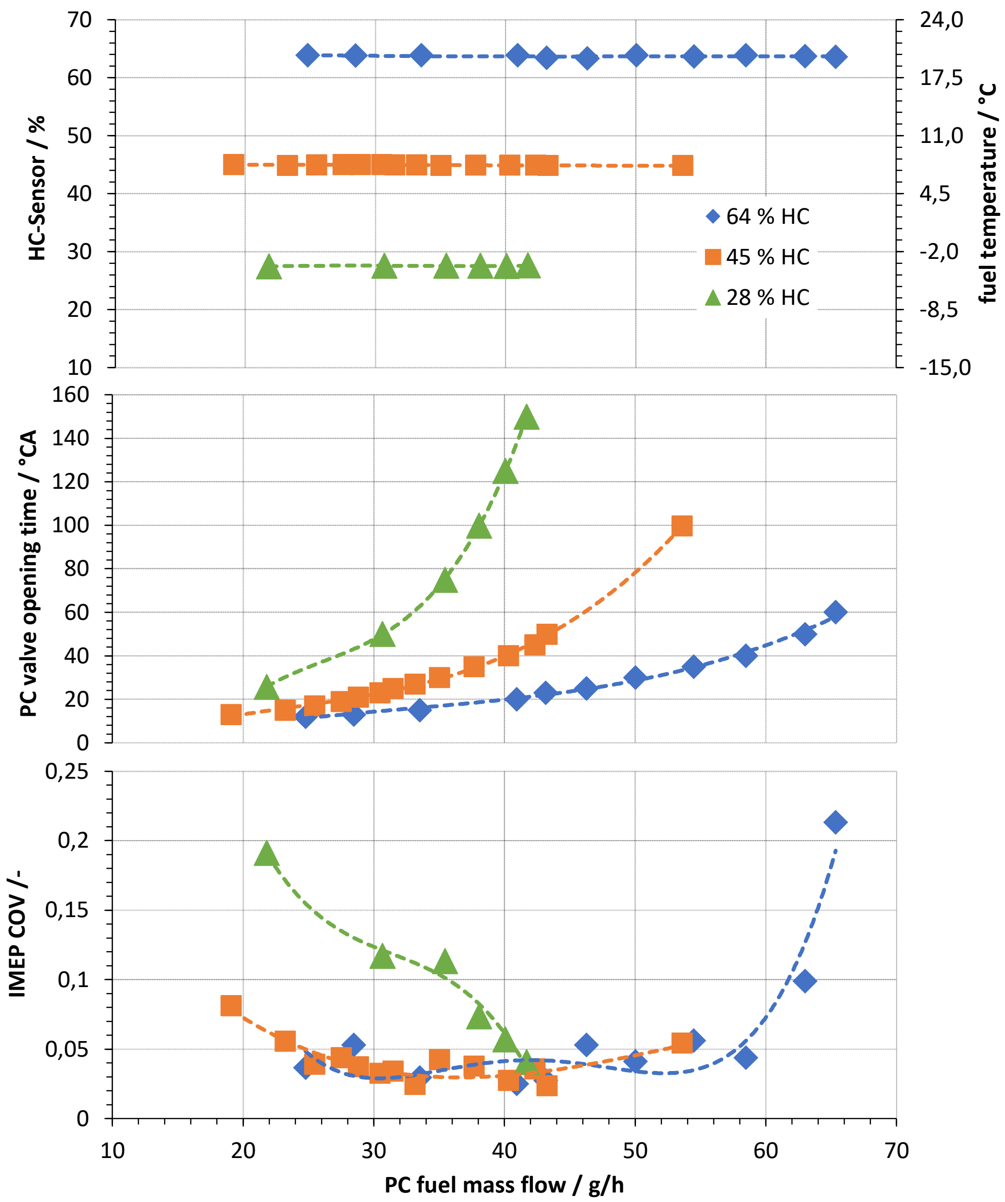

Figure 8: Variation of PC fuel mass flow for different HC-concentrations with VK 3 at λ = 1.65, 4.5 bar IMEP and 1500 rpm

The combustion timing shows also an interesting behaviour in this variation. In Figure 9 it is shown, that the desired centre of combustion is closely reached by all measurement points.

However, there is a significant difference between the measured HC-concentrations. Just as Figure 8 already indicates, the low HC-concentration shows longer combustion durations at low fuel mass flows. Combustion stability is insufficient even if the combustion duration can be compensated with an earlier ignition timing (the centre is close to 8°CA before TDC). From Figure 8 it is known, that the high HC-concentration shows problems at very high mass flow rates, due to over rich pre-chamber. The IMEP COV is above 10 %. This is not visible in the timing diagrams. When the pre-chamber ignites, the burning duration is on level with the other values. Nevertheless, it is not visible, that more than one out of ten cycles do not lead to any combustion at all. The only hint is the larger variance of the 50 % MFB point.

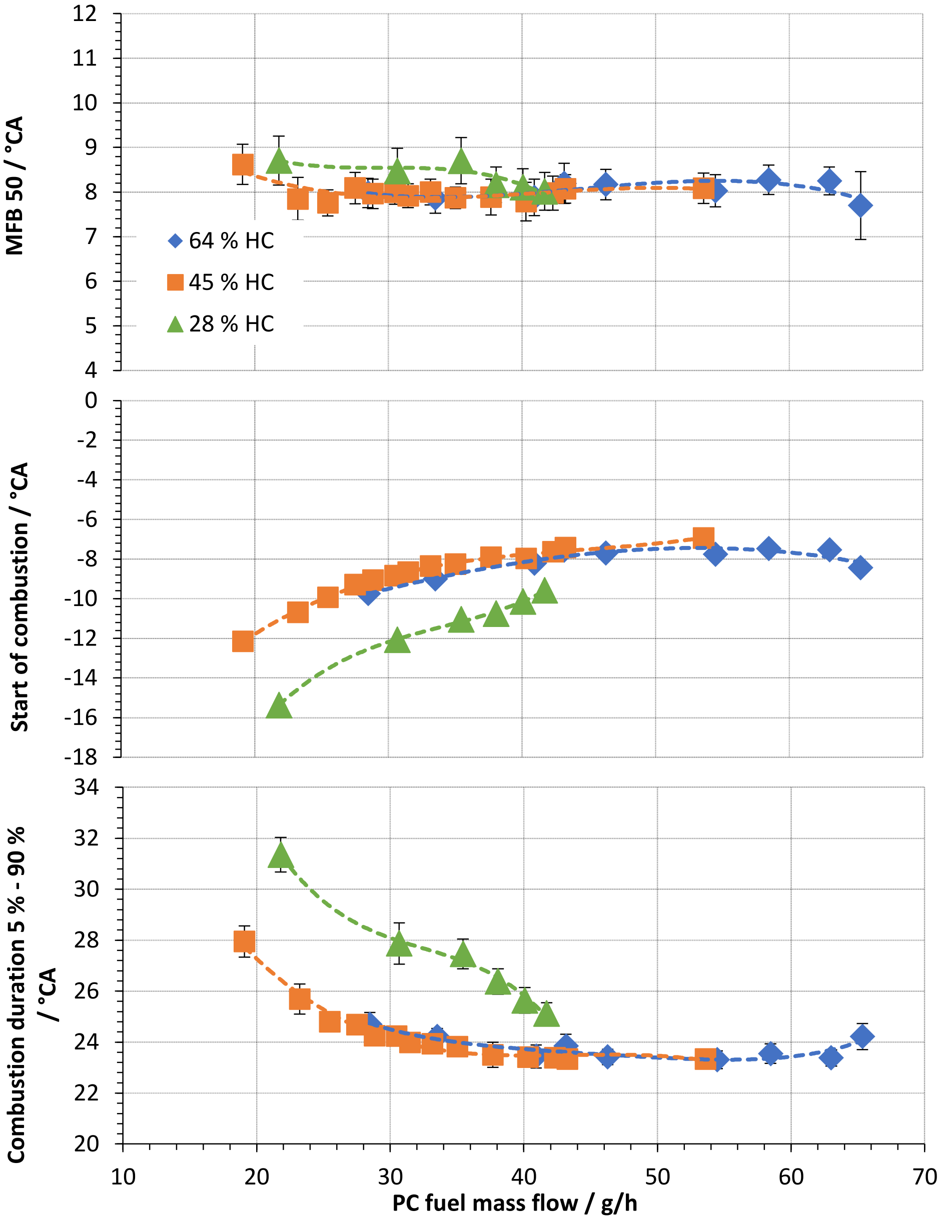

Figure 9: Combustion timing values for the variation of PC fuel mass flow for different HC-concentrations with VK 3 at λ = 1.65, 4.5 bar IMEP and 1500 rpm

In the design of the active scavenged pre-chamber ignition system, the supply of air-gasoline-vapour-mixture into the pre-chamber is one major objective of development. As explained earlier, this system uses the combination of a solenoid fuel injector and a check valve to dose the gasoline-vapour-air-mixture into the pre-chamber. Figure 10 shows the pressure in the injection line of the pre-chamber between injector and check valve together with the triggering signal of the injector. The injector is triggered at 270°CA before TDC and opens for 20°CA, 30°CA and 60°CA, respectively.
The graphs in Figure 10 show that the setup is working well in all conditions. Opening of the injector leads to a quick raise in the pressure, dependant on the opening duration. This pressure peak relaxes into the pre-chamber until the compression pressure is higher than the line pressure due to the piston movement. Only a very small amount of air-gasoline-vapour-mixture is pushed back into the line until the check valve closes. This is visible in the small pressure raise between 120°CA and 60°CA before TDC. Generally, the pressure signal shows a small spread with, so the pre-chamber did not suffer from unstable fuelling conditions.

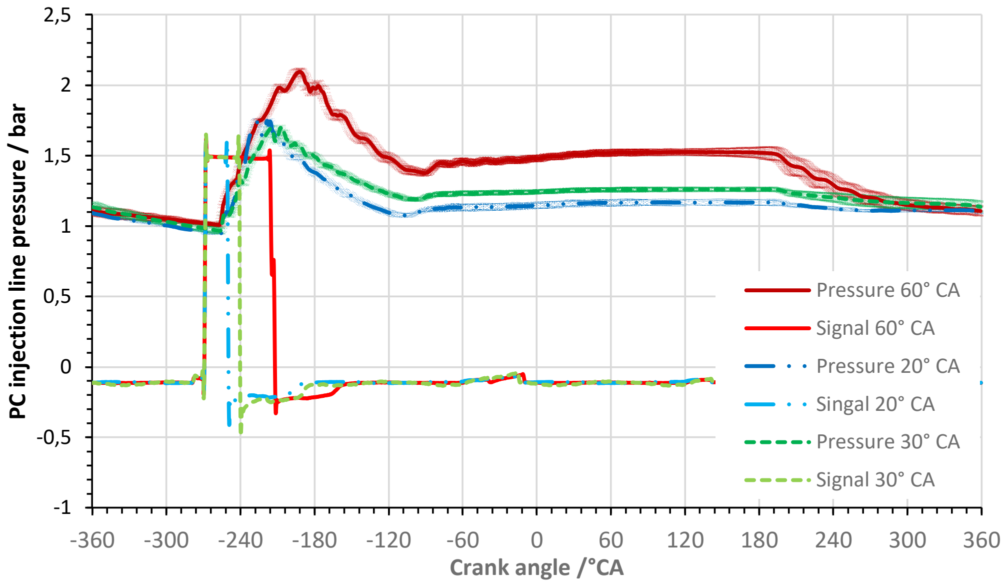

Figure 10: Pressure in the PC fuel line between solenoid injector and check valve for long, medium and short injector duty cycle

3.3 Active Pre-Chamber scavenging with air

One problem of passive pre-chamber systems without active scavenging is the tolerable EGR-rate. Inside the pre-chamber a higher concentration of exhaust gasses can be found due to the scavenging conditions. This problem can also be found at high EGR-rates when the pre-chamber is scavenged with a rich mixture of air and fuel vapour. To overcome this problem, an additional solenoid valve was installed to the pre-chamber fuelling line to inject air additionally to the air-gasoline-vapour-mixture. Thus, the amount of oxygen inside the pre-chamber can be increased, what results in better combustion stability. For air and the air-gasoline-vapour-mixture two different valves are used, so the scavenging with air can be done in the exhaust stroke. An example is shown in Figure 11.

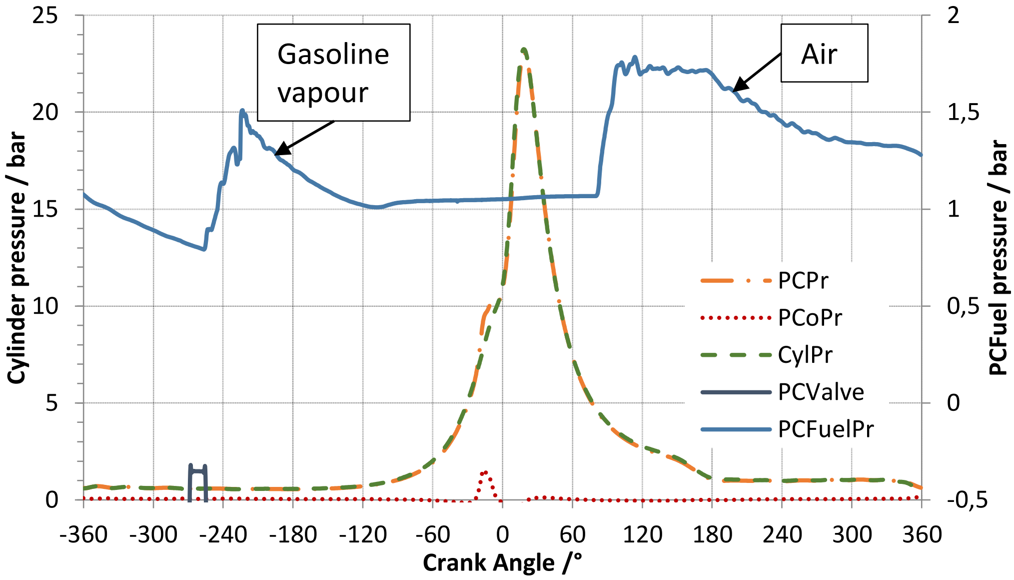

Figure 11: Pre-chamber (PC) at high EGR-rate with additional air scavenging. PCPr: PC pressure; PCoPr: PC overpressure; CylPr: cylinder pressure, PCValve: PC fuel valve duty cycle; PCFuelPr: pressure in PC injection line

For one cycle, the pressure values in the main cylinder, in the pre-chamber and in the pre-chamber fuelling line are plotted. In the pressure of the pre-chamber injection, two pressure peaks are visible. The first one during the intake stroke provides the rich air-gasoline-vapour-mixture, the second one in the exhaust stroke scavenges parts of the burned gases out of the pre-chamber and increases the oxygen concentration in the pre-chamber.

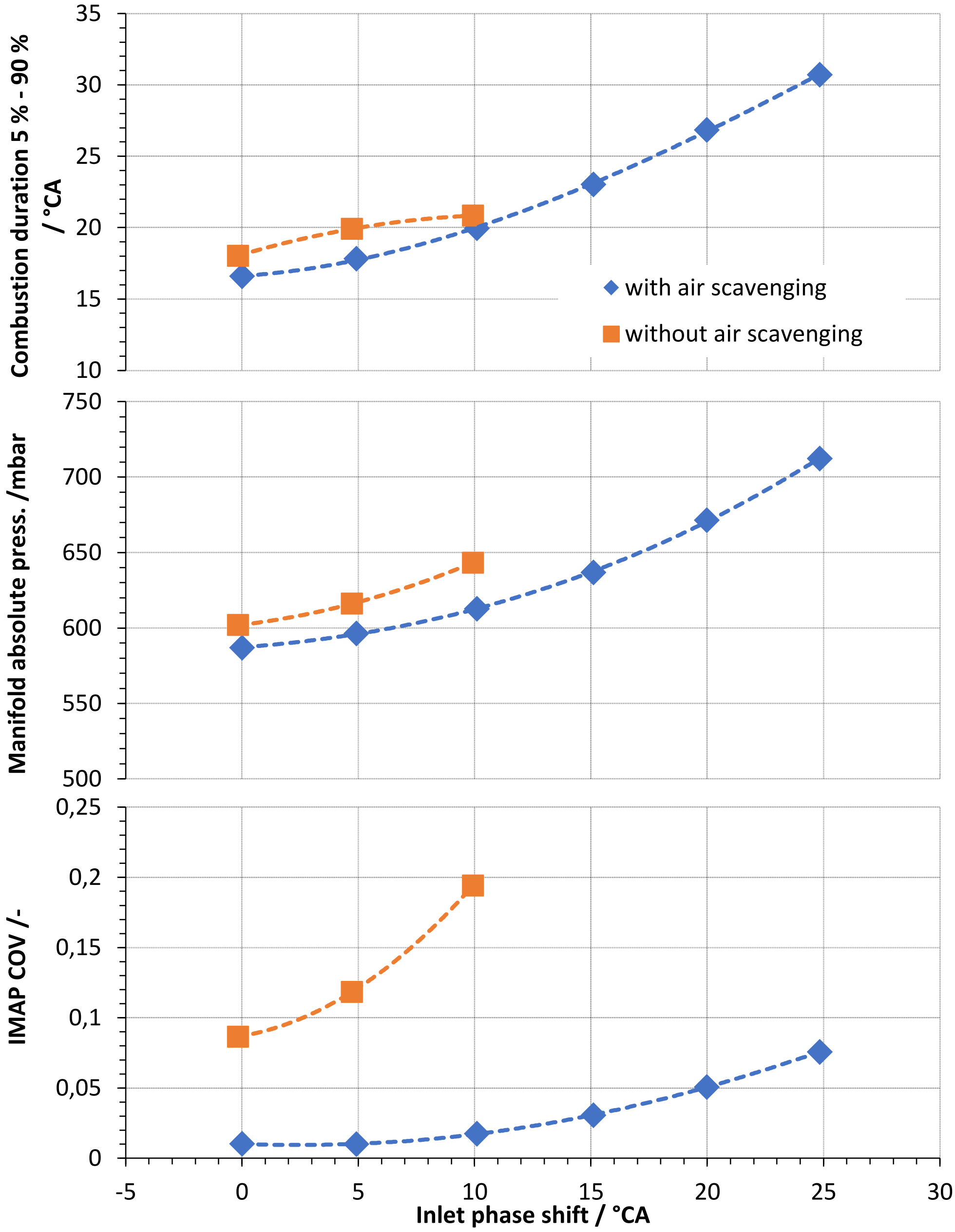

Figure 12: Combustion duration, manifold absolute pressure and combustion stability for the variation of intake phase timing at λ = 1.0, 4.5 bar IMEP and 1500 rpm

In this measurements, only internal EGR trough shift of the intake valve timing was investigated. By moving the inlet phase towards early valve timing, parts of the burned gases are flushed backwards into the manifold and cause the dilution of the cylinder charge. The variation of the intake phase shift is shown in Figure 12 for VK 3 with and without additional scavenging with air. In both measurements the pre-chamber was

supplied with air-gasoline-vapour-mixture at 270°CA before TDC as shown in Figure 11. The operation point was again at 4.5 bar IMEP and 1500 rpm. As the dilution is done by burned gas, the fuel-air ratio is stochiometric. The highest EGR-rate was approx. 20 % according to a 1D-simulation.
It is visible in the lowest diagram in Figure 12, that the pre-chamber has a lower combustion stability at $\lambda = 1$ than in the lean operation points. Over all measured points, the usage of the additional air can significantly increase the combustion stability. The manifold absolute pressure shows the increasing dilution. However, the system suffers from higher combustion durations at high EGR-rates. Further investigations are needed to find better operation parameters for shorter combustion durations at high EGR-rates. Here both the amount and timing of the air scavenging must be optimized together with the amount of gasoline-vapour-air-mixture and the timing of the two solenoid injectors.

4 Conclusions

This paper describes an actively scavenged pre-chamber system as a possibility of high-energy ignition for the homogeneous lean or diluted operation of a passenger car gasoline engine. A gasoline-vapour-air-mixture, as present in car fuel tanks above the liquid phase, is used to supply the pre-chamber. The HC-concentration of this mixture is detected by an HC-sensor and the mixture is actively fed to the pre-chamber via a timing valve controlled by time and quantity. The pre-chamber is designed in such a way that it can replace the spark plug in a current engine without requiring additional installation space. Various pre-chamber geometries and operating states were investigated. It was shown that the best results were achieved with a pre-chamber volume of approx. 2.5 % of the compression volume and a diameter of the transfer orifices of 1.0 mm. This enabled stable operation at $\lambda = 1.85$ with a combustion stability of below 3 % COV. Smaller bores and long bores show a higher ignition delay between the pre-chamber and the main combustion chamber. The investigation of various gasoline-vapour-air-mixture has shown that stable operation with a wide range of HC-concentrations is possible if the flushing parameters are adjusted accordingly. Operation with high residual gas contents can also be enabled with the active pre-chamber if it is actively purged with fresh air after each cycle. The potential of the actively fuelled pre-chamber ignition with a fuel vapour-air mixture from the tank could be demonstrated. For stable operation, however, it is necessary to understand the geometry and operating parameters, as these have a major influence on the performance of the system.

Literature

[1] N. Iida, Research and Development of Super-Lean Burn for High Efficiency SI Engine: Challenge for Innovative Combustion Technology to achieve 50% thermal efficiency, Okayama, Japan, 2017.
[2] J. Hahn, M. Schenk, F.X. Schauer, C. Sauer, G. Weber, C. Schwarz, From glow tube to corona – challenges to the ignition systems of future SI engines, in: J. Liebl, C. Beidl (Eds.), Internationaler Motorenkongress 2017, Springer Fachmedien Wiesbaden, Wiesbaden, 2017, pp. 123–145.
[3] S. Lorenz, Impulskettenzündung mit passiv gütegeschalteten Laserzündkerzen unter motorischen Bedingungen. Dissertation.

[4] W.U. Roessler, A. Muraszew, Evaluation of Prechamber Spark Ignition Engine Concepts: EPA-650/2-75-023, 1975.
[5] J.D. Dale, A.K. Oppenheim, Enhanced Ignition for I. C. Engines with Premixed Gases, in: SAE International Congress and Exposition, SAE International, Warrendale, PA, United States, 1981.
[6] E. Toulson, H.J. Schock, W.P. Attard, A Review of Pre-Chamber Initiated Jet Ignition Combustion Systems: SAE Paper 2010-01-2263, SAE International, Warrendale, PA, 2010.
[7] L.A. Gussak, V.P. Karpov, Y.V. Tikhonov, The Application of Lag-Process in Prechamber Engines: SAE Paper 790692, in: Passenger Car Meeting & Exposition, SAE International, Warrendale, PA, United States, 1979.
[8] L.A. Gussak, M.C. Turkish, D.C. Siegla, High Chemical Activity of Incomplete Combustion Products and a Method of Prechamber Torch Ignition for Avalanche Activation of Combustion in Internal Combustion Engines: SAE Paper 750890, in: SAE Automobile Engineering and Manufacturing Meeting, SAE International, Warrendale, PA, United States, 1975.
[9] S. Yamaguchi, N. Ohiwa, T. Hasegawa, Ignition and burning process in a divided chamber bomb, Combustion and Flame 59 (1985) 177–187.
[10] S. Couet, P. Higelin, B. Moreau, APIR: A New Firing Concept for the Internal Combustion Engines - sensitivity to knock and in-cylinder aerodynamics: SAE Paper 2001-01-1954, in: International Spring Fuels & Lubricants Meeting, SAE International, Warrendale, PA, United States, 2001.
[11] J. Getzlaff, J. Pape, C. Gruenig, D. Kuhnert, R. Latsch, Investigations on Pre-Chamber Spark Plug with Pilot Injection: SAE Paper 2007-01-0479, in: SAE World Congress & Exhibition, SAE International, Warrendale, PA, United States, 2007.
[12] W.P. Attard, H. Blaxill, A Single Fuel Pre-Chamber Jet Ignition Powertrain Achieving High Load, High Efficiency and Near Zero NOx Emissions: SAE Paper 2011-01-2023, SAE International, Warrendale, PA, 2011.
[13] E. Toulson, H.C. Watson, W.P. Attard, The Effects of Hot and Cool EGR with Hydrogen Assisted Jet Ignition: SAE Paper 2007-01-3627, in: Asia Pacific Automotive Engineering Conference, SAE International, Warrendale, PA, United States, 2007.
[14] H.C. Watson, Lawrence J., "A High Efficiency Low Emission Biofuel Heat Engine, Solar '97, Australian and New Zealand Solar Energy Society, Ed. Canberra, A.C.T. 1997.
[15] E. Toulson, H.C. Watson, W.P. Attard, Gas Assisted Jet Ignition of Ultra-Lean LPG in a Spark Ignition Engine: SAE Paper 2009-01-0506, in: SAE World Congress & Exhibition, SAE International, Warrendale, PA, United States, 2009.
[16] A.A. Boretti, H.C. Watson, The lean burn direct injection jet ignition gas engine, International Journal of Hydrogen Energy 34 (2009) 7835–7841.
[17] J. Geiger, S. Pischinger, R. Böwing, H.-J. Koß, J. Thiemann, Ignition Systems for Highly Diluted Mixtures in SI-Engines: SAE Paper 1999-01-0799, in: International Congress & Exposition, SAE International, Warrendale, PA, United States, 1999.
[18] W.P. Attard, P. Parsons, A Normally Aspirated Spark Initiated Combustion System Capable of High Load, High Efficiency and Near Zero NOx Emissions in a Modern Vehicle Powertrain: SAE Paper 2010-01-2196, SAE International, Warrendale, PA, 2010.

[19] W.P. Attard, H. Blaxill, A Gasoline Fueled Pre-Chamber Jet Ignition Combustion System at Unthrottled Conditions: SAE Paper 2012-01-0386, SAE International, Warrendale, PA, 2012.

[20] M. Bunce, H. Blaxill, Sub-200 g/kWh BSFC on a Light Duty Gasoline Engine: SAE Paper 2016-01-0709, in: SAE 2016 World Congress and Exhibition, SAE International, Warrendale, PA, United States, 2016.

[21] M. Bunce, H. Blaxill, Methodology for Combustion Analysis of a Spark Ignition Engine Incorporating a Pre-Chamber Combustor: SAE Paper 2014-01-2603, SAE International, Warrendale, PA, 2014.

[22] P. Chinnathambi, M. Bunce, L. Cruff, RANS Based Multidimensional Modeling of an Ultra-Lean Burn Pre-Chamber Combustion System with Auxiliary Liquid Gasoline Injection: SAE Paper 2015-01-0386, in: SAE 2015 World Congress & Exhibition, SAE International, Warrendale, PA, United States, 2015.

[23] A. Shah, Improving the Efficiency of Gas Engines using Pre-chamber Ignition: Doctoral Thesis (2015).

[24] A. Shah, P. Tunestal, B. Johansson, Effect of Pre-Chamber Volume and Nozzle Diameter on Pre-Chamber Ignition in Heavy Duty Natural Gas Engines: SAE Paper 2015-01-0867, in: SAE 2015 World Congress & Exhibition, SAE International, Warrendale, PA, United States, 2015.

[25] M. Schumacher, M. Wensing, A Gasoline Fuelled Pre-Chamber Ignition System for Homogeneous Lean Combustion Processes, in: SAE 2016 International Powertrains, Fuels & Lubricants Meeting, SAE International, Warrendale, PA, United States, 2016.

[26] K. Na, Y.P. Kim, I. Moon, K.-C. Moon, Chemical composition of major VOC emission sources in the Seoul atmosphere, Chemosphere 55 (2004) 585–594.