

# Investigation of a Methane Scavenged Prechamber for Increased Efficiency of a Lean-Burn Natural Gas Engine for Automotive Applications

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## ABSTRACT

Scarce resources of fossil fuels and increasingly stringent exhaust emission legislation push towards a stronger focus to alternative fuels. Natural gas is considered a promising solution for small engines and passenger cars due to its high availability and low carbon dioxide emissions. Furthermore, natural gas indicates great potential of increased engine efficiency at lean-burn operation. However, the ignition of these lean air/fuel mixtures leads to new challenges, which can be met by fuel scavenged prechambers.

At the Institute of Internal Combustion Engines of the Technische Universitaet Muenchen an air cooled natural gas engine with a single cylinder displacement volume of 0.5 L is equipped with a methane scavenged prechamber for investigations of the combustion process under real engine conditions. The main combustion chamber is supplied with a lean premixed air/fuel mixture. As the excess air restrains a direct ignition by a spark plug, the prechamber must additionally be scavenged with methane to achieve nearly stoichiometric conditions and hence improve the ignition.

To enhance the understanding of the processes occurring, the empirical studies are accompanied by supplementary three-dimensional computational fluid dynamics investigations. The influences of the start of injection and the injection duration are evaluated with regard to the ignition probability and the combustion conditions at the electrodes of the spark plug. Furthermore, the flow and the mixture formation within the prechamber as well as the turbulent kinetic energy are investigated by simulation. The numerical results are validated with experimental data from the test bench regarding parameters such as combustion duration, lean-burn limit and cycle-to-cycle variations.

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## INTRODUCTION

The growing scarceness of fossil fuels and increasingly stringent legislation concerning exhaust gas emissions raise a need for appropriate alternatives. Due to a favorable C/H-ratio and the higher availability of methane a stronger focus is pushed towards these gaseous fuels. In combination with a lean combustion  $\text{CH}_4$  provides the opportunity for high efficiencies. Still, in part-load operation, for methane/air mixtures with high excess air ratios the ignition process becomes more challenging. In this paper, an air cooled natural gas engine with a single cylinder displacement volume of 0.5 L is equipped with a methane scavenged prechamber, one approach to achieve stable combustion. The flow within the prechamber concerning the mixture formation of the methane, the fresh air mixture and residual exhaust gas is investigated using Three-Dimensional Computational Fluid Dynamics (3D-CFD) simulations.

The conclusions drawn from the simulations are validated by test bed measurements and presented in the following sections. Since the orifice diameter is known to have a main impact on the combustion characteristics, variations of their geometry are a major part of this

article. The most suitable configuration is used for further investigations concerning the influences of the scavenging process. Thereby, the impact of the timing as well as the total amount of the scavenging fuel mass are investigated.

The investigations conclude with a comparison of different operation modes like conventional ignition systems, unscavenged and scavenged prechambers for one engine operation point. In this chapter the different setups are evaluated with regard to engine efficiency and combustion characteristics.

## Lean-Burn Combustion

Lean-burn technologies for natural gas engines play a key role in the efforts to reduce engine emissions while simultaneously increasing engine efficiency. The combustion process of lean mixtures can reduce the engine emissions below the given legal limitations without additional catalytic converters. Due to increasing amounts of dilution air in the combustion chamber the maximum temperatures are reduced significantly yielding a strong reduction of nitrogen oxide emissions, cf. Figure 1. Moreover, the higher excess air ratio leads to

greater efficiency in the combustion process, as the isentropic exponent rises. However, the increasing amount of excess air influences the mixture's ignitability and reduces the laminar flame speed, impeding the ignition of the air/fuel mixture and resulting in the lean misfire limit.

One way to avoid unstable combustion and misfire is to increase the speed of flame propagation by enhancing the level of Turbulent Kinetic Energy (TKE) inside the combustion chamber. A different approach is to split the combustion chamber into a main- and a prechamber to improve the combustion characteristics by enlarging the initial flame area in the main combustion chamber and hence increasing the flame propagation speed. With the gas being pushed through small orifices the macroscopic charge motion like tumble or swirl breaks into microscopic motion creating high amounts of turbulent kinetic energy. However, the application of a prechamber means increased constructional effort, extends the surface to volume ratio and leads to higher thermal losses. Nevertheless, a multitude of various prechamber systems has been investigated, due to the remarkable benefits regarding the extension of the lean-burn limit and the increased flame propagation rate.

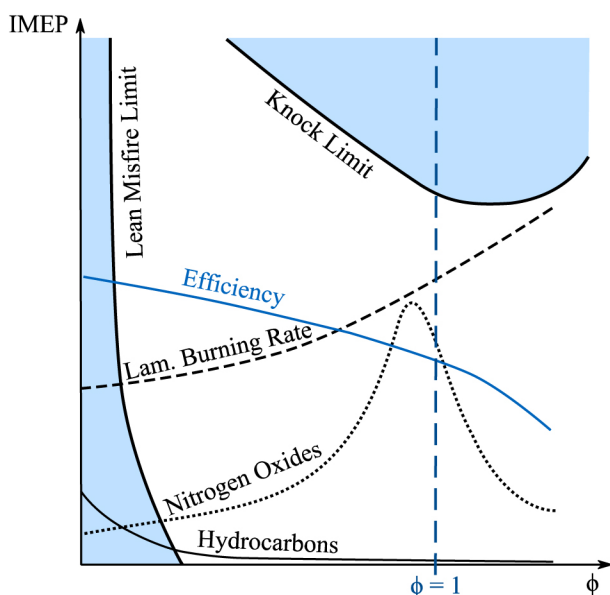


Figure 1. Engine efficiency, emissions and further combustion characteristics in qualitative dependency of the global equivalence ratio.

### State of the Art

Pre-combustion chambers are a state of the art technology for large gas engines. Thus, most of the published reviews focus either on prechamber systems in large gas engines or on specific combustion enhancement systems such as Pulsed Flame Jet, Jet Plume Ignition and Combustion and Auto-inflammation Pilotée par Injection de Radicaux, e.g. [1, 2, 3, 4, 5]. This article contributes to the current prechamber research for small passenger car applications. In this context, the following section gives a short overview of corresponding investigations.

The influence of the prechamber's nozzle Orifice Diameter (OD) and thereby the total cross-sectional area of the nozzle orifices has been investigated several times, for instance in [6, 7, 8, 9]. Thereby, Roethlisberger et al. show in [6] by means of 3D-CFD simulations that a larger orifice diameter mainly reduces the pressure difference between main- and prechamber as well as the respective flow velocity in the nozzle orifices. The increased cross-sectional area lowers the pressure drop between the combustion- and prechamber, raises the mass flow and causes a slightly higher velocity in the upper part of the prechamber. Owing to the decreased velocities through the orifices the turbulence in the entire prechamber reaches lower levels. The influence of the orientation and number of nozzle orifices connecting the prechamber and the main combustion chamber, as well as the prechamber's internal volume and shape is evaluated. The effects of larger nozzle orifice diameters on the combustion characteristics based on actual test bed measurements are shown in [7]. It is stated, that these orifice diameters cause slower combustion combined with a significantly lower rate of heat release. Due to pressure transducers in the main- and prechamber, the decreasing pressure drop was confirmed. Moreover, the minimum reachable air/fuel ratio tends to be leaner for an increasing nozzle orifice diameter. The presented results are important in this context insofar as the basic shape of the prechamber is comparable. Nevertheless, the prechamber system described by Roethlisberger et al. is larger, unscavenged and integrated into the engine's cooling system.

The influence of different nozzle orifice diameters is also investigated by Kawabata et al. in [8]. They mainly focus on optical investigations of the flame jets emerging from the prechamber and the dependencies of the number of nozzle orifices on the flame jet distribution. Thereby, the rate of heat release as well as the maximum cylinder pressure rise with the number and the total cross-sectional area of the nozzle orifices. The impact of the prechamber nozzle orifice diameters is further examined by [5, 9]. Crane et al. focus on a prechamber design, where two divided chambers are connected by an orifice and in turn the lower with the main chamber. During the investigations the orifice diameter is varied. Robinet et al. use the orifice diameter to quench the emerging flame jets and thus to induce active radicals into the main combustion chamber. Hence, the main chamber mixture is ignited as soon as the required conditions are met and multiple flame fronts propagate through the premixed charge. In this specific case, the nozzle orifice diameter varies between 0.5 mm and 0.8 mm (0.0197 in and 0.0315 in).

Recent investigations of the scavenged prechamber system Turbulent Jet Ignition are published by Attard et al. [10, 11, 12, 13]. Due to the comparatively long orifices, the prechamber flame front is extinguished. Therefore, active radicals enter the main chamber and initiate the combustion at a multitude of distributed ignition sites, resulting in a fast rate of heat release. The rapid flame propagation enables stable operation even at increased excess air. Additionally, the advantages of the system depending on type and quality of the fuel are demonstrated.

## ENGINE SPECIFICATIONS / PRECHAMBER DESIGN

### Engine Specifications

The engine used for the experimental investigations is the air cooled two-cylinder Hatz 2G40 diesel engine modified for natural gas operation. The detailed engine specifications are given in [Table 1](#). As only one prechamber system is applied to the engine one cylinder was decommissioned for the presented measurements. The intake channel of the original engine is designed as a swirl port, which results in an swirl charge motion within the cylinder. During previous investigations, different piston geometries have been analyzed concerning their influence on the charge motion and the combustion characteristics. A star-like shape, shown in [Figure 2](#), was chosen for the lean-burn combustion process. The distinct swirl flow pattern is broken up by the inner edges of the piston bowl which leads to an increased turbulence intensity and thus to a shorter Combustion Duration (CD) and higher combustion stability. The piston bowl has been optimized in order to give unhindered flame propagation and a compact geometry to keep the surface area and thereby the heat transfer losses on a comparatively low level. More detailed information on the piston shape design is given in [\[14\]](#).

The engine is controlled by a Trijekt electronic control unit. The actual operating point is defined by operating maps for the manifold port injection, the Ignition Timing (IGN) and the throttle position. The spark plug used for the experiments is a NGK CR8E, with an electrode clearance of 0.7 mm (0.0276 in). As the prechamber can be scavenged with methane, an additional control for the gas injection was realized with a National Instruments LabVIEW code compiled for FPGA usage on a CompactRIO system. It is Crank Angle (CA) based and allows an independent control of the start of injection timing as well as the total current feed time and the crank angle duration, respectively. A power amplifier generates the required signal for the Hitachi DI-injector taken from a gasoline engine. The gas used for all the measurements is methane with a purity of 99.5 %, supplied from an external gas supply.

Table 1. Specifications of the engine.

|                          |                                                  |
|--------------------------|--------------------------------------------------|
| Number of cylinders      | 1                                                |
| Number of intake valves  | 1                                                |
| Number of exhaust valves | 1                                                |
| Bore                     | 92 mm (3.62 in)                                  |
| Stroke                   | 75 mm (2.95 in)                                  |
| Displacement volume      | 498.57 cm <sup>3</sup> (30.425 in <sup>3</sup> ) |
| Connecting rod length    | 129.3 (5.09 in)                                  |
| Compression ratio        | 11:1                                             |
| Exhaust valve opening    | 42° CA before BDC                                |
| Exhaust valve closure    | 8° CA after TDC                                  |
| Intake valve opening     | 5° CA before TDC                                 |
| Intake valve closure     | 44° CA after BDC                                 |

### Prechamber Design and Integration

The prechamber system is designed modularly to facilitate various parameter studies. The assembly of the main parts of the prechamber system is shown in [Figure 2](#) consisting of the prechamber itself (2), a jacket with the required thread for the spark plug (1) and a retainer (not shown). The methane supply system comprises a DI-injector (3) mounted on the prechamber jacket and is controlled independently from the one of the intake manifold.

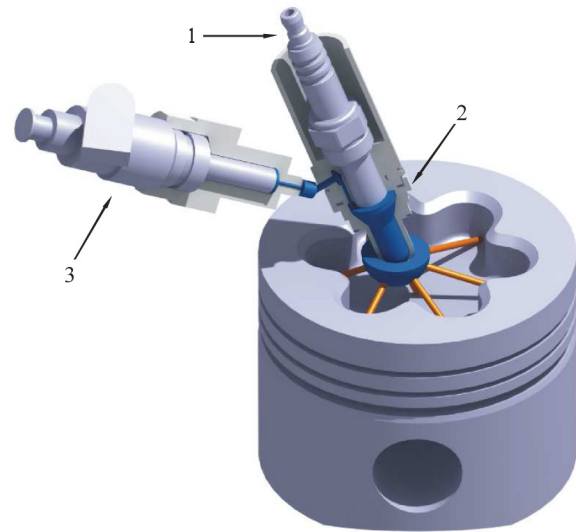


Figure 2. Assembly of the modular prechamber system (1: spark plug; 2: prechamber; 3: DI-injector).

The gas injected into the inlet runner is throttled to 3 bar gas pressure relative to ambient conditions, whereas the gas pressure for prechamber injection is set to 9 bar. The higher pressure allows a wider range for injection timing variation due to the rising back pressure from the compression stroke, as described in the following sections. The total volume of the prechamber is 1745 mm<sup>3</sup> (0.107 in<sup>3</sup>), which corresponds to about 3.5 % of the engine's clearance volume. In addition, [Figure 2](#) shows the volume of the gas canal and the internal prechamber volume as well as the orientation of the prechamber nozzle orifices, which are aligned according to the optimized piston geometry mentioned above.

### Experimental Conditions and Evaluation of Test Results

Relevant experimental conditions are summarized in [Table 2](#). The evaluation of measured engine test conditions is based on a MATLAB data acquisition tool, defining 540° CA as the Bottom Dead Center (BDC) and 720° CA as the firing Top Dead Center (TDC). The pressure signals of intake and exhaust manifold as well as the cylinder pressure trace are recorded with a resolution of 0.1° CA and arithmetically averaged over 50 consecutive engine cycles. These measured values are then used for a detailed combustion analysis to determine the rate of heat release and the burned mass fraction. The combustion duration is defined as the difference between the angular position where 5 % and 95 % of the overall heat has been released.

Table 2. Experimental Conditions used for the prechamber measurements.

|                                |                 |
|--------------------------------|-----------------|
| Crankshaft rotational speed    | 1100 ± 5 rpm    |
| Gas pressure intake (rel.)     | 3 ± 0.05 bar    |
| Gas pressure prechamber (rel.) | 9 ± 0.2 bar     |
| Intake air pressure (abs.)     | 0.88 ± 0.02 bar |

## CONSIDERATIONS FOR 3D-CFD SIMULATIONS

### Mesh Generation and Setup

For the 3D-CFD simulation FIRE by AVL and the provided fuel databases including methane were used. The employed mesh covers the volume of the prechamber between the DI-injector and the nozzle orifices, as can be seen in [Figure 2](#) and in greater detail in [Figure 3](#). The combustion chamber is simplified to a spherical volume representing the main chamber volume adjoining the nozzle orifices. This simplification is justified since there is no distinct charge motion in the combustion chamber due to the piston design and the charge motion in the prechamber mainly depends on the nozzle orifices. Since the CFD simulations focus on the conditions in the prechamber present at a possible ignition timing, combustion is not modeled. As it can be seen in the right part of [Figure 3](#), the mesh is generated using one single custom refinement at the nozzle orifices only. Otherwise, the cell size is automatically adjusted between 0.05 mm (0.00197 in) and 0.5 mm (0.0197 in). Since the orientation of the actual prechamber deviates from the piston by 20° and the mesh generation follows the main axis of the computational domain, the prechamber is rotated by the same angle to obtain a well-structured mesh. The mesh for the main chamber volume shows a significant intersection just before the boundary surface. This intersection is due to a mesh extrusion operation whereby the outermost cell layer is extruded to generate several orthogonal layers of boundary cells. Hence, the performance and stability of the simulations is significantly enhanced. The turbulence is calculated by the  $k-\zeta$ -f model implemented in FIRE. In contrast to the  $k-\epsilon$  model, the near-wall region is treated by an elliptic relaxation model instead of using empirical correction functions meaning the wall boundary conditions are taken into consideration in the modeling approach [15]. Besides, this model is modified to reduce its sensitivity concerning the mesh structure [16].

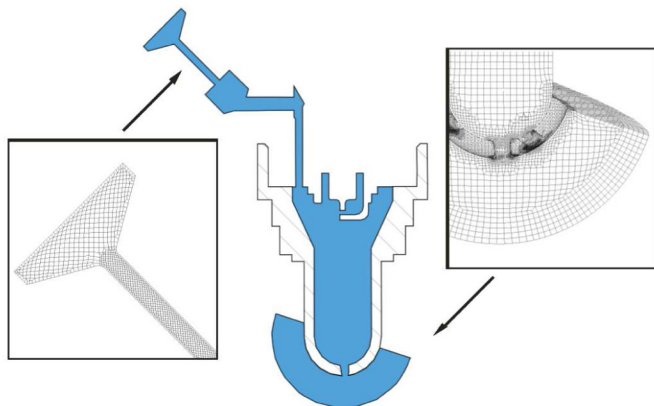


Figure 3. Cut through the computational domain and the generated CFD mesh.

### Initialization of the Gas Canal

Although the gas canal comprises only a small volume its gas composition before the injection is supposed to cause effects. Especially during short injections, a composition consisting of residual gas in the canal will delay the methane flow. To determine the mass fractions of residual exhaust gas (REG), fresh mixture and methane for the subsequent simulations, a pre-simulation is carried out. This simulation starts at the end of a methane injection, initializing the gas canal with 100 % methane and the prechamber volume with  $\phi = 0.556$ . Due to the rising back pressure during compression, the lean mixture is pushed into the gas canal until the ignition point. To consider the conversion of the gas composition during combustion the prechamber is reinitialized with 100 % residual exhaust gas. Resulting from the progressive pressure rise residual gas is pushed into the canal and the present components are compressed in the upper part of the volume. During the expansion stroke parts of the gas blend and leave the canal. The geometry of the connections of the canals lead to dead volumes with different compositions than the rest.

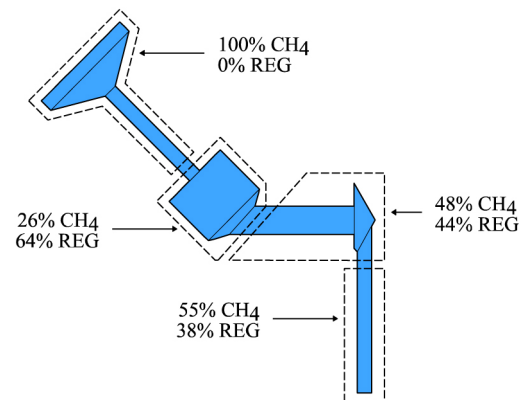


Figure 4. Initialization of the gas canal based on preliminary simulation.

The calculated gas composition regarding methane and residual exhaust gas is averaged for determined sections of the gas canal and defined as initial condition for all the prechamber simulations. [Figure 4](#) illustrates the resulting initial conditions of the gas canal whereby the remaining percentage is initialized with fresh air. Within the volumes resulting from the geometric connection of different pipes, exhaust gas accumulates and cannot escape during the methane scavenging of the canal.

### Initial and Boundary Conditions

Reasonable assumptions based on measured temperatures of the cylinder head had to be made to determine thermal boundary conditions for the simulations. The simulation domain holds one inlet selection at the gas canal and one at the boundary surface of the main chamber volume. The latter is a static pressure condition taken from a reference simulation with the entire main chamber geometry. For the gas canal inlet condition the methane mass flow has been measured using a coriolis flow meter Rheonik RHM 015 under motored engine operation. The results for relative gas pressures of 5 bar and 9 bar are plotted in [Figure 5](#) allowing two significant conclusions. On the one hand, the gas pressure does not only impact the mass flow through the injector at a given injection duration but also influences the