

The Influence of Crevices on Hydrocarbon Emissions from a Diesel-Methane Dual Fuel Engine

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ABSTRACT

Emissions of unburned methane are the Achilles heel of premixed gas engines whether they are spark ignited or diesel pilot ignited. If the engine is operated lean, lower temperatures prevail in the combustion chamber and several of the mechanisms behind the hydrocarbon emissions are aggravated. This paper presents an experimental investigation of the contribution from combustion chamber crevices and quenching to the total hydrocarbon emissions from a diesel-methane dual fuel engine at different operating conditions and air excess ratios.

It is shown that the sensitivity to a change in top-land crevice volume is greater at lean conditions than at stoichiometry. More than 70% of hydrocarbon emissions at air excess ratios relevant to operation of lean burn engines can be attributed to crevices.

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INTRODUCTION

Gas engines are projected to increase in number due to favorable fuel price and environmental concerns. From an efficiency point of view it is desirable to operate gas engines lean. There are several reasons for this; lean operation results in less throttling losses, less heat transfer to the coolant and a higher ratio of specific heats. The sensitivity to knock is also reduced and for this reason a higher compression ratio can be permitted.

The drawbacks of lean operation include risk of misfire, lower flame speed and lower power density. In addition to these factors, the emissions of unburned hydrocarbons, HC, increase. The diesel dual fuel, DDF, engine, which uses a small diesel pilot injection instead of a spark plug successfully addresses the first two issues; the powerful ignition source eliminates misfire and the ensuing combustion is rapid. However, the issue of HC emissions remains.

The EURO VI emissions legislation stipulates a limit of 0.5 g/kWh of CH₄ for SI gas engines. As of today it is not possible to certify diesel dual fuel engines in the EU but it is reasonable to assume that the same limit for CH₄ will apply.

The following conditions shape the demands of future dual fuel engines with regards to unburned hydrocarbons:

- A legislated limit of 0.5 g/kWh CH₄
- The typical conversion efficiency for a methane oxidation catalyst is 90%. Catalyst ageing degrades conversion efficiency during vehicle life.
- New in-use compliance testing requires fulfillment of legislation for aged vehicles.

Considering the above, a reasonable limit for engine-out CH₄ emissions is 4 g/kWh. As seen in [Figure 1](#) HC emissions increase monotonically with λ and diesel substitution.

In order to stay below the limit of 4 g/kWh the option is to either run at relatively low λ or to reduce the diesel substitution rate; neither of which is desirable from an economic point of view. For this reason it is of great importance to understand the sources of HC emissions at lean conditions so that proper countermeasures can be taken.

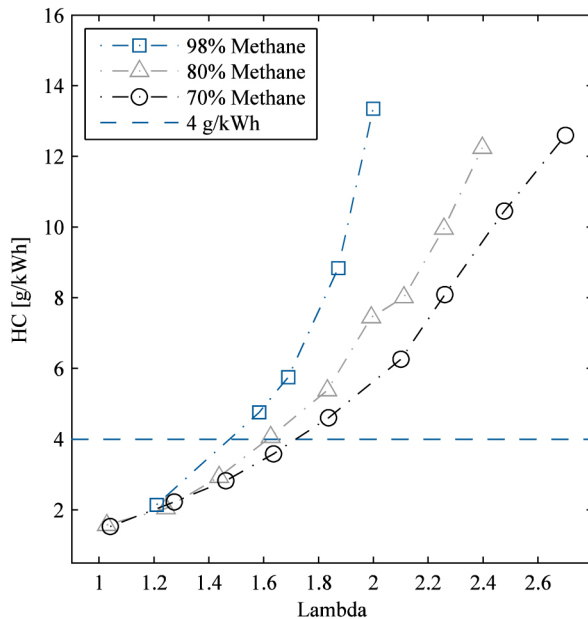


Figure 1. HC emissions versus λ for three different diesel substitution rates [1].

For SI gasoline engines, 6 sources of HC are identified [2].

1. Combustion chamber crevices
2. Absorption and desorption of fuel in oil
3. Absorption and desorption of fuel in deposits
4. Flame quenching, both wall and bulk
5. Liquid fuel in the cylinder, bad preparation
6. Exhaust valve leakage

Of these six mechanisms, only some are applicable to DDF operation. Mechanism two does not contribute to HC emissions since methane is practically insoluble in oil [3]. Absorption in deposits, mechanism three, is considered small but cannot be ruled out. Quenching in parts of the bulk charge cannot be allowed for reasons of emissions, drivability and efficiency and must be addressed by appropriate combustion and control strategies. The contribution from wall quenching to HC emissions is considered small since most of the fuel diffuses out of the boundary layer and into the burned charge and oxidizes [4]. However, since the dual fuel engine is operated mostly at lean conditions and methane is resistant to oxidation, the fuel that diffuses into the burned zone is subject to lower temperatures and might escape oxidation. Hence the contribution from mechanism four is unknown. Mechanisms five and six are ruled out since methane is a gaseous fuel and since exhaust valve leakage should not be a factor in a modern engine in good condition.

HC emissions from diesel engines are discussed in [4] and three sources are identified:

1. Overmixing of fuel and air during ignition delay

2. Overriching of the mixture beyond stoichiometric

3. Fuel escaping from nozzle sac during expansion

Mechanism one is ruled out during dual fuel operation since the diesel is injected into a homogenous mixture of methane and air and no overly lean zones should exist. Mechanism two should not be an issue, since any rich zones would be in or around the diesel pilot. These zones burn during the early part of combustion and are subjected to the highest temperatures, so as long as the global mixture is lean HC from mechanism two should oxidize. The remaining mechanism, number three, occurs in the DDF engine but since the HC from the nozzle sac consists of diesel, which is much easier to oxidize in a catalyst, this is not considered a major problem. Hence, mechanisms contributing to HC emissions from DDF operation are, listed in perceived order of importance:

1. Combustion chamber crevices
2. Flame quenching, both wall and bulk
3. Absorption and desorption in deposits
4. Fuel escaping from nozzle sac during expansion

Of these four, category one is considered most influential followed by category two. Category three is likely small and the HC from category four is not methane but diesel and therefore presents a much lesser problem.

Hydrocarbon Emissions from Crevices

Crevices are narrow regions of the combustion chamber where heat transfer to the surrounding material prevents flame propagation. In an SI engine, the most influential crevices and their approximate contribution to the total hydrocarbons from crevices, crevice-HC, are [4]:

- Ring pack crevices, the volume in top-land and second land above the first and second compression rings. These account for 80% of crevice-HC.
- Head gasket crevice, 13% of crevice-HC.
- Spark plug threads and volume around center electrode, 5% of crevice-HC.
- Other, including valve seat crevices, 2%.

In the HD test engine used in this project; the cylinder head seals tightly against the cylinder liner and therefore no head gasket crevice exists. Naturally the engine also lacks a spark plug but the slit around the diesel injector tip likely contributes in a similar manner. For this reason the ring pack crevices, and most importantly the top-land volume, are considered most influential in dual fuel engines.

As the flame front propagates outwards, the hot, expanding gases behind the flame force an increasing amount of unburned mixture into the crevices. The pressure is uniform between the crevices and the combustion chamber but because of much lower temperature in the crevices the

density is higher. The gas temperature in the crevice is assumed to be 450 K, which is the mean of the piston temperature and liner temperature. Piston and liner temperature were acquired from engine simulations in a commercial simulation software. Since the temperatures acquired from simulation can not be expected to be entirely accurate and to reduce complexity the temperatures from one representative operating point were used to analyze all cases in this publication. For the test engine in question, based on the ideal gas law and a topland volume corresponding to 2% of the compression volume, 10% of the charge is forced into the topland crevice at maximum in cylinder temperature and $\lambda = 1$. At leaner mixtures, the temperature in the combustion chamber is lower and less temperature difference between bulk gas and crevices exist, at $\lambda = 2$ a maximum of 6% of the charge is forced into the crevice. This is shown in Figure 2.

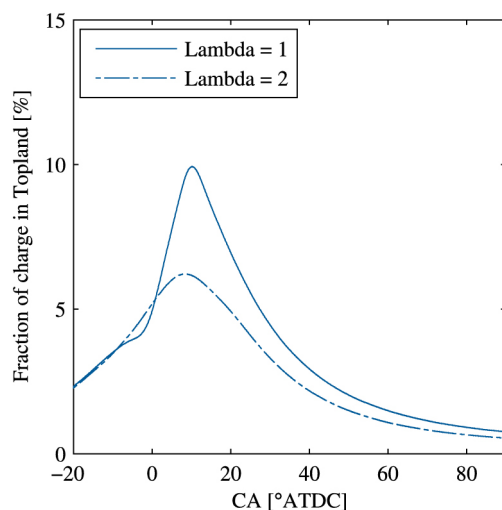


Figure 2. Fraction of charge forced into topland crevice versus crank angle for two different air excess ratios.

In spite of less HC forced into the crevices, the measured engine-out HC emissions are higher by a factor of five for the case at $\lambda = 2$. This shows that the fate of the hydrocarbons as they reemerge into the combustion chamber from the crevices is of greater importance than the maximum amount trapped.

As the expansion progresses and the pressure in the combustion chamber start to drop, HC from the crevices are reintroduced. It is shown in [2] that the crevice gas exits the crevice at a lower speed compared to the piston and hence is deposited in a thin layer along the cylinder liner. If the temperature is sufficiently high and oxygen is present, the HC can oxidize. For stoichiometric SI engines running on gasoline, the temperatures during the closed part of the cycle are sufficient so that emerging HC oxidizes to a great extent while HC reintroduced after exhaust valve opening, EVO, escapes into the exhaust [5], [6]. In the DDF engine, three mechanisms aggravate the problem of crevice-HC:

- Methane is one order of magnitude slower to oxidize than gasoline-like fuels at relevant temperatures (1000-1200K) [3].

- Lean operation lowers the temperature in the combustion chamber, potentially stopping oxidation of HC from crevices before EVO [6] and causing a larger amount of crevice-HC to reach the exhaust.

- Lean operation lowers the temperature close to the cylinder wall. This is likely to cause a reduction in the oxidation of hydrocarbons from the topland crevice deposited along the liner during the expansion stroke.

Combined with the difficulty to oxidize methane in a catalyst, crevice-HC is a challenge which must be given sufficient attention if stringent emissions legislation is to be fulfilled. A substantial amount of research has been done in this field regarding Si-operation for conditions close to stoichiometry. For DDF operation, however, where lean combustion of methane is combined with a diesel combustion system with large crevices, no previous work has been found which investigates the impact of combustion chamber crevices.

OBJECTIVES

The objectives of this research is to determine the contribution from combustion chamber crevices to the total amount of hydrocarbon emissions for operating conditions relevant to lean burn gas engines.

EXPERIMENTAL SETUP

Engine

The measurements have been carried out on a single cylinder Scania HD lab engine available at the Royal Institute of Technology in Stockholm, Sweden.

The engine has a displacement of 2 l and is equipped with a high pressure common rail system, Scania XPI, as well as gas injectors from Keihin, placed in the intake runners. The rail-pressure for the methane was maintained at 2.8 bars above intake pressure by an automatic pressure regulator. The piston and piston crown are from Scania's current line of Euro V diesel engines. Further specification of the engine is located in Table 1

Table 1. Engine specifications

Feature	Value
Displacement [cm ³]	1950
Bore [mm]	127
Stroke [mm]	154
Connecting rod length [mm]	255
Geometric compression ratio	17.3
IVO[°ATDC]	346
IVC[°ATDC]	-154
EVO[°ATDC]	145
EVC[°ATDC]	355

Auxiliary systems are in place for heating or cooling of intake air, oil, fuel and water.

Piston Modifications

In order to provoke a response and investigate the sensitivity of HC emissions to the volume of the top-land crevice, a set of pistons with known increases in top-land volume were manufactured. A circumferential cut was made in the top-land as illustrated in [Figure 3](#).

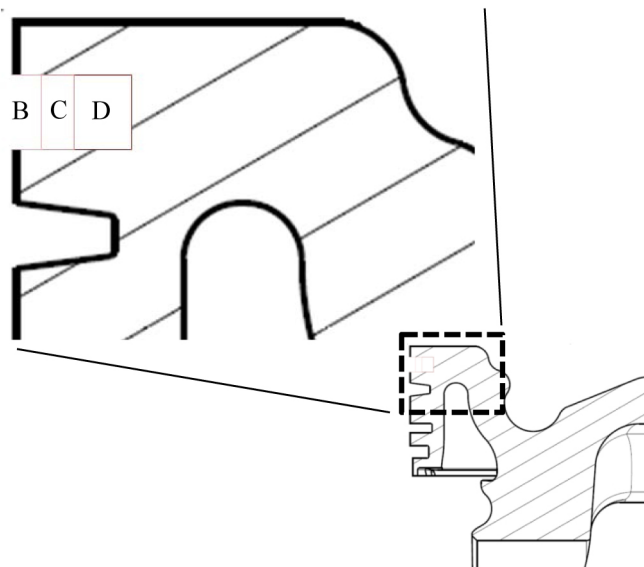


Figure 3. Sketch of piston top-land illustrating the modifications.

The cut was made in three depths on three separate pistons resulting in the top-land crevice volumes listed in [Table 2](#).

Table 2. Piston modifications

<i>Piston version</i>	<i>Hot top-land volume[cm³]</i>	<i>Top-land volume relative to standard[%]</i>
A(Standard)	4.1	-
B	5.75	140
C	7.65	186
D	10.5	256
E	1.83	44

In addition to the modifications made to the piston top-land volume, a fourth piston, piston E, was machined with a 45 degree chamfer around of edge as seen in [Figure 4](#).

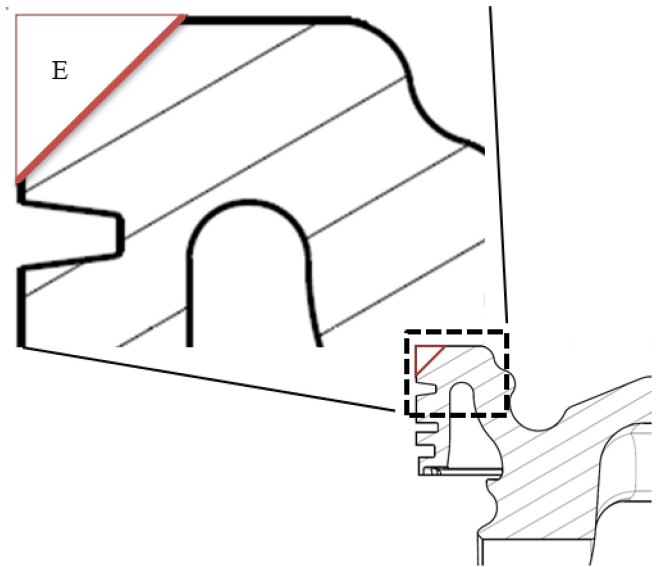


Figure 4. Sketch of piston top showing the circumferential chamfer of the piston edge.

The amount of material removed from piston E was sufficient to lower the compression ratio of the engine from 17.3 to approximately 16 but none of the other modifications were large enough to significantly affect the compression ratio. This is shown in [Figure 5](#).

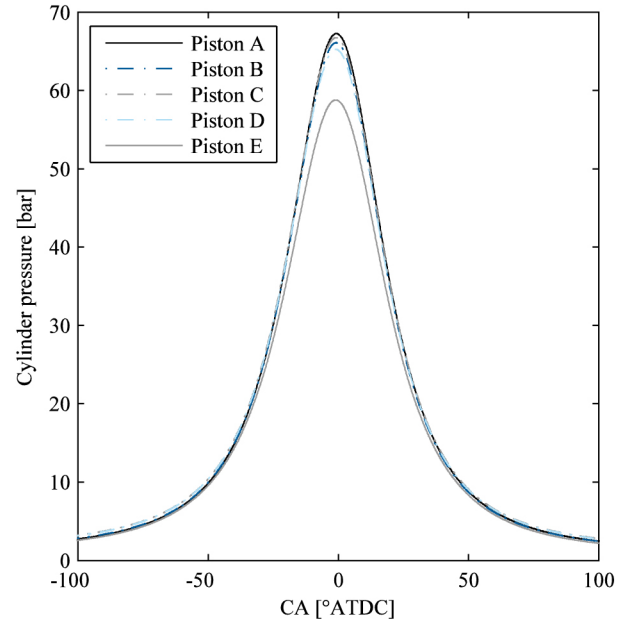


Figure 5. Motored cylinder pressure versus crank angle for different top-land crevice volumes.

Control System

Both data acquisition and communication with the test bed were performed using an in-house software; which runs on a PC and is supported by several PIC-processors.

For direct control of the injection system, and indirect control of the test bed, the AVL Rapid Prototyping Controller was used. The Rapid Prototyping Controller is an open source rapid prototyping system for engine control which is based on Simulink models and executed on dSPACE hardware [7].

Data Acquisition

Emissions of NO_x , HC and CO and the EGR level were measured using a Horiba EXSA-1500 exhaust analyzer. Smoke was measured with an AVL 415S smoke meter. For fuel metering, an AVL 733 fuel balance was used for the liquid fuel and an Alicat Scientific 500 SLPM-type gas flow meter for the methane.

The pressure history was recorded at 0.1 CA increments, using an AVL QC32D pressure transducer and a Kistler 5011 charge amplifier.

Crank Angle Resolved Measurements of Hydrocarbon Emissions

Crank angle resolved hydrocarbon emissions were acquired using a Combustion HFR400 Fast FID. The sample point was located in the exhaust runner 15 cm downstream of the exhaust valves. A typical output from the FastFID including error bars corresponding to one standard deviation is shown in Figure 6.

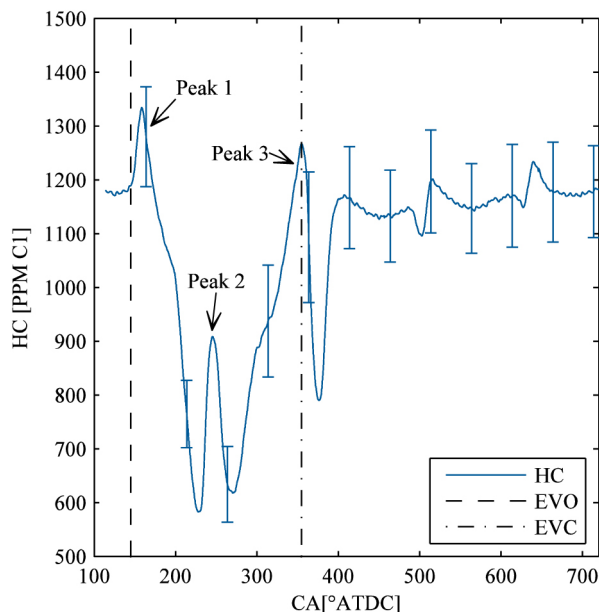


Figure 6. HC emissions versus crank angle, averaged over 100 cycles, sampled in the exhaust runner 15 cm downstream of valves.

The appearance of the curve differs depending on the sample point. When sampling close to the exhaust valve the measurements are sensitive to the radial placement of the probe [8], [9], but at distances in excess of approximately 5.5 cm no radial gradients remain, [10]. The output from the FastFID also depends on the axial placement of the probe. The measurements presented in this publication have a similar appearance to measurements acquired at similar distance from the exhaust valves by other authors [11], [10], [12].

The FastFID is equipped with a constant pressure chamber which ideally removes the influence of changes in exhaust pressure. However, rapid pressure fluctuations can still affect the measurements. This is seen between 450 and 700°CA in Figure 6; there are standing waves in the exhaust pipe which cause slight increases in the measured concentration of HC.

While the utilization of fast hydrocarbon analyzers has been described by several authors, there is not complete consensus with regards to how the results should be interpreted. There are three distinct peaks seen on the curve. Suggestions for the interpretation of peak 1:

- HC from end of previous cycle residing in the port, [3], [12], [13], [14].
- HC from quench layers and crevices close to exhaust valve, [3], [11], [10], [15], [12], [16], [17], [18].

Considering the pressure sensitivity of the FastFID discussed previously a plausible third explanation for the first peak can be postulated; a combination of HC from the previous cycle intensified by the pressure increase from the blowdown pulse.

The second peak is more controversial; some authors claim that it is influenced by changes to the ring pack [3]. The second peak would be a result of turbulent release of HC trapped in second land. However, measurements carried out during the work presented in this paper, where the ring gap of the second piston ring was increased show no influence on the second peak making this theory unlikely.

Others authors postulate that the reverse flow which can occur immediately following the blowdown pulse causes some of the HC from peak 1 to be measured twice [11], [10], [18]. This theory has further support as [13] show a correlation between the height of the second peak and the level of HC present in the port before EVO.

Regarding the third peak, there seems to be a general agreement that it is the result of HC from piston crevices. These hydrocarbons have been deposited along the liner by the downward motion of the piston during the expansion stroke and are subsequently collected again in a vortex at the edge of the piston during the exhaust stroke [2] [3]. This is also supported by the tests with increased second ring ring-gap previously mentioned. Increasing the ring-gap, causing HC in second land to flow into the crank case to a greater