

The Influence of In-Cylinder Flows on Emissions and Heat Transfer from Methane-Diesel Dual Fuel Combustion

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ABSTRACT

In order for premixed methane diesel dual fuel engines to meet current and future legislation, the emissions of unburned hydrocarbons must be reduced while high efficiency and high methane utilization is maintained.

This paper presents an experimental investigation into the effects of in cylinder air motion, swirl and tumble, on the emissions, heat transfer and combustion characteristics of dual fuel combustion at different air excess ratios. Measurements have been carried out on a single cylinder engine equipped with a fully variable valve train, Lotus AVT. By applying different valve lift profiles for the intake valves, the swirl was varied between 0.5 and 6.5 at BDC and the tumble between 0.5 and 4 at BDC. A commercial 1D engine simulation tool was used to calculate swirl number and tumble for the different valve profiles. Input data for the simulation software was generated using a steady-state flow rig with honeycomb torque measurements. To measure heat transfer, thermocouples were fitted in the cylinder head and heat exchangers on the coolant circuit and the engine oil.

The study shows that swirl has a strong effect on the heat transfer; increasing the swirl from 0.5 to 6.5 increases the heat transfer to the coolant by 50%. With regards to emissions; swirl has the effect of increasing oxidation of hydrocarbons returning from crevices. For this reason a 20% reduction of hydrocarbon emissions can be achieved by increasing the swirl from 0.4 to 3. At high λ of 1.9, combustion is very sensitive to mixing between the gas and the air. The mixing is affected by the turbulence generated over the intake valves. A difference in engine out HC emissions by a factor of two can be achieved by varying the valve lift curve and hence varying the turbulence generated during the intake event. The timing of the gas injection can also improve mixing and achieve similar results. Compared to SI, dual fuel combustion is relatively insensitive to tumble.

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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 way that DDF is usually applied, is that a homogenous mixture of methane and air is inducted into the cylinder and compressed. Late during the compression stroke a small pilot injection of diesel fuel is direct injected and upon self-ignition provides an extended ignition source for the methane. DDF heat release typically consists of three parts [1]:

1. Combustion of the diesel pilot
2. Combustion of methane in and in the immediate vicinity of the diesel spray
3. Flame propagation through the methane-air mixture

Depending on load and a number of parameters, each of these parts will contribute to a different degree to the accumulated heat release. The term diesel substitution rate, or $E\% \text{ CH}_4$, is used to quantify the contribution from each fuel to the total amount of energy supplied. 0% diesel substitution means pure diesel operation and 98% diesel substitution means that the engine is operating almost entirely on methane.

The Euro VI emissions legislation stipulates a limit of 0.5 g/kWh of CH_4 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 CH_4 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 CH_4 .
- The maximum conversion efficiency for a commercial methane oxidation catalyst is approximately 90% [2].
- New in-use compliance testing requires fulfillment of legislation for aged vehicles.

Considering the above, a reasonable limit for engine-out CH_4 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 reduce the emissions of unburned hydrocarbons at high λ and high rates of methane utilization.

In-cylinder air motion is currently used in both SI and CI engines to promote efficient combustion and reduce emissions formation. Complex in-cylinder flows are commonly reduced to two circular bulk motions, swirl and tumble.

Swirl can be measured in swirl number, SN, and tumble in tumble number, TN [4]. As the compression stroke progresses, the geometry of the in-cylinder volume flattens close to TDC, causing the tumble to break down into small scale turbulence which is known to improve combustion speed in SI engines [5] [6].

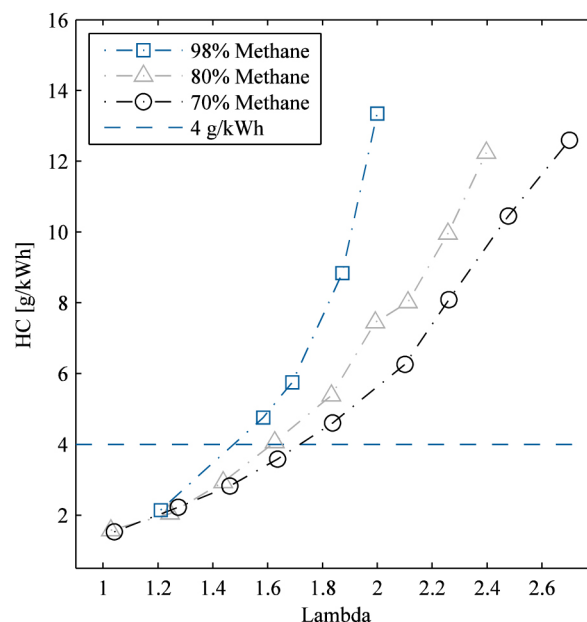


Figure 1. HC emissions versus λ for three different diesel substitution rates [3].

The swirl is forced into the smaller radius of the piston bowl and conservation of momentum causes the speed to increase close to TDC. The swirl survives both compression and combustion and is used in diesel engines mainly to promote after oxidation of soot [7] [8]. For this reason; port designs which enable swirl on diesel engines have historically been of great importance [9].

At high rates of methane utilization in a dual fuel engine, combustion consists mainly of flame propagation and can therefore be expected to respond to changes in tumble motion. As with all premixed engines, air motion will affect the mixing between methane and air. In addition, both the combustion speed and the emissions formation are greatly affected by the distribution of the diesel pilot in the combustion chamber. This in turn is affected by the swirl number. Hence both swirl and tumble can be expected to influence dual fuel combustion in ways that are difficult to anticipate.

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.

Variable Valve Train

For these tests the engine was fitted with cylinder head incorporating a fully variable valve train from Lotus Engineering [10]. Variable valve trains are commonly used in light duty engines, but have not yet made their way into production heavy duty diesels [11]. However, research and development is in progress [12]. The Lotus system allows for different valve lift profiles and hence enables control of the air motion in the cylinder. The cylinder head had special maskings installed in the intake valve seats to increase the maximum swirl number, SN. The SN could then be controlled between 0.4 and the maximum value of approximately 7. This was achieved by varying the lift height or profile of the valves or deactivating one valve entirely. The deactivation of one valve typically results in an increase of the SN by a factor 2.

Simulations

The airflow parameters, swirl and tumble were evaluated using measurements performed in a steady-state flow rig. A 1D engine simulation program, GT-POWER v7.1, was used to calculate the resulting flow parameters for the different valve profiles tested in the single cylinder engine. Some of the valve profiles used are shown in [Figure 2](#).

STD denotes standard valve profile and TPZ trapezoid valve profile. The method used by the 1D simulation tool for calculating swirl differs slightly from the commonly used Thien-method [13]. See [14], [15] and [16] for further details. The predictions of tumble are more approximative and the values for TN presented in this paper should therefore be interpreted with care. For more detailed information regarding the simulations, the experimental setup and the methodology, see [4].

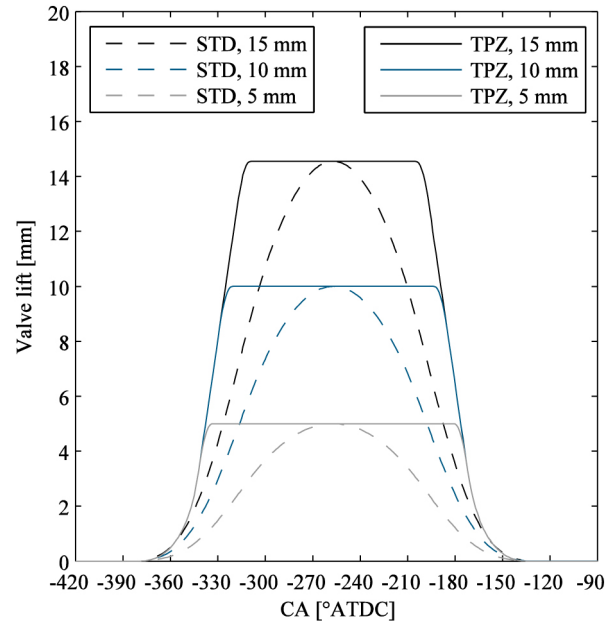


Figure 2. Lift height versus crank angle for some of the valve profiles used in the tests. The STD, 15 mm profile is the standard valve profile.

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 Micro Soot Sensor and 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 GH12D pressure transducer and a Kistler 5011 charge amplifier.

Heat Transfer Instrumentation

Additionally four thermocouples were installed in the cylinder head. Three of the thermocouples were of a conventional type, installed at approximately 2 mm depth. These were used to record a time averaged steady-state temperature. Two of these were installed in the bridge between the exhaust valves which is a critical location with regards to thermal loading. A special rapid response type J surface thermocouple was also used. The surface thermocouple was flush mounted, allowing direct contact between the hot gases and the surface of the thermocouple. This setup enabled a crank angle resolved measurement of the temperature. The thermocouple has similar properties to the cylinder head so the measured temperature should be representative of the cylinder head surface temperature.

The engine was also fitted with heat exchangers on the oil and coolant circuit. By controlling the coolant flow through the heat exchangers a large temperature difference between the coolant entry and exit point could be achieved. The

entering coolant was at room temperature. The flow of coolant was measured using a Sartorius balance. The flow multiplied by the temperature difference across the heat exchangers enables accurate calculation of the steady state heat transfer power.

Piston Modifications

In order to investigate the sensitivity of HC emissions to the volume of the top-land crevice, a piston with a known increase in top-land volume was manufactured. A circumferential cut was made in the top-land, increasing the volume to approximately 186% relative to the standard piston, see Table 2.

Table 2. Modification to crevice volume. Volumes calculated for piston at operating temperature.

Piston version	Hot top-land volume[cm ³]	Top-land volume relative to standard
A(Standard)	4.1	-
B	7.65	186 %

This modification did not significantly change the compression ratio [17]. A sketch of the modification is shown in Figure 3.

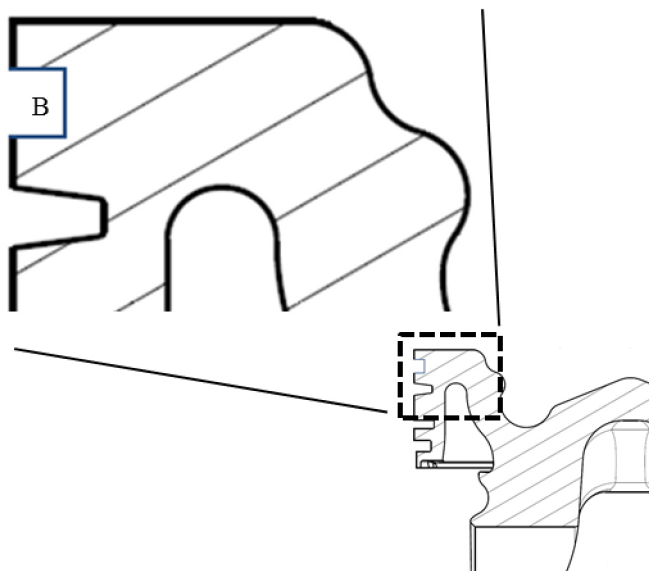


Figure 3. Sketch of piston top-land illustrating the modification. The cut is marked with B.

Test Matrix

The equipment previously described was used to create 32 cases of varying swirl and tumble; these are shown in Figure 4.

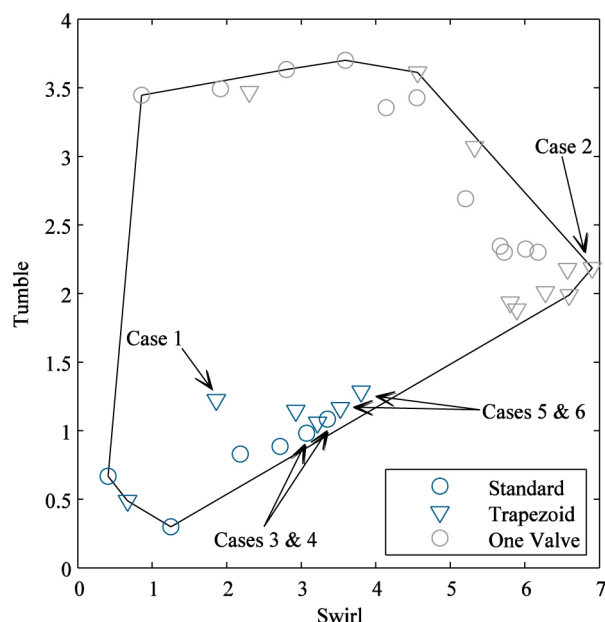


Figure 4. 32 cases with varying swirl and tumble. The two valve cases are shown in blue and the cases utilizing only one valve in grey.

Three different valve strategies were used, standard profile, trapezoid profile and deactivation of one valve. Deactivation was used in combination with both STD and TPZ valve profiles. Six cases which will be examined in further detail are also marked in Figure 4.

These 32 cases were run for three different operating conditions shown in Table 3.

Table 3. Operating points 1,2 and 3

Operating point	1	2	3
Speed [RPM]	1225	1225	1225
IMEP [bar]	7	7	7
Total λ	1.7	1.7	1.9
E% CH ₄ [%]	95	95	95
Piston	A	B(186%)	A
Common rail pressure [bar]	1200	1200	1200
SOI [°BTDC]	10	10	12
CH ₄ SOI [°BTDC]	350	350	350
Intake pressure, absolute [bar]	1-1.4	1-1.4	1.2-1.7
Intake temperature [°C]	30	30	30
Exhaust pressure, absolute [bar]	1	1	1