

Ethanol-Diesel Fumigation in a Multi-Cylinder Engine

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ABSTRACT

Fumigation was studied in a 12 L six-cylinder heavy-duty engine. Port-injected ethanol was ignited with a small amount of diesel injected into the cylinder. The setup left much freedom for influencing the combustion process, and the aim of this study was to find operation modes that result in a combustion resembling that of a homogeneous charge compression ignition (HCCI) engine with high efficiency and low NO_x emissions. Igniting the ethanol-air mixture using direct-injected diesel has attractive properties compared to traditional HCCI operation where the ethanol is ignited by pressure alone. No preheating of the mixture is required, and the amount of diesel injected can be used to control the heat release rate. The two fuel injection systems provide a larger flexibility in extending the HCCI operating range to low and high loads.

It was shown that cylinder-to-cylinder variations present a challenge for this type of combustion. By using closed-loop cylinder-individual control of pressure derivatives and IMEP with the amounts of fuels injected, combustion was successfully harmonized between the cylinders.

Successful fumigation operation was verified up to 18.4 bar BMEP at a fixed engine speed of 1450 rpm. Two load points (4.6 bar BMEP and 9.2 bar BMEP) were studied in detail. Different diesel injection timings, diesel ratios, and EGR rates were investigated, and comparisons were drawn to pure diesel operation of the same engine. At medium load (9.2 bar BMEP), it was possible to obtain a stable HCCI-like combustion with low NO_x emissions (0.1 g/kWh), reasonably high brake efficiency (38 %), and low pressure derivatives (5 bar/CAD). High load operation (18.4 bar BMEP) resulted in low pressure derivatives (5.5 bar/CAD), acceptable brake efficiency (38 %), and relatively low NO_x emissions (0.34 g/kWh).

INTRODUCTION

A variety of engine concepts have been presented as alternatives to the traditional SI and CI engines in order to combine high efficiency with low emissions of NO_x and particulate matter, thus complying with current and future emission legislation. Many of these concepts fall into the category of homogeneous charge compression ignition (HCCI) combustion. Various methods of achieving HCCI have been suggested, differing with respect to fuels used and injection location. Each of these methods have their respective advantages and drawbacks.

With port fuel injection (PFI) of high octane number fuels [1, 2, 3, 4], e.g. gasoline, it is easy to obtain the homogeneous mixture of fuel and air but there are problems controlling the combustion timing, and intake preheating of the air or high levels of internal residuals are required to obtain sufficient temperatures for the mixture to ignite. Studies of PFI of low octane number fuels [5, 6], e.g. diesel, report problems of premature ignition and knocking and the need of preheating the intake charge to high temperatures to avoid accumulation of liquid fuel on intake system surfaces.

Direct injection (DI) of the fuel is a more viable option for diesel HCCI, and different schemes have been suggested [7, 8]. The main problem with this approach is to obtain a homogeneous mixture at the time of ignition. At higher loads, there may not be enough time for both fuel injection and mixing with air prior to ignition.

With the challenges reported using both port fuel injection and direct injection in mind, it is interesting to examine what could be achieved using a combination of the two systems. What is ultimately desired is a system that enables HCCI-like combustion over a large operating range

with low emissions and acceptable efficiency. The system should ensure a stable operation with moderate peak pressures and pressure derivatives so that the noise level is kept low. It is also essential to be able to switch to conventional SI or CI combustion outside the HCCI operating range.

Fumigation is a form of dual-fuel engine operation where part of the fuel is premixed with the intake air, and part of the fuel is injected directly into the cylinder as in a traditional diesel engine. The premixed and direct-injected fuels may be of the same kind, or fuels with different properties may be used to achieve the desired mode of combustion.

Various motivations to study fumigation have been presented. With low levels of premixed fuel, operation is close to that of a diesel engine. In this case, the introduction of the premixed fuel can reduce the ignition delay and pressure oscillations, ensuring a smoother operation at high loads [9]. With diesel used for both the premixed and direct injected fuel, fumigation is a way of avoiding the knock problem associated with pure PFI diesel HCCI [10, 11]. With high levels of premixed high-octane number fuel, engine operation is close to that of a spark ignition engine with the direct injected fuel replacing the spark [12, 13]. With biogas as the premixed fuel and biodiesel as the direct injected fuel, this method has been presented as an engine concept for alternative fuel operation [14].

The approach based on igniting the port injected fuel using a small injection of fuel directly into the cylinder is pursued here. Ethanol, with high resistance to auto-ignition, is injected into the intake port. The premixed ethanol is then ignited by injecting diesel into the cylinder. This way, no preheating is required in order for the ethanol to autoignite. Since the two fuels have different autoignition properties and enter into the cylinder at different stages of the engine cycle, this setup leaves much freedom for influencing the combustion process. The engine is run unthrottled and with the EGR controlled manually.

Ideally, a combustion process similar to that of HCCI operation is desired. Such operation requires that the majority of the fuel be mixed with air prior to combustion, meaning that rapid mixing of the diesel fuel with the ethanol-air charge is needed. A challenge in making the HCCI engine feasible is the control of combustion phasing. One approach that has been presented to this control problem is dual fuel actuation [4]. The ratio of two fuels, both port-injected, is here varied to achieve the desired combustion phasing.

The fumigation setup used in this work has some similarities to dual-fuel HCCI operation. By increasing the ratio of fuel energy coming from diesel, combustion becomes faster. The fumigation setup also presents additional features and complications. With the direct-injection diesel system, it is possible to modify not only the amount of

diesel injected but also injection timing. With early injection, operation is closer to that of HCCI because the diesel fuel gets time to partially mix with the ethanol-air mixture before combustion. Advantages with the fumigation setup is that operation is possible at low loads without preheating of the mixture, and straightforward switching to traditional diesel operation outside the HCCI operating regime. A complication is the difficulty of obtaining HCCI-like emissions, due to inhomogeneities in the fuel-air mixture.

There are three main contributions of this work. The first is the control system that proves to be essential for running the multi-cylinder engine without exhaustive calibration for each cylinder individually. The second contribution is the experimental results at 9.2 bar BMEP where it is shown that the fumigation approach results in HCCI-like emissions with reasonably high efficiency and low pressure derivatives. The third contribution is the high load experiments, where it is shown that the engine can operate using fumigation at 18.4 bar BMEP with low maximum pressure derivatives dp and relatively low emissions of NO_x .

EXPERIMENTAL SETUP

The experimental setup consisted of a turbo charged six cylinder heavy duty Volvo diesel engine which has been used in previous studies, see [15, 16]. The engine specifications are presented in Table 1. The engine was equipped with port fuel injection as a complement to the standard DI system. For these experiments the diesel was injected with the DI system and ethanol was injected in the port injection system. The engine was equipped with a low-pressure EGR system with an EGR cooler. The amount of EGR could be adjusted by two electrical servo-operated valves.

Table 1: Engine specification

Operated cylinders	6
Displaced Volume	2000 cm ³
Bore	131 mm
Stroke	150 mm
Connecting Rod Length	260 mm
Number of valves	4
Compression ratio	18.5 : 1
Fuel supply	PFI + DI
IVC baseline	570@1 mm

The control system was a combined data acquisition and engine control system. The engine control system was based on a PC-computer running a Linux operating system. The use of a PC-computer made it possible to have a flexible platform for development of custom control systems. The cycle-to-cycle control was possible through the good soft real-time properties of Linux.

All data coming from various sensors were collected by the main control program. This data included the various temperatures and pressures collected via a slow data logger. Smoke was measured using an SWRI opacime

ter. Smoke data is presented as a percentage, ranging from 0% (no smoke) to 100% (completely opaque exhaust gas). All the cylinders were equipped with piezo-electric water-cooled Kistler 7061B pressure transducers. The cylinder pressure data acquisition was controlled by an encoder that was connected to the crank shaft of the engine. A sample was taken every encoder pulse, i.e. every 0.2 crank angle degree. The cylinder pressure data were sampled by a Microstar DAP 5400a/627 data acquisition card. As soon as one cylinder passed exhaust valve opening, pressure data was transferred from the AD card to the control program. When a new pressure curve was available in the control program, a simplified rate of heat release calculation was performed where controller inputs $IMEP_n$ and dp were computed. By pressure derivative dp , it is meant $dp/d\theta$, maximum pressure derivative with respect to crank angle degree during the cycle. The simplified heat release rate calculation was based on the apparent heat release rate with a fixed ratio of specific heats [17].

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} p(\theta) \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V(\theta) \frac{dp}{d\theta}$$

The controllers were implemented using Simulink and converted to C-code using the automatic code generation tool Real Time Workshop. The C-code was then compiled to an executable program which communicated with the main control program.

MULTI-CYLINDER ASPECTS

It is well known that in a multi-cylinder engine, combustion can differ significantly between cylinders even though all tunable parameters such as fuel amount and injections timing are set equally. In the type of combustion concept studied here, with auto-ignition triggered by local thermodynamic and chemical conditions, these differences are significant.

CYLINDER-TO-CYLINDER VARIATIONS

There are many plausible causes for cylinder-to-cylinder differences in combustion, including:

- Fuel injectors do not inject equal amounts of fuel.
- Different compression ratios in cylinders due to geometrical variations.
- Volumetric efficiency varies due to variations in intake manifold fluid flow.
- Cooling water temperatures differ, resulting in different thermodynamic conditions for auto-ignition.

Cylinder-to-cylinder variations were reported as an issue in [13], where EGR concentration variations were thought to be the main cause. In that work, only one cylinder

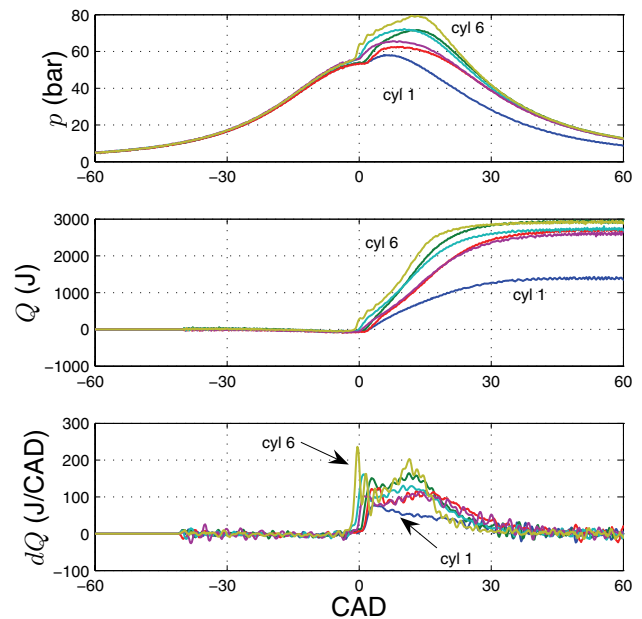


Figure 1: Pressure, cumulative heat released, and filtered heat release rate for six cylinders during a cycle at steady-state conditions.

was equipped with a pressure sensor so that cylinder-individual information on combustion was limited, and no compensation was applied for the variations. The variations were reported to limit the operating range that could be tested, and to reduce efficiency and increase certain emissions.

Also in this research the cylinder-to-cylinder variation was a challenge which is illustrated in Figure 1, showing typical pressure curves of the six cylinders with the same nominal amount of fuel injected and the same injection timings. It can be seen that there are large differences between the cylinders. In cylinder 1 (blue curve), the initial heat released from the diesel combustion appears to be too small to ignite the ethanol leading to misfire-like combustion where only half of the injected fuel is burned. In cylinder 6 (yellow curve), the diesel combustion is rapid, leading to complete combustion but high maximum pressures and pressure derivatives. In Figure 2, it can be seen that the cylinder-to-cylinder variations are systematic, and larger than the cycle-to-cycle variations.

A major contributor to cylinder-to-cylinder variations in combustion is thought to be the diesel fuel injection system. Unit injectors designed for conventional diesel operation were used. These injectors are calibrated for substantially larger diesel flows, so that the resolution is very poor at the levels used in the setting of this work. That implies that even though the injection duration was nominally the same for all cylinders, the injected amount may vary by a large factor among the cylinders.

It could be argued that if this combustion concept would be put in production, a diesel injection system adapted to

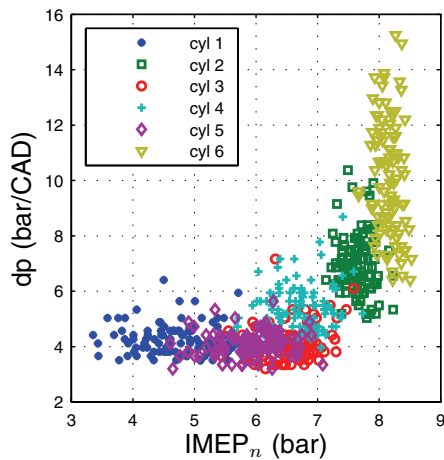


Figure 2: Maximum pressure derivative (dp) and $IMEP_n$ for 100 consecutive cycles for the six cylinders, with all nominal conditions set equally.

the low diesel flows would be applied and the cylinder-to-cylinder variations would be reduced. Even so, there are other sources of cylinder-to-cylinder combustion variations that must be addressed, as noted in [13]. There is therefore a need to compensate for these variations.

CONTROL STRUCTURE

The approach taken here to cylinder-to-cylinder variations is to apply closed-loop control based on cylinder pressure measurements. In a research setting, this approach has several advantages. Eliminating cylinder-to-cylinder variations results in well-defined operating conditions. Emissions and efficiency can be related to the actual combustion process observed in the pressure trace. Most importantly, differences in amount of fuel injected are automatically compensated for at each operating point, so that no extensive calibration of the fuel system is required.

To harmonize the combustion between the cylinders, the amounts of the two fuels were manipulated. As feedback variables, the maximum pressure derivative during the cycle, dp , and the net indicated mean efficient pressure, $IMEP_n$, were used. It was found that if these variables were equal, pressure traces between the cylinders were similar.

Experimental testing showed that maximum pressure derivative was strongly correlated with amount of diesel injected. Therefore, the duration of diesel injection was used to control pressure derivative to a setpoint, and amount of ethanol injected was used to control $IMEP_n$ to a setpoint. The effect of varying the amount of diesel could be studied by varying the maximum pressure derivative setpoint. Both the dp and the $IMEP_n$ controllers were of PI-type. The control structure is shown in Figure 4.

The $IMEP_n$ and dp for 100 consecutive cycles for the six cylinders with the controller activated are shown in Figure 3. Note the difference from Figure 2.

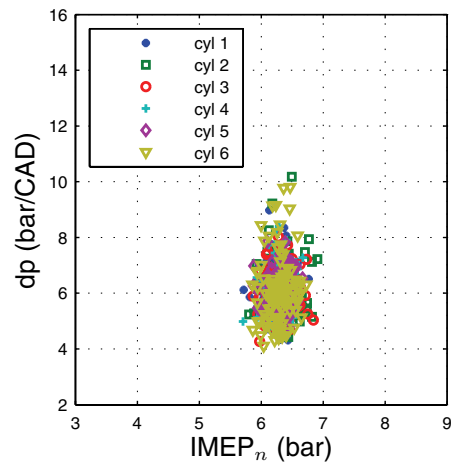


Figure 3: Maximum pressure derivative (dp) and $IMEP_n$ for 100 consecutive cycles for the six cylinders, with controller activated.

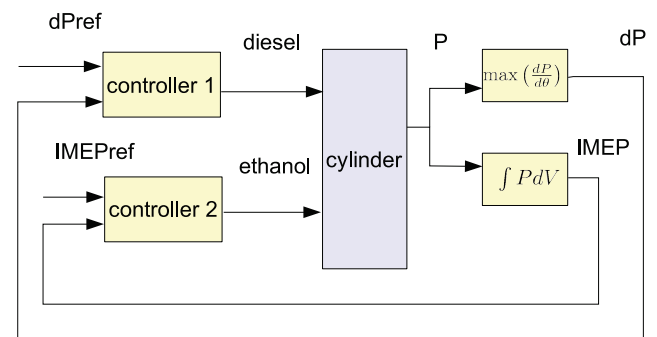


Figure 4: Control structure used for control of dp and $IMEP_n$.

Controllers were designed ignoring the coupling between the two control variables, diesel and ethanol injected, and the two measured variables, dp and $IMEP_n$. To avoid exciting the cross-coupling, the controllers were tuned conservatively, which was satisfactory for the steady-state operation experiments presented here. For transient operation, a MIMO control design that takes into account the cross-coupled dynamics of the process should be developed.

OPEN-LOOP OPERATION

Closed-loop control based on cylinder pressure sensors is a powerful tool in the engine development process, but within the foreseeable future such sensors cannot be incorporated into a production engine due to cost and durability concerns. It is therefore of interest to examine whether operation is possible in open loop.

Experiments were conducted to examine open-loop op-