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Direct Use of Wet Ethanol in a Homogeneous Charge Compression Ignition (HCCI) Engine: Experimental and Numerical Results

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Homogeneous Charge Compression Ignition (HCCI) engines are amenable to a large variety of fuels as long as the fuel can be fully vaporized, sufficiently mixed with air, and receive sufficient heat during the compression stroke to reach the autoignition conditions. This study investigates an HCCI engine fueled with ethanol-in-water mixtures, which we call "wet ethanol". The motivation for using wet ethanol fuel is that significant energy is required for distillation and dehydration of fermented ethanol (from biosources, not from petroleum), thus direct use of wet ethanol could improve energy balance. Recent modeling studies have predicted that a HCCI engine can operate using fuel containing as little as 35% ethanol-in-water, with surprisingly good performance and emissions. With the previous modeling study suggesting feasibility of wet ethanol use in HCCI engines, this paper focuses on experimental operation wet ethanol in a 4-cylinder 1.9 liter engine running in HCCI mode. This study investigates the effect of the ethanol-water fraction on the engine's operating limits, intake temperatures, heat release rates, and exhaust emissions for the engine operating with 100%, 90%, 80%, 60%, and 40% ethanol-in-water mixtures.

1.0 Introduction

1.1 Homogeneous Charge Compression Ignition (HCCI)

HCCI has promise in that it takes advantage of favorable characteristics inherent to both the spark ignition (SI) engine and the diesel engine [1-4]. It is premixed like a spark ignition (SI) engine and compression ignited like a diesel engine. Because of the typically lean fueling rates, the low flame temperatures result in significantly reduced emissions of oxides of nitrogen (NOx). Since the fuel and air in the HCCI engine are premixed, there is no diffusion flame burning (as in a Diesel engine) so particulate matter (PM) emissions are also quite low. HCCI engines also run at high compression ratios which directly increase the efficiency of the engine and consequently increases the fuel economy by a significant amount.

Fuel flexibility is another interesting aspect of HCCI engines. The combustion of gasoline, propane, diesel fuels, and other common hydrocarbon fuels has been studied in depth. This paper looks into HCCI combustion of alcohols, specifically ethanol (EtOH). The production of ethanol from biological sources instead of petroleum allows one to look at EtOH as a renewable fuel.

Currently, ethanol is often included in standard gasoline as a knock suppressor and since it is a liquid at standard conditions the current infrastructure for widespread use is largely in place.

1.2 Wet Ethanol HCCI

HCCI engines are inherently fuel flexible and can run on low-grade fuels as long as the fuel can be heated to the point of autoignition. In particular, HCCI engines can run on “wet ethanol”, ethanol-in-water mixtures with high concentrations of water. Considering that much of the energy required for processing fermented ethanol (from biosources, not from petroleum) is spent in distillation and dehydration, direct use of wet ethanol in HCCI engines considerably shifts the energy balance in favor of ethanol.

In a recent study by Flowers et al. [4], numerical predictions show that a HCCI engine with efficient heat recovery can operate on a 35% ethanol-in-water mixture while achieving a high brake thermal efficiency (38.7%) and very low NOx (1.6 parts per million, clean enough to meet any existing emissions standards). Direct utilization of ethanol at a 35% volume fraction reduces water separation cost to only 3% of the energy of ethanol and coproducts (versus 37% for producing pure ethanol), and improves the net energy gain from 21% to 55% of the energy of ethanol and coproducts. Figures 1 and 2 show the net energy balance for ethanol. The full circle represents the energy of ethanol and coproducts. The figure shows energy consumption in all stages of ethanol production from corn, as a percent of the heating value of ethanol and coproducts. The energy that remains after accounting for all the energy consumption is the net energy gain, and it has two components: net energy in the ethanol and net energy in the coproducts [5,6].

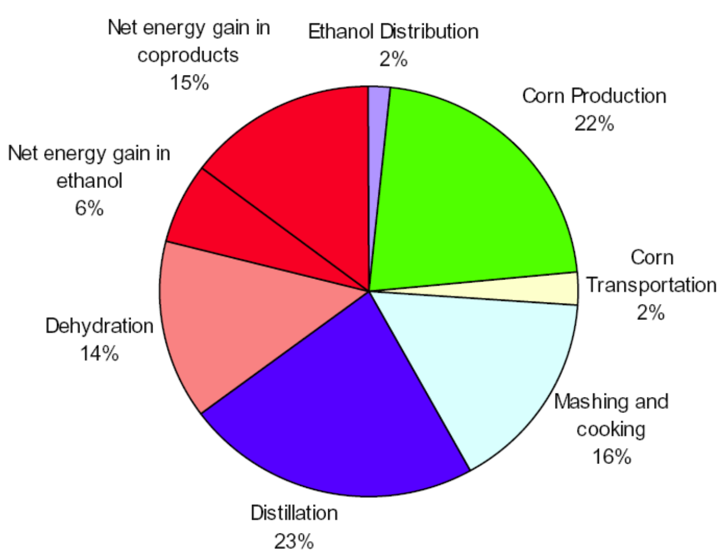


Figure 1. Net energy balance for ethanol. The full circle represents the energy of ethanol and coproducts. The figure shows energy consumption in all stages of ethanol production from corn, as a percent of the heating value of ethanol and coproducts. The energy that remains after accounting for all the energy consumption is the net energy gain, and it has two components: net energy in the ethanol and net energy in the coproducts.

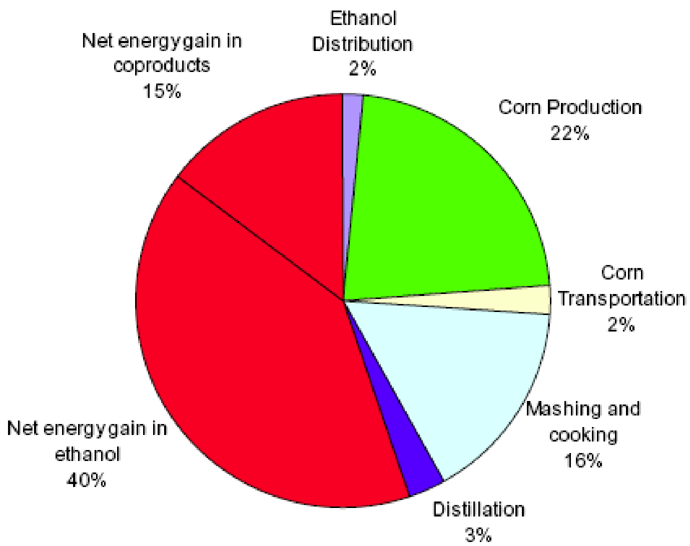


Figure 2. Net energy balance for ethanol, considering that ethanol is used as a 35-65% mixture of ethanol and water. The figure shows energy consumption in all stages of ethanol production from corn, as a percent of the heating value of ethanol and coproducts. The energy that remains after accounting for all the energy consumption is the net energy gain, and it has two components: net energy in the ethanol and net energy in the coproducts.

The modeling results indicate that running with ethanol-in water mixtures containing as high a fraction of water as the engine will tolerate results in improving the overall energy balance of bio-derived ethanol. This paper focuses on running an experimental engine in HCCI mode with fuels containing various ratios of ethanol to water. We seek to assess the performance and emissions of the engine operating on wet fuel, as well as any other practical operational issues.

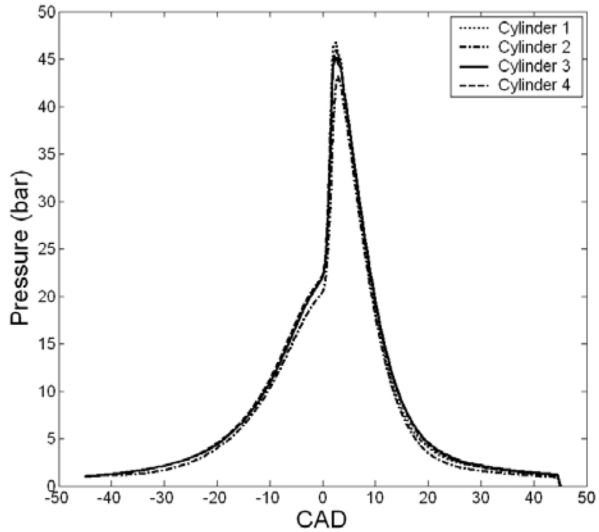
The following set of experiments show the operation of a modified VW 4-cylinder 1.9L engine running in HCCI mode with varied water-in-ethanol mixtures. The engine was run on water-in-ethanol mixtures containing 100%, 90%, 80%, 60%, and 40% ethanol-in-water (by liquid volume). The ethanol fuelling rates are held constant (overall equivalence ratio of PHI = 0.28) in all experiments. The following subsections discuss the operation limits of the engine, the effect of water addition on required intake temperature, changes in heat release profiles, and exhaust emissions.

2.0 Operating Limits

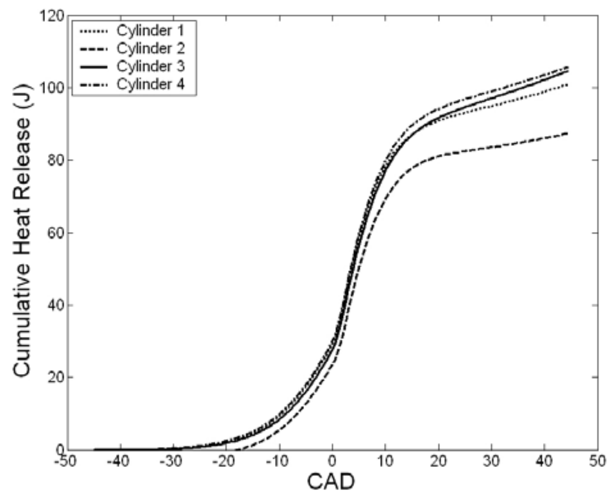
Running a HCCI engine with extra water in the fuel provides challenges beyond those encountered during normal HCCI operation, mainly that the fuel and water must be fully evaporated and the fuel-air-diluent mixture must be heated to ignition. From a mechanical point of view, the conventional fuel delivery system is not designed for water. Fuel injectors and the fuel pump are susceptible to damage. Care must be taken to avoid these damages (that is, flushing the system with gasoline after each use). Additionally, in order to fully evaporate the fuel and water, additional heat must be added to the intake charge to evaporate the fuel and achieve intake temperatures required to control combustion timing. Significantly more heat is required to vaporize the fuel and water than for ethanol alone, in part because of the additional mass of water to be evaporated with the fuel, and because water has a higher latent heat of vaporization than pure ethanol. We consider full evaporation of fuel and water to be a requirement, deeming it undesirable to introduce unevaporated liquids into the combustion chamber because of potential damage to the engine.

We first investigate how little distillation is required to achieve satisfactory performance, or what is the lowest fraction of ethanol-in-water that operates stably in the HCCI engine. Figure 3 shows the combustion performance of the VW TDi running on a mixture of 60% ethanol-in-water. At 60% ethanol-in-water, the required intake temperatures are sufficiently low that a wide range of combustion timings are possible. Each cylinder is firing near TDC with only slight differences in peak pressure. However, when the concentration of ethanol is decreased to 40%, the ability to achieve HCCI becomes extremely difficult, as seen in Figure 4. Each cylinder autoignites differently and cylinder balancing becomes nearly impossible at 40% ethanol-in-water.

The poor performance at 40% ethanol-in-water may be due to the limits of the laboratory intake preheater. The heat provided by the electrical preheater may not be sufficient to both evaporate the mixture and achieve intake temperatures needed for autoignition. The wet ethanol modeling study [REF] used exhaust heat recovery, which may provide more heat than the electrical preheater. The upper limits of the intake heater limit the ability to achieve the combustion at earlier combustion timings, causing concern that not all the injected water evaporates.

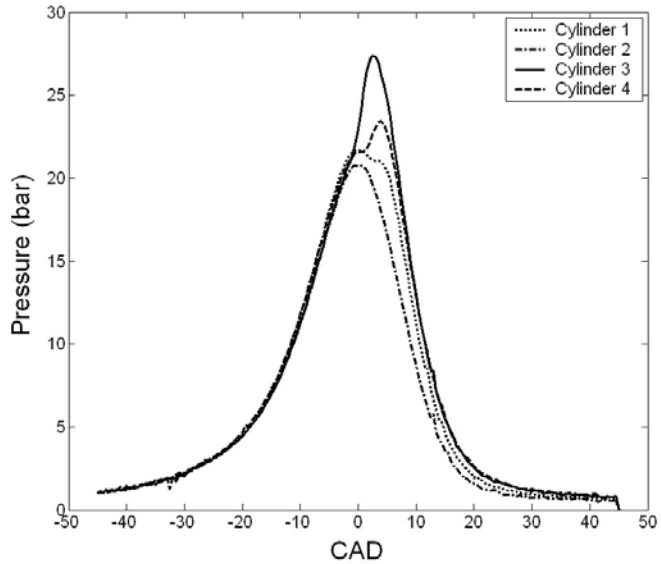


(a) Pressure Traces.

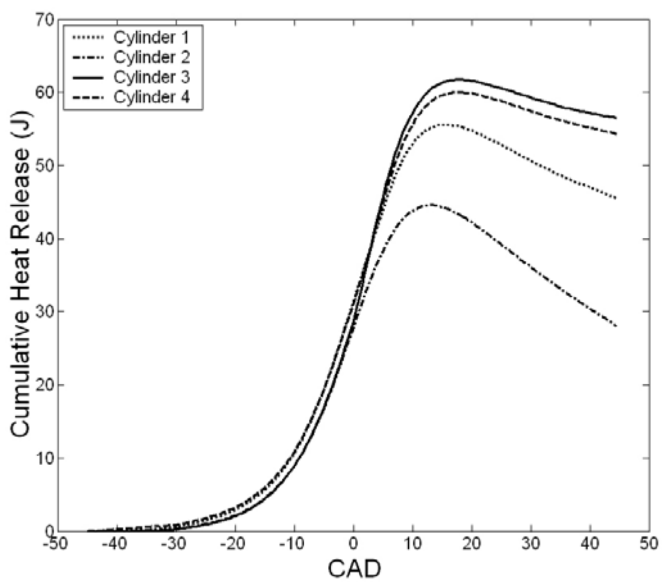


(b) Cumulative Heat Release Profiles.

Figure 3. Pressure traces and cumulative heat release profiles for each cylinder of the VW TDi HCCI engine running on a 60% EtOH in water mixture.



(a) Pressure Traces.



(b) Cumulative Heat Release Profiles.

Figure 4. Pressure traces and cumulative heat release profiles for each cylinder of the VW TDi HCCI engine running on a 40% EtOH in water mixture.

3.0 Experimental Results

3.1 Experimental Methods

A VW 1.9L 4-cylinder engine was used in this set of experiments. Table 1 provides the engine specifications.

Table 1. VW 1.9L HCCI Engine Specifications.

Configuration	4 cylinder
Firing order	1-3-4-2
Displacement	1.9 L
Compression ratio	17.0
Bore	79.5 mm
Stroke	95.5 mm
Connecting rod length	144.0 mm
Valves (intake, exhaust)	2, 2
Intake valve open (IVO)	16 CAD ATDC
Intake valve close (IVC)	25 CAD ABDC
Exhaust valve open (EVO)	28 CAD BBDC
Exhaust valve close (EVC)	19 CAD ATDC
Engine speed	1800 +/- 5 RPM

Although the original VW 1.9L engine is a turbo-charged direct-injected Diesel engine, the turbo charger and direct fuel injectors were removed. In place of the fuel injectors, water-cooled piezoelectric pressure transducers were installed in order to obtain pressure traces.

Figure 5 shows a schematic of the experimental set-up. The engine is mounted to an Eaton 30 kW induction motor and runs at a fixed speed of 1800 RPM. A 3600-count BEI optical encoder is attached to the crankshaft and used for engine-position synchronization. AVL QH33D water-cooled quartz pressure transducers are mounted in the direct injector ports and used to measure cylinder pressure. A Sierra Instruments 840H mass flow meter is used to measure the fuel flow. An 18 kW Osram-Sylvania 3-element quartz heater is used to preheat the intake air. Gaseous fuels, such as propane and acetylene, are injected approximately 1 meter upstream and controlled with a fuel flow meter (Sierra Instruments Side-Trak 840 mass flow controller). Liquid fuels are port injected by MSD injectors and controlled with the MSD software.

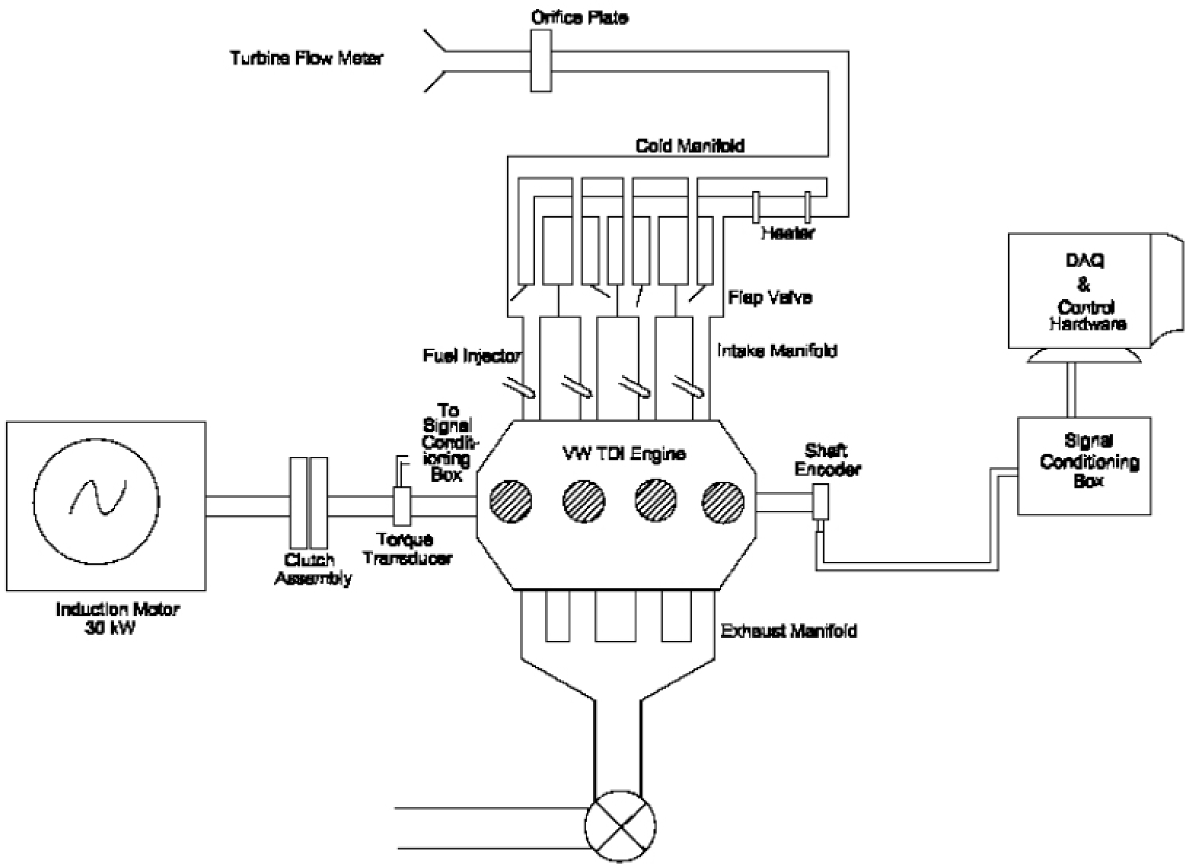


Figure 5. Schematic of the Volkswagen TDI 4-cylinder experimental HCCI engine at the UC Berkeley Combustion Analysis Laboratory.

This engine has used a number of different methods for controlling the autoignition event. The experiments in this paper use the intake air heater to control the intake air temperature and thus the combustion timing. Since the VW TDi is a multi-cylinder engine, the cylinders should be "balanced" so that each cylinder experiences the same combustion timing. Balancing is done using the fast thermal management system seen in Figure 5 which mixes separate hot and cold air streams for each cylinder so that timing can be changed independent of other cylinders.

3.2 Effect on Intake Temperature

Start of combustion was controlled using intake heating for all fuel mixtures. Figure 6 shows the required intake temperature for 4 different water-in-ethanol fuel blends at a variety of combustion timings (an average for all 4 cylinders). An increase of 40 degrees Celsius is needed to achieve combustion at later CA50 when the composition of the mixture is changed from 100% ethanol to 40% ethanol. For the 40% ethanol mixture, earlier combustion timings are not possible due to the constraints on the intake heater.

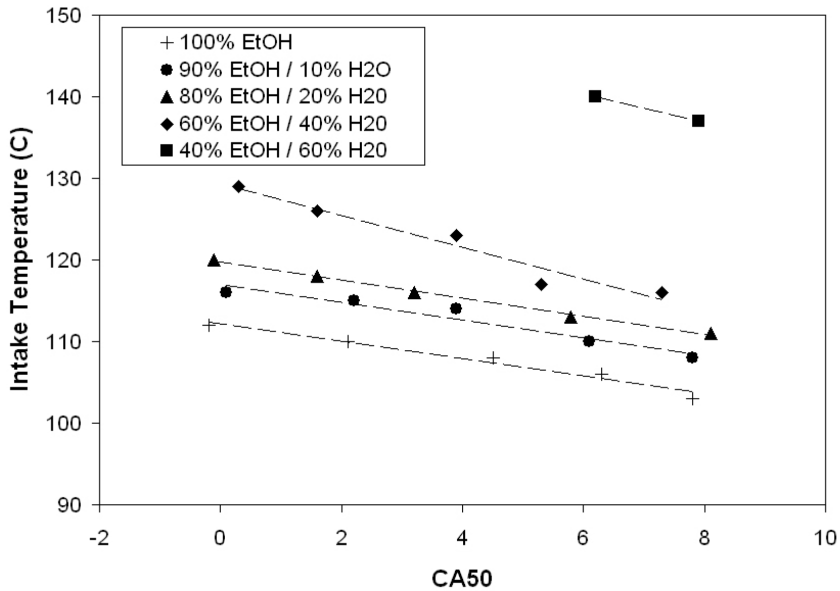


Figure 6. Intake temperature before fuel injection for mixtures of water and ethanol. The temperature is plotted against an averaged CA50 for all 4 cylinders of the VW TDi HCCI engine.

3.3 Effect on Heat Release

The heat release profiles for a single cylinder of the VW TDi running on different fuel blends of wet ethanol is shown in Figure 7. The start of combustion (CA50) in each cylinder is approximately 4 degrees ATDC. The peak cumulative heat release decreases with the addition of water to the fuel blend. This is most likely due to energy lost from the combustion event to vaporization of the water present in the mixture.

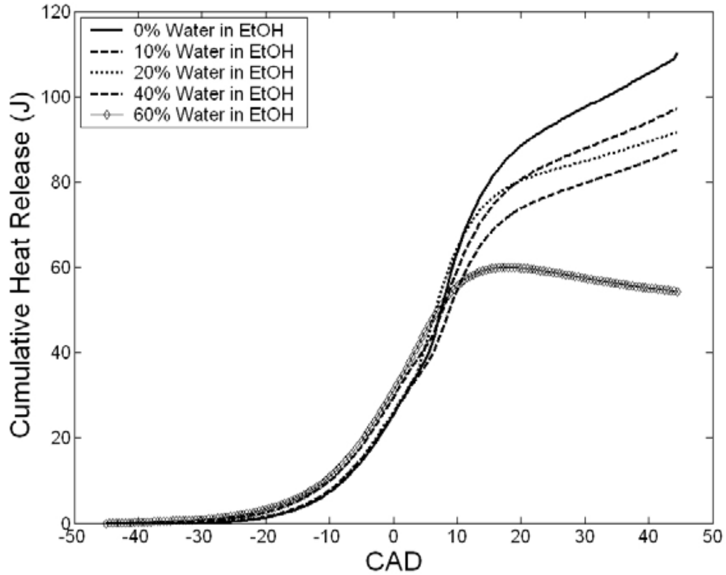


Figure 7. The cumulative heat release profiles for different blends of water in ethanol in a single cylinder. Higher concentrations of water decrease the total amount of heat release. The ethanol fuelling rates are held constant (overall equivalence ratio of PHI= 0.28) and combustion timings are similar (CA50 around 4 degrees ATDC).

3.4 Emissions

The exhaust emissions of a HCCI engine operating on wet ethanol are shown in Figures 8-11. Data is provided for fuel mixtures of 100%, 80%, and 60% ethanol-in-water. Higher concentrations of water resulted in unreliable combustion (complete misfire on some cylinders) and yielded erroneous readings.

As water concentration in fuel increases the HC emissions increase as shown if Figure 8 indicating incomplete conversion of ethanol to combustion products. Figure 9 shows that less oxygen is present in the exhaust of higher water concentration mixtures. This is due to the displacement of oxygen by water vapor in the combustion chamber. An increase in water vapor means less air can be inhaled during the intake stroke.

Incomplete combustion also leads to the increase in CO emissions at the later CA50 shown in Figure 10. The NOx emissions shown in Figure 11 are extremely low, as is expected in HCCI engines. An explanation for the increase in NOx emissions with increased water concentration may come from the discussion on the nitrous oxide (N2O) mechanism. Higher pressures resulting from the increase of water in the combustion chamber may encourage the nitrous oxide (N2O) mechanism. The low peak temperatures and lack of a flame front do not affect the thermal and prompt mechanisms in the formation of NOx. Numerical modeling may give further insight into NOx formation.

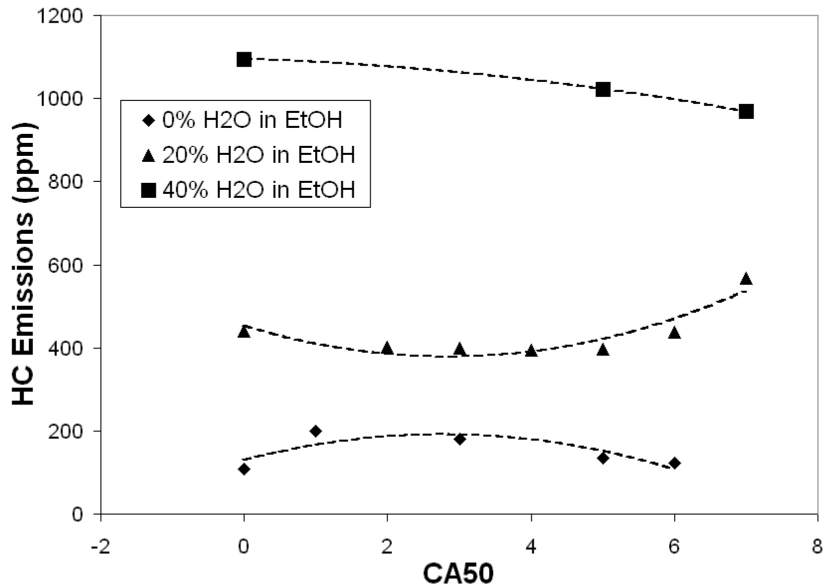


Figure 8. Experimental HC emissions data for the 4 cylinder VW TDi HCCI engine. Emissions are plotted against estimated CA50 averaged for all cylinders.

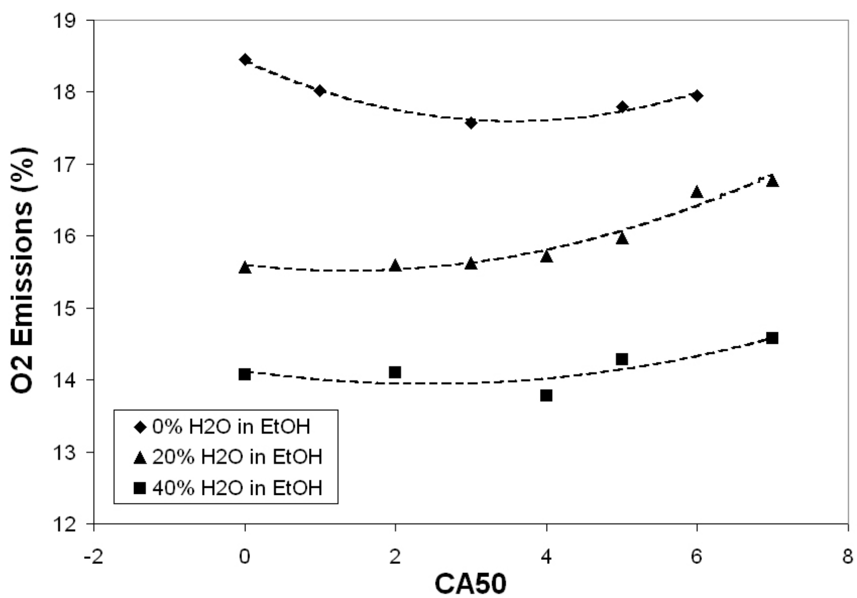


Figure 9. Experimental O2 emissions data for the 4 cylinder VW TDi HCCI engine. Emissions are plotted against estimated CA50 averaged for all cylinders.

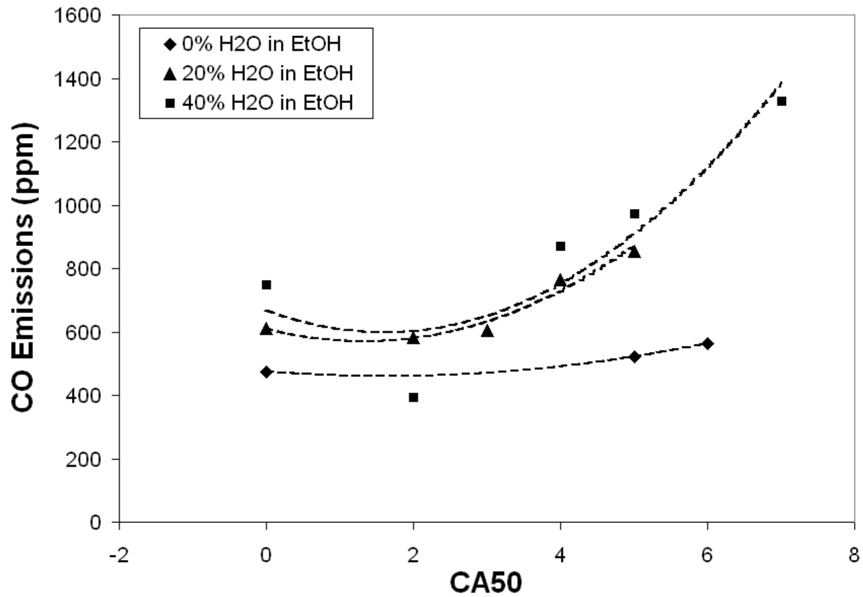


Figure 10. Experimental CO emissions data for the 4 cylinder VW TDi HCCI engine. Emissions are plotted against estimated CA50 averaged for all cylinders.

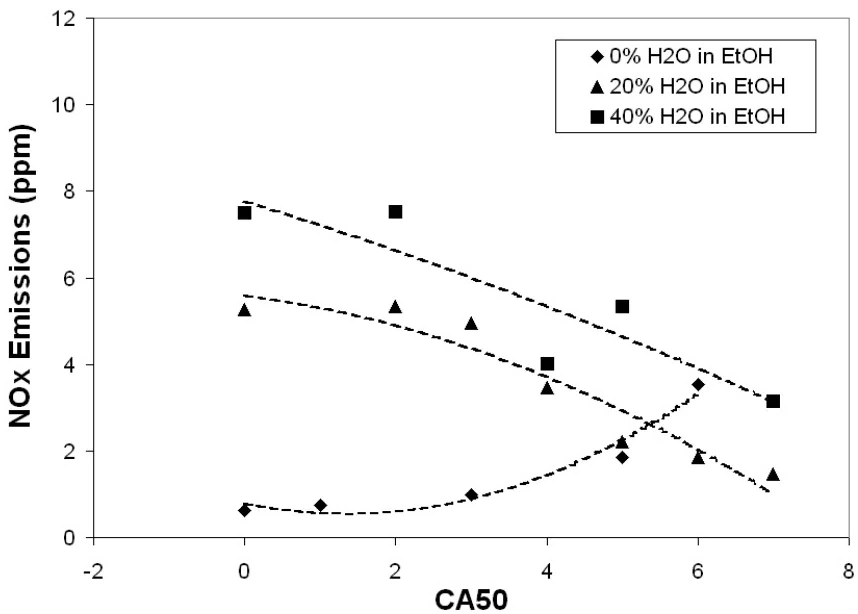


Figure 11. Experimental NOx emissions data for the 4 cylinder VW TDi HCCI engine. Emissions are plotted against estimated CA50 averaged for all cylinders.

4.0 Concluding Remarks

This paper discussed experimental results from a HCCI engine running on wet ethanol. Fuel mixtures studied range from 100% ethanol to 40% ethanol-in-water. HCCI operation was obtained in mixtures up to 40% ethanol-in-water. Incomplete combustion and excessive intake temperatures limited the operating range at higher water concentrations. The maximum value of the cumulative heat release profiles deceases with an increase in water concentration. Exhaust emissions data is also presented and discussed. Hydrocarbon and carbon monoxide emissions tend to increase with increasing fuel water content. The next step in understanding how wet ethanol mixtures behave in a HCCI engine is numerical modeling. Multi-zone models that describe a multi-cylinder engine are extremely difficult, but a single zone numerical study using an appropriate mechanism could prove quite useful in investigating wet ethanol HCCI combustion.

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