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# Fumigation of a heavy duty common rail marine diesel engine with ethanol–water mixtures

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ABSTRACT

A heavy duty common rail marine diesel engine operating with two stage injection is tested under load on a test bench with vapourised ethanol–water mixtures mixed into the inlet air at various rates. Ethanol/water mixture strengths of 93%, 72% and 45% by mass are tested. Results are presented for two engine loads at 1800 rpm, with brake mean effective pressure (BMEP) 17 bar and 20 bar. At each test point, constant engine speed and brake torque are maintained for various rates of aqueous ethanol addition. Small increases in brake thermal efficiency are measured with moderate rates of ethanol addition at a BMEP of 20 bar. Exhaust emissions of oxides of nitrogen, carbon monoxide, hydrocarbons, oxygen and carbon dioxide, and exhaust opacity are measured. CO emissions and exhaust opacity tend to increase with increased ethanol addition. NOx emissions tend to decrease with increased ethanol addition and with increased water content. Hydrocarbon emissions remain low, near the detection limit of the analyser. Cylinder pressure and the electronically controlled two stage liquid fuel injection timing are recorded with a high speed data acquisition system. Apparent heat release rate is calculated from the measured cylinder pressure. The apparent heat release rate and fuel injection timing together allow analysis of the mechanism of the combustion process with ethanol fumigation. Two stage injection involves a small pre-injection of diesel fuel to reduce early pressure rise rates in normal diesel engine combustion. Even though injection timing is retarded by the Engine Control Unit as more ethanol is added, combustion timing effectively advances due to the effect of two stage injection. Where the ethanol/air mixture strength is above the lower flammability limit at compression temperatures, the mixture is ignited by the pre-injection and begins to burn rapidly by flame propagation and/or autoignitive propagation before the main liquid fuel injection begins. This occurs for ethanol energy substitution rates greater than 30%. Two distinct peaks in heat release rate appear at the higher ethanol rates. Severe knock becomes apparent for 34% ethanol. Two stage injection may be disadvantageous in these circumstances.

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## 1. Introduction

Diesel engines can be readily configured to run in dual fuel mode, with gas mixed into the air intake while liquid diesel fuel is injected as normal, but at a reduced rate. This is sometimes called fumigation. Various gases and gas mixtures have been used for this purpose including methane, ethane, propane, butane, hydrogen, ethylene, liquefied petroleum gas, landfill gases and process gases.

The work described here was conducted as part of a project to assess alternative fuels for fishing vessels. Fishing vessel operators are looking to reduce fuel costs to make their operations more viable. Marine engines in such applications tend to be operated at steady loads with relatively high brake mean effective pressure (BMEP), for long periods of time. A common rail, electronically controlled marine engine with two stage injection is used in the present study. In normal engine operation, two stage injection is used to reduce the severity of the second phase of combustion, by reducing the amount of fuel/air mixture formed during the delay period. The use of aqueous ethanol in an electronically controlled diesel engine with two stage injection is of interest in because of possible interactions of the added gas with the ignition process. Adding gaseous fuel to the intake air will increase the amount of fuel air mixture available during and after the delay period and thus impact on the ignition delay and combustion rate.

Ethanol can be produced from grains and sugar. Ethanol produced in this way is typical of a first generation biofuel. Alternative scenarios for ethanol involve the use of ligno-cellulosic biomass (second generation biofuel). As a biofuel it has the potential to reduce greenhouse gas emissions [1]. Flowers et al [2]demonstrate that the energy costs of removing water from ethanol produced by fermentation become very significant as the azeotropic

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concentration is approached. Thus, if aqueous ethanol mixtures can be used in engines then the lifecycle $CO_2$ emissions will be reduced compared with the use of anhydrous ethanol. The most straightforward means of utilising ethanol is to add it to the air intake. This method has been known for some time [3] but there are no commercial systems available.

It is also possible that small amounts of ethanol, around 10% by energy content, will increase engine fuel efficiency by enhancing combustion rates, and thus provide fuel savings even if ethanol cost is similar to normal diesel fuel cost. This has yet to be confirmed experimentally.

Numerous studies have been published on ethanol fumigation of diesel engines, with the ethanol being introduced by various techniques such as carburetion, continuous injection under pressure after the turbocharger and multipoint sequential injection. Ottikkutti et al [4] injected ethanol/water mixtures as an atomised spray into the intake air after the turbocharger in a bowl-in-piston direct injection diesel engine. They found no significant changes in thermal efficiency at 100% load or at 50% load, over a range of energy substitution rates. Exhaust temperature and NOx decreased. Hayes et al [5] reviewed the work of a number of earlier trials of ethanol fumigation using a range of ethanol/water ratios. No clear trends in the effect of ethanol addition on thermal efficiency were reported from the earlier works. NO emissions tended to reduce, the reduction being greater with greater water content, as would be expected.

Hayes et al [5] used direct injection of ethanol/water mixes of varying water contents into the inlet port of each cylinder of a turbocharged six cylinder direct injection engine with mechanical governor. Injection timing in such an engine would be fixed at a given speed, for all loads. At high load, thermal efficiency increased with increased ethanol flow rate, for all values of ethanol/water ratio. The maximum gain in thermal efficiency was about 5% of the diesel fuel only efficiency at the highest load test point at 8 bar BMEP. At intermediate load, thermal efficiency was unchanged. At low loads, thermal efficiency decreased with increased ethanol addition, down by up to 25% of the diesel fuel only value, at maximum ethanol flow rate. The ethanol/water ratio had little effect on this trend. At 8 bar BMEP NO emissions decreased with increased water content and fumigation rates, but a small increase in NO emissions was measured for ethanol/water mixtures greater than 68% by mass.

Chen et al [6] injected ethanol through an atomising nozzle after the compressor in a turbocharged four cylinder diesel engine. They found that thermal efficiency increased with increased ethanol at high load, but decreased at low load. Measurements of exhaust temperature for individual cylinders led the authors to believe that the distribution of ethanol from cylinder to cylinder was not uniform. There was no aftercooler to enhance mixing and the pressure of the ethanol supplied to the ethanol nozzle was relatively low at 3.4 bar. Abu-Qudais et al. [7] added ethanol to the intake air of a naturally aspirated single cylinder engine by use of a spray nozzle. They used a single energy substitution rate of 20% by energy over a range of speeds at full load. CO and HC emissions increased significantly for all tests, exhaust opacity and mass of particulates decreased. Thermal efficiency increased by more than 7% of the diesel fuel only value. It was postulated that the increased ignition delay with ethanol fumigation led to more rapid combustion and thus improved thermal efficiency.

Kowalewicz [8] used port injection of 92% by volume ethanol in a single cylinder direct injection diesel engine operating on Rape oil methyl ester as the main fuel. Ethanol was supplied up to 50% of total fuel energy. Exhaust temperature, ignition delay, thermal efficiency and smoke emissions increased with increasing ethanol supply rate, but combustion duration decreased. NOx emissions reduced at low loads but increased at high loads with increasing ethanol supply rate.

Surawski et al [9,10] found increased emissions of CO, HC and particle mass from a four cylinder naturally aspirated diesel engine fumigated with ethanol, while NO emissions reduced. They characterised particle emissions and found that ethanol fumigation increased the volatility of particles and increased the concentrations of particle related reactive oxygen species, potentially requiring the use of a Diesel oxidation catalyst, which would also reduce CO and HC emissions.

Abu-Qudais et al. [7] tested a single cylinder diesel engine with ethanol fumigation and at 20% ethanol by energy found increased thermal efficiency, increased emissions of CO and HC, but reduced exhaust opacity (smoke) and reduced particle mass.

In summary, previous work suggests that thermal efficiency tends to increase at high loads and decrease at low loads. NOx and mass of particle emissions tend to decrease for all loads and CO and HC emissions tend to increase for all loads. The effects of ethanol usage on the nature of particle emissions may be undesirable.

For heavy duty engines it is feasible to restrict addition of the fumigant to moderate to high load operating conditions, to avoid low loads where fumigation may be detrimental to engine efficiency.

Karim [11] postulated that there are three distinct stages in dual fuel combustion, where the primary fuel is gaseous and ignition is by pilot injection of standard diesel fuel. The first stage involves combustion of some of the pilot fuel and some gaseous fuel entrained into the fuel spray. The second stage involves combustion of the remaining pilot fuel and gaseous fuel in the immediate surroundings of the pilot fuel. The third stage involves flame propagation through the remainder of the gaseous fuel–air mixture. For fumigation of a diesel engine where the majority of the fuel energy comes from standard diesel fuel, the combustion processes may differ from this description. However, the three modes described by Karim may be a guide to understanding the combustion process.

The aim of the present work is to investigate the changes in thermal efficiency and exhaust emissions in a heavy duty turbocharged diesel engine when vapourised ethanol–water mixtures are mixed into the inlet air, with no modifications to the liquid diesel fuel injection system or Engine Control Unit. A further aim is to determine the limits to the fumigation rates and to understand the combustion processes involved when a heavy duty engine with electronic two stage injection is fumigated.

## 2. Test procedures

### 2.1. Engine testbed

The engine is a Cummins QSB5.9-305MCD six cylinder in-line, common rail, electronically controlled marine diesel engine rated at 224 kW (305 bhp) at 2600 rpm, turbocharged with water cooled aftercooler, compression ratio 17.2:1, bore 102 mm, stroke 120 mm, no EGR.

A gas analyser is used which consists of an NDIR bench for $CO_2$, CO and HC and electrochemical cells for NO and $O_2$. A NOx convertor converts $NO_2$ to NO so that the NO cell measures total NOx. The convertor is placed before a refrigerated drier so $NO_2$ is not lost by dissolving in the condensate. A Bosch opacity meter draws directly from the unfiltered exhaust stream.

Labview is used as the data acquisition system. Two National Instruments acquisition cards are used – a slow speed card

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