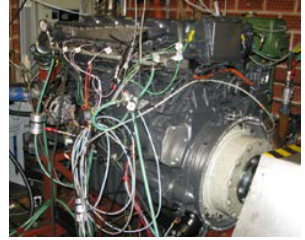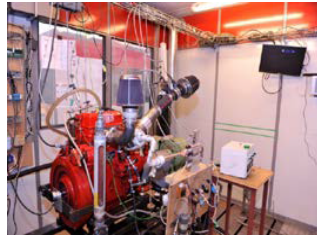Annex 46

A Report from the IEA Advanced Motor Fuels Implementing Agreement

Alcohol Applications in Compression Ignition Engines

Edited by
Jesper Schramm
Technical University of Denmark

January 2016

Preface

Summary

The report describes the results of a collaboration project: Annex 46," Alcohol application in CI engines", carried out under the umbrella of IEA Advanced Motor Fuels IA. The partners involved in the project were Scania AB, Sweden, VTT Technical Research Centre of Finland, The Technical University of Denmark (DTU) and Technological Institute Denmark (DTI).

Different activities took place: In part A tests were carried out at VTT, involving a Scania heavy duty high compression ignition engine, designed for alcohol application. The purpose was to study the behavior of neat ethanol and methanol and ethanol and methanol together with a commercial ignition additive and two new alternative additives. In part B the same fuels were tested in an experimental engine at DTU. The main purpose was to study the combustion behavior based on the measured heat release patterns. In part C the influence of alcohol addition to diesel was investigated at DTI. Different blends of diesel and alcohols, methanol, ethanol and butanol, were tested in a Scania heavy duty diesel engine in this context. In connection with the presentation of the results, a workshop with the purpose to discuss the future possibilities for alcohol application in compression ignition engines were held in Copenhagen. The results from this workshop are described in part D.

From part A the most important observation was that engine operation was rather insensitive to fuel composition, and in fact the engine operated in practice normally on all other fuels except on pure methanol. The combination of indirect and direct fuel injection turned out to be an interesting option. Injecting a limited amount of fuel into the intake manifold starts some pre-reactions and facilitates ignition of the main fuel shot.

Part B digs deeper into the combustion characteristics of the three fuels. The present experiments showed that there are differences in the ignitability of the fuels during PCCI (Partially Premixed Compression Ignition) operation. The ignition delay of the fuels can be distinguished clearly. The additives seemed to have no influence on the ignitability during HCCI (Homogeneous Charge Compression Ignition) combustion. In all fuel cases this engine was not able to burn the fuel properly during HCCI operation.

In part C measurements on the particulate emissions showed that the particulate number is reduced with increasing quantities of alcohol. The simplest alcohols ethanol and methanol gave the largest reductions, which where proportional to the fraction of alcohol in the blend. The best result was obtained with 20 % methanol and   10 % butanol as cosolvent. This blend reduced the particulate number up to 75 % in loaded operation, compared to the diesel reference. With 30 % ethanol, the reduction was up to 60 % in loaded operation. With 30 % butanol, a reduction of up to 40 % was found.

The gaseous emissions were found to be increasing with alcohol blends, but only in idle condition. The alcohol containing blends were found to increase emission of specific aldehydes, carbon monoxide and hydrocarbons. The higher cylinder and exhaust temperatures in loaded operation mean that the combustion is less sensitive to the fuel composition than at idle, and therefore only minor differences was found in loaded operation. The emission of nitrogen oxides was also affected, but no consistent increasing or decreasing trends were found.

Accurate measurements of the fuel consumption show that the brake thermal efficiency improves with increasing quantities of ethanol and methanol. Butanol does not appear to improve efficiency as significantly.

Content   Page:

Purpose and Objectives

The goal is to report the possibilities for implementation of alcohols in diesel engines. One of the main objectives of the project is to secure the supply of fuels for diesel engines, in this way by focusing on sustainable biofuels in the form of alcohols. The project involves one of the main diesel engine producers in Europe and well established research institutions and universities in Europe. The project will thus contribute to the achievement of many relevant political goals like supporting a sustainable energy policy, independency of fossil energy and reduced emissions, including carbon dioxide.

Background

In Europe, Directive 2009/28/EC on renewable energy sets transport bioenergy obligation in 2020 in minimum 10% of transport energy use. Modern spark-ignition vehicles are compatible with 10% ethanol in gasoline (E10), which represents 6 % bioenergy content. Higher ethanol content can be used only with limited car population, which limits ethanol use even if higher amounts would be commercially available (so called "blending wall"). Alcohols represent superior fuels for the SI engine, with respect to key properties like octane number and latent heat of vaporization. Basically alcohols have the ability to withstand high pressures and temperatures without igniting uncontrollably. Ethanol has become widely used in low concentration blends with gasoline in many parts of the world and has more limited use in high concentration blends. In the case of low-ethanol blends (E5-10), it is possible to produce fuels with a slightly higher or similar octane number compared to that for regular gasoline. In that case the most modern

cars are able to regulate the ignition timing and advance the timing to a degree that increases engine efficiency by a few percentage points. High share of ethanol, up to 85%, can be used in special spark-ignition FFV cars.

Alcohol and ignition improvers for diesel engines

In these years fuel economy is becoming an important issue and an obvious goal is to achieve efficiencies similar to diesel engines with the alcohol applications. However, direct application of alcohols in a diesel engine requires a fuel additive to ignite the unburned mixture. An option is to use an additive together with ethanol in heavy-duty ethanol diesel engines. Such engines are today manufactured by Scania. These engines are running with so called Etamax D fuel consisting of 95 % hydrous ethanol together with ignition improver, corrosion inhibitor and denaturants (MTBE, isobutanol). This fuel is manufactured by SEKAB in Sweden. With this concept, relatively small modifications are required in the engine. Compression ratio is increased and fuel system modified. Exhaust catalyst is developed to prevent excessive aldehyde emissions. This concept is used in buses for example in Stockholm. In total, around 1000 heavy-duty vehicles are running with Scania's ethanol engines.

The most interesting option would be a "Flex fuel" diesel engine capable to run with both ethanol and diesel fuel without pilot injection technology. Engines that can use only ethanol are potential only in restricted areas, where availability of fuel can be controlled. Diesel engines and their control technologies have developed considerably over past years. For example, common-rail system enables fine adjustments of injection. Ethanol diesel engines could be used in road transport, machinery, marine sector and especially in the countries, where ethanol is produced in large scale.

Since alcohols, particularly produced from biomass, are obvious fuels for

more intense combustion engine application in the near future it is of interest to start up a general study on the best way to do it. The goal is to combine good fuel economy with low emissions. Scania's current technology is based on an additive package including ignition improver and lubricity additive (and a high compression ratio of the engine). The project is suited for an IEA AMF study since many member countries are interested in the fuel/additive development. There is interest in looking at alternative, locally produced additive packages in e.g. Brazil, Finland and Thailand. In addition, there are ideas for new combustion schemes, e.g. RCC (reactant controlled combustion), which could eliminate the need for the dedicated ignition improver additive.

Different additives are tested in this investigation. Tests have been carried out in a production diesel engine and in a research engine. The results from these investigations are reported in part A and part B respectively of this report.

Diesel/alcohol blends in diesel engines

It is generally thought that diesel can be substituted up to 25% with ethanol when using emulsifiers. However, 15% is often considered as the optimum substitution [1]. The benefit is that the addition of an oxygenated fuel can lead to a substantial reduction in the formation of particulate matter. The effect on the other undesired emissions in the exhaust such as nitric oxides, carbon monoxide and hydrocarbons is less well established and therefore more likely dependent on the specific engine technology and operational conditions.

The use of diesel fuel blends with oxygenated components such as alcohols may be relevant in particular for older engines that do not comply with the newer standards for emissions. These engines are typically not equipped

with advanced exhaust after treatment systems, which can reduce emissions of particulate matter and nitric oxides. Engines that are not built or adapted to comply with the strict road vehicle legislation are typically found in e.g. non-road machinery, railway locomotives, all sizes of marine engines and diesel generators. Emissions from these engines are significant and sometimes the major sources of particulate matter and nitric oxides.

Blending of diesel with methanol, ethanol and butanol was investigated in this context, and the results are described in part C of this report.

Diesel-alcohol dual fuel engines

This is a rather unexplored concept. Most reported investigations inject ethanol into the intake air manifold or the intake port. The fuel is ignited through direct injection of diesel followed by compression ignition. Investigations have shown prosperous engine operation with up to about 60% diesel substitution [2-3]. Fuel consumption and particulate emissions were seen to decrease, whereas NOx emissions were inconclusive.

The concept was not investigated further in this investigation.

References

[1]   Abu Qudais, M et. al. "The effect of alcohol fumigation on diesel engine performance and emissions. Energy Conversion Management, vol 41, pp 389-99, 2000.

[2]   Rezende, H et. al. "Ethanol diesel truck engine" SAE Technical paper series, paper no. 2011-36-0319, 2011.

[3]   Padala, S et. al. "Ethanol utilization in a diesel engine using dual-fueling technology". Fuel, vol. 109, pp 597-607, 2013.

Part A:

Testing of various fuel and additive options in a compression-ignited heavy-duty alcohol engine

Report from VTT, Finland, presented at "The 21st International Symposium on Alcohol Fuels – 21st ISAF"

Authors: Nils-Olof Nylund, Timo Murtonen, Mårten Westerholm, Christer Söderström, Timo Huhtisaari, Gurpreet Singh.

Abstract

The objective of this study was to evaluate the effects of fuel composition and additive options on the performance of a compression-ignited heavy-duty alcohol engine. All in all the testing shows that the direct injection ethanol engine concept has some built-in multifuel capabilities.

The IEA Implementing Agreement on Advanced Motor Fuels has initiated an activity on alcohols fuels for diesel engines. Two research partners, DTU of Denmark and VTT of Finland teamed up in this activity.

VTT carried out measurements with a Scania DC9 E02 270 Euro V/EEV certified ethanol engine installed in an engine dynamometer. VTT's test programme comprised a comprehensive fuel matrix with alternative additive packages, varying water content of the fuel and varying additive dosing. A combination of indirect (intake manifold) and direct fuel injection was tested as well. In addition, some tests were carried out with methanol based fuels.

The most important observation was that engine operation was rather

insensitive to fuel composition, and in fact the engine operated in practice normally on all other fuels except on pure methanol. The combination of indirect and direct fuel injection turned out to be an interesting option. Injecting a limited amount of fuel into the intake manifold starts some pre-reactions and facilitates ignition of the main fuel shot.

1. Introduction

The International Energy Agency Implementing Agreement on Advanced Motor Fuels has initiated an activity, Annex 46, on alcohols fuels for diesel engines. The goal is to report the best possibilities for implementation of alcohols in diesel engines. One of the main objectives of the project is to secure the supply of fuels for diesel engines, in this case by focusing on sustainable biofuels in the form of alcohols. Two research partners, DTU Technical University of Denmark and VTT of Finland teamed up in this activity, with technical support from Scania. DTU and VTT have their respective research agendas. DTU is carrying out work with an experimental engine, whereas VTT has conducted work using a commercial Scania heavy-duty ethanol engine.

Especially in Europe, there is a shortage of middle distillates. The commercial vehicles are in practise running on diesel fuel only, and diesel fuelled passenger cars have become increasingly popular. The demand of aviation kerosene is increasing, as well as the demand of distillate fuels in the marine sector, due to the new limits on sulphur dioxide emissions. Fig. 1 shows the need to find diesel substitutes.

Substitution of fossile diesel oil is crucial!

Threats	Solutions
• Already shortage of Diesel oil in Europe	– Biofuels ought to be be used in heavy vehicles
• Higher share of dieselcars due to regulations of fuel consumption.	– Biomethane
	– Synthetic diesel oil
	– Dimethyleter (DME)
• Aviation fuels and diesel oil are competitors	
• Environmental marine regulations will raise demand for diesel oil?	– Electrification of roads….

Fig. 1 The need for diesel substitutes [1]

Ethanol is the most widely used biofuel, and also the most widely used alternative fuel in the world. IEA estimates that in 2010, the share of ethanol within biofuels was more than 80 % on the world level [2]. Aakko-Saksa estimated that in 2011, the world total use of alternative fuels was some 145 Mtoe, of which some 45 Mtoe was ethanol and some 40 Mtoe natural gas [3].

The greater part of the ethanol is used for low-level blending into petrol. Using ethanol in diesel engines would bring about two major benefits:

- Alleviate the shortage of middle distillates
- Enable the use of ethanol with high engine efficiency

However, ethanol as such is not suited as fuel for conventional diesel engines, as the ignitability from compression only is low. In principle this means that either the engine or the fuel has to be modified. In the 1980s and 1990s there were

some projects on direct-injection alcohol engines with ignition aid either by diesel pilot injection or by glow plugs. Detroit Diesel had, for a short while, a two-stroke glow-plug assisted alcohol engine available [4].

The only concept that has reached commercial maturity is Scania's technology with additive treated ethanol. Ethanol buses manufactured by Scania have been in operation in Swedish cities since 1989. More than 600 buses have been supplied. Stockholm Public Transport (SL) decided as early as the mid-1980s to start replacing its diesel buses with buses running on renewable fuels on the inner-city lines. Today, ethanol buses complemented with some biogas buses are used on all inner-city routes, and diesel technology is no more in use.

The ethanol engine is an adaptation of Scania's 9-litre diesel engine. The ethanol version features, among other things, elevated compression ratio (28:1) to facilitate ignition, higher fuel delivery to compensate lower energy density of the fuel, and special materials for the fuel system [5]. Now a Euro VI certified version of the engine is available [6].

Unlike gas engines, the ethanol engine delivers diesel-like efficiency. This has been verified by VTT in earlier measurements [7].

There are currently five heavy-duty ethanol vehicles running in Finland, three trucks and two buses. Figure 2 shows one of the test buses on VTT's heavy-duty chassis dynamometer. The Finnish company St1 is producing ethanol from waste. St1 and North European Oil Trade (NEOT), partly owned by St1, are interested in promoting diesel ethanol technology. This is why these companies became sponsors of the Scania ethanol engine testing at VTT. The rationale of the testing was to test different additive packages and evaluate ways to reduce the cost for the additives needed for diesel ethanol.

One idea was to inject a small amount of the fuel into the intake manifold to enhance the ignition of the main fuel shot injected directly into the combustion

chamber. The combustion system using a combination of indirect and direct fuel injection is often called partially pre-mixed combustion (PPC). This concept has been researched by, e.g., University of Lund in Sweden [8]. PPC is seen as a way to improve fuel efficiency of petrol engines or a way to reduce emissions of diesel engines. Now PPC was tested as a possible way to reduce the amount of ignition improver in diesel ethanol. It is important to notice that the same fuel is injected both into the intake and into the combustion chamber, no additional fluid is needed.

Fig. 2 One of the field test vehicles on VTT's heavy-duty chassis dynamometer

2. Experimental Apparatus

The fuel and additive testing was carried out in VTT's engine laboratory with a commercial Scania ethanol engine, corresponding to the engines of the five fleet tests vehicles. Data for the Euro V/EEV certified engine is given in Table 1.

Table 1 Data for the Scania ethanol engine used for testing at VTT [5]

Emission level	Euro V & EEV
Configuration	Charge-cooled in-line 5-cylinder 4-valve cylinder heads Unit injectors, EGR
Displacement	8.9 litres
Comp. ratio	28:1
Power	199 kW (270 hp) at 1900 rpm
Torque	1200 Nm at 1100 – 1400 rpm

One cylinder was equipped with a pressure transducer to measure cylinder pressure. Start and duration of injection was determined from the control signal of the unit injector type fuel nozzle. In addition, a system enabling fuel injection into the individual intake ducts was added to the engine. The system comprised five fuel injectors, a fuel pump and a simple control system.

The engine is normally equipped with an oxidation catalyst, but to get a

better understanding of how fuel affects engine-out emission performance, all tests were run without catalyst.

The engine was installed in a test cell with a steady-state engine dynamometer and equipment for exhaust emission measurements. The instrumentation is presented in Table 2. For emission measurements, the apparatus corresponds to the requirements of the European Directive 1999/96/EC on emission measurements of heavy-duty engines. However, as a steady-state engine dynamometer was used, the measurements were basically carried out using the European Steady Cycle (ESC) procedure. Emissions of unburned alcohol and aldehydes were measured using an FTIR instrument.

Table 2 Instrumentation

Parameter/equipment	Instrument
Engine dynamometer	Eddy current Froude Consine AG 400
Fuel balance	AVL FB 733
Cylinder pressure indication system	AVL INDICOM, Kistler transducers
Exhaust flow measurement system	Full-flow Pierburg CVS 120 WT
Exhaust gas analyser system, regulated components	Pierburg AMA 4000
Instrument for measurement of alcohols and aldehydes	Gasmet FTIR CR-2000
Intake manifold injection system	Hestec Harinen 32 ECU, Bosch injectors

Fig. 3 shows the engine in the test cell, Fig. 4 the additional fuel injectors installed in the intake manifold and Fig. 5 the schematics of the additional fuel injection system.
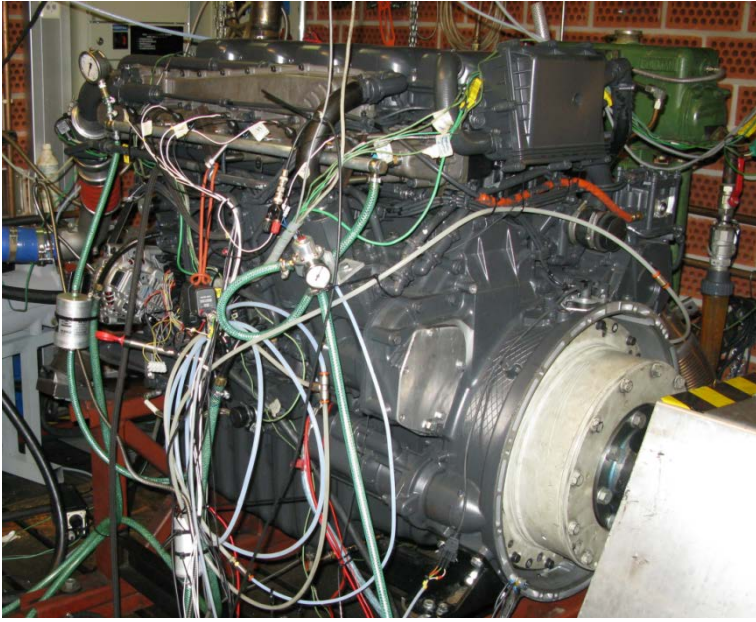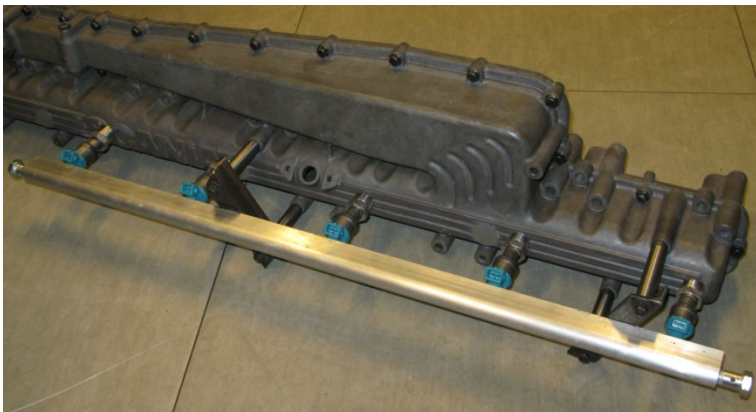
Fig. 3 The engine test cell

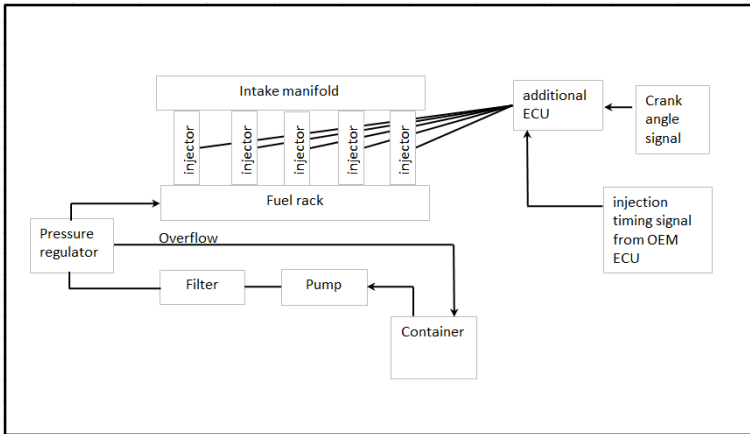**Fig. 4 The additional fuel injectors and the associated fuel rail**

Fig. 5 Schematics of the additional fuel injection system

3. Test programme

The objectives of the test programme were to test:

- Three alternative additive packages, all approved by Scania
- Three different ethanol water concentrations (appr. 0, 5 and 10 % water by weight)
- Two fuels containing methanol; one blend of ethanol and methanol and neat methanol
- Whether injection of fuel into the manifold would facilitate ignition on the main fuel shot

Here it must be pointed out that Scania doesn't approve the use of methanol containing fuels in its ethanol engine.

Performance indicators monitored included, among other things:

- Energy consumption
- Carbon dioxide (CO$_2$) emission
- Regulated emissions (carbon monoxide CO, unburned hydrocarbons

HC[1], nitrogen oxides NO_x, particulate matter PM)
- Unregulated emissions (unburned alcohol, aldehydes)
- Ignition delay and heat release

The emission measurements were carried out using the ESC cycle (Fig. 6). The same load points were used for fuel consumption and cylinder pressure measurements as well, but the running times for each point were, due to the nature of these measurements, longer than in the standardised procedure. In addition to ESC load modes, cylinder pressure and fuel consumption were measured on 10 % engine load at the engine speeds of the ESC cycle. The combination of low load and high engine speed is the most critical for fuel ignition, and using such a combination for testing should accentuate differences between additives or fuels.

Master batches from three additive manufactures were used for blending the "baseline" diesel ethanol fuels test fuels 1, 2 and 3. Blending was done according to the manufacturers' instructions, and the blended fuels were analysed for certain physical and chemical properties. The additive package of fuel 1 was used for blending the fuels 4 and 5 varying the water content, and also for the fuels containing methanol, fuels 6 and 7. The ESC testing was repeated four times with fuel one, which was used as a reference fuel, two measurements to start with, and two measurements at the end of the test matrix to check for possible drift in the performance of the engine. The ESC tests were repeated twice with the other fuels.

The tests with fuel 7 (methanol) are not fully compatible with the other tests, as the engine could not reach full torque due to the low volumetric heating value of methanol. Table 3 presents analysis of some fuel properties.

---

[1] When running on alcohols, the hydrocarbon result is not accurate. However, the HC values can with caution be used for fuel to fuel comparisons.
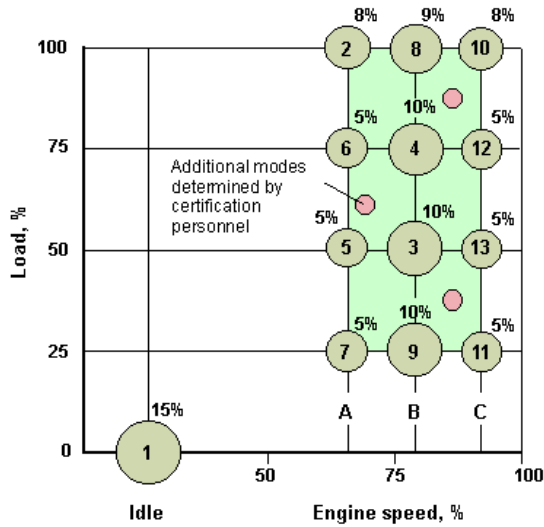
Fig. 6 The European Stationary Cycle (ESC). www.dieselnet.com

Mode	Engine Speed	Load, %	Weight, %	Duration
1	Low idle	0	15	4 minutes
2	A	100	8	2 minutes
3	B	50	10	2 minutes
4	B	75	10	2 minutes
5	A	50	5	2 minutes
6	A	75	5	2 minutes
7	A	25	5	2 minutes
8	B	100	9	2 minutes
9	B	25	10	2 minutes
10	C	100	8	2 minutes
11	C	25	5	2 minutes
12	C	75	5	2 minutes
13	C	50	5	2 minutes

Table 3 Analysis of the test fuels

20

	Density	Ethanol	H2O	Methanol	Carbon	Hydrogen	LHV
	kg/m3	mass-%	mass-%	mass-%	mass-%	mass-%	MJ/kg
Fuel 1	819	88,9	5,7	0,1	48,8	13,1	24,9
Fuel 2	814	90,5	5,5	0,5	47,8	13,3	24,7
Fuel 3	819	88,7	5,8	0,5	47,2	12,6	25,0
Fuel 4	804	92,3	0,4	0,1	49,6	12,5	26,8
Fuel 5	832	84,0	10,1	0,1	45,1	12,4	23,9
Fuel 6	820	57,0	5,1	28,7	43,2	12,3	23,2
Fuel 7	808	3,2	0,3	87,9	37,7	12,1	20,4

The intake manifold injection was tested in two load points: 1800 rpm at 10 and 25 % load. The fuel used in these tests was basically fuel 1. However, for these tests the additive package was dosed at 100, 50 and 25 % of the normal concentration.

4. Results

The following results will be presented:
- Energy consumption, Fig. 7 (ESC)
- CO_2 emission, Fig. 8 (ESC)
- Regulated emissions, CO, HC, NO_x and PM, Figs 9, 10, 11 and 12 (ESC)
- Unregulated emissions, ethanol, methanol, acetaldehyde, Figs 12, 14 and 15 (ESC)
- Cylinder pressure data
- Effects of intake manifold injection

Figures 7 and 8 show only negligible differences between fuels, even for neat methanol (fuel 7). The energy consumption of 8.5 MJ/kWh translates into an efficiency of 42.5 % over the ESC cycle.

The gaseous regulated emissions behave more or less as expected, very little differences between fuels 1, 2 and 3. Methanol seems to decrease CO and HC emissions, but with neat methanol NO_x emissions increase. The effects of water are also as could be expected. In comparison with the reference fuel 1, leaving out the water increases both CO and NO_x emissions, whereas adding water reduces both these emissions marginally. The engine is certified for a NO_x level of 2 g/kWh, and the engine delivered exactly this NO_x level.

The results for particulate emissions are quite interesting. The differences between fuels 1 to 5 are rather marginal, but methanol, especially neat methanol increases PM emissions. However, as methanol contains no carbon-to-carbon bonds, it is hard to image that methanol would generate carbonous particles. The appearance of the test filters, which are quite white, supports this. The result must be an indication of semivolatile components or artifacts. Due to low volumetric heating value the end of injection is delayed compared to operation on ethanol, and this can be one contributing factor.
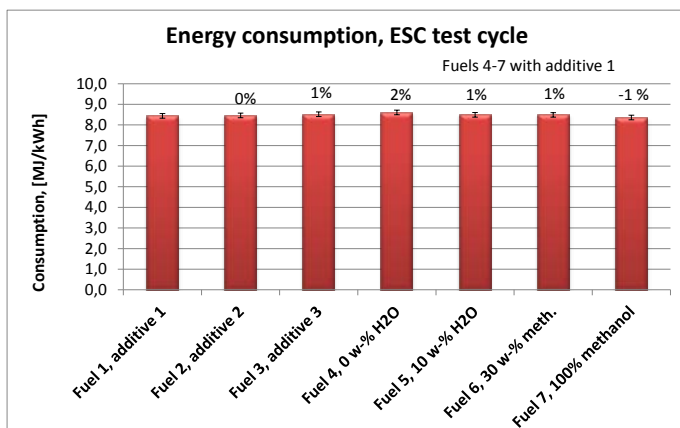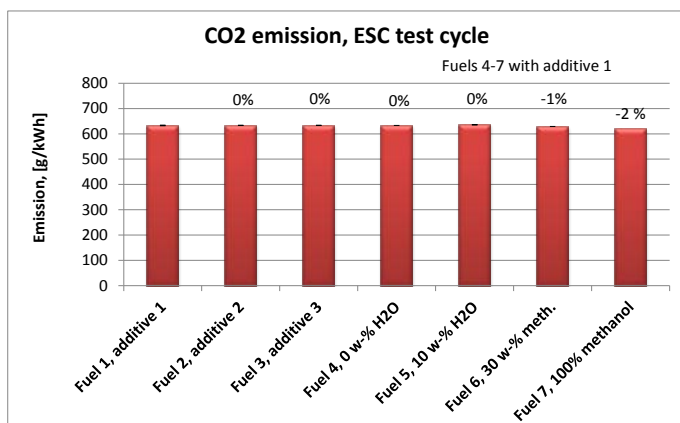
Fig. 7 Energy consumption

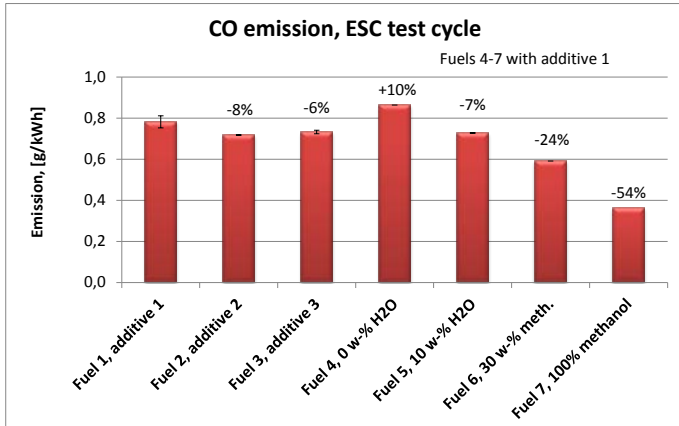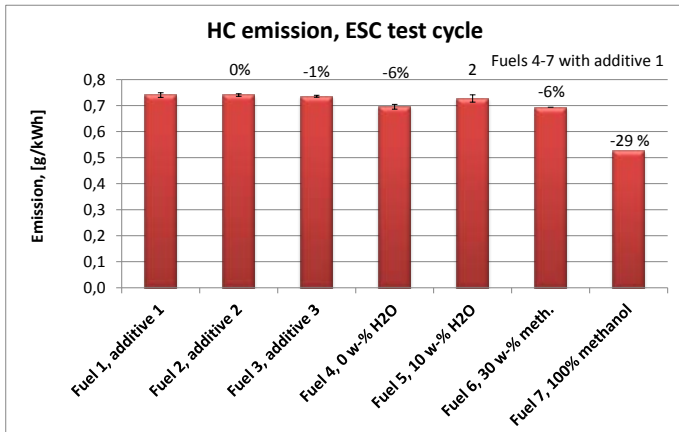**Fig. 8 CO_2 emission**

Fig. 9 CO emission
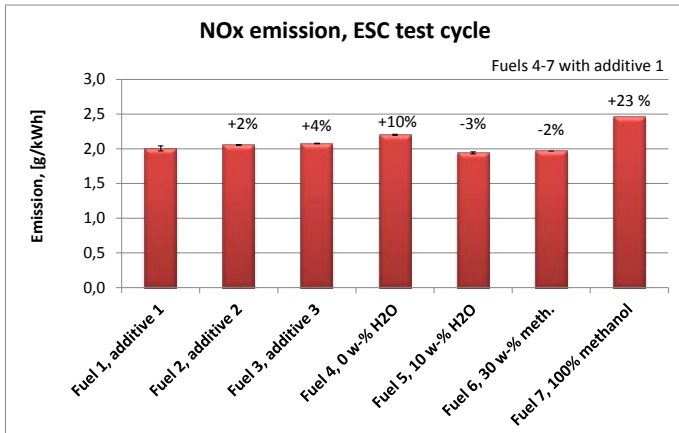

Fig. 10 HC emission

Fig. 11 NO$_x$ emission

Fig. 12 PM emission

25
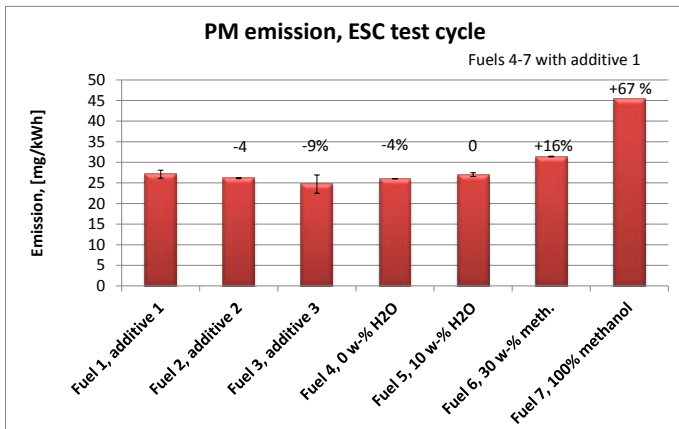
For fuels 1 – 5, HC and ethanol emissions follow exactly the same pattern, dry ethanol delivers somewhat lower emissions than the hydrous fuels. For fuels 1 – 5 ethanol emission is around 0.8 g/kWh, at the same level as the methanol emission with fuel 7.

Aldehyde concentrations were low, on an average clearly below 10 ppm, even though the engine was operated without a catalyst. The methodology used for analysing FTIR spectra has lower detection limit for acetaldehyde than for formaldehyde. Formaldehyde levels were around or below the detection limit. Therefore only acetaldehyde is shown in Fig. 15.
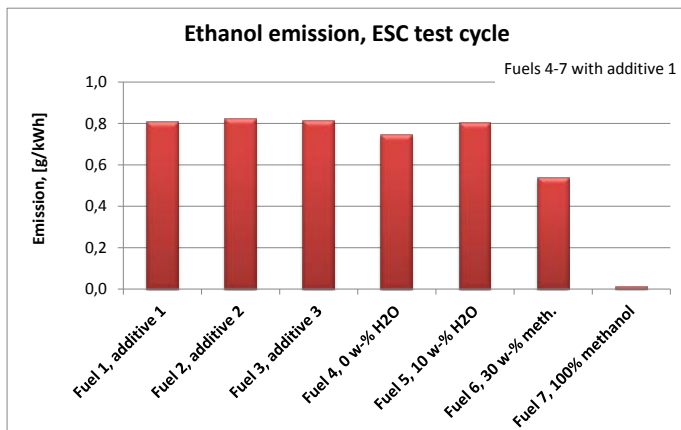
Fig. 13 Ethanol emission
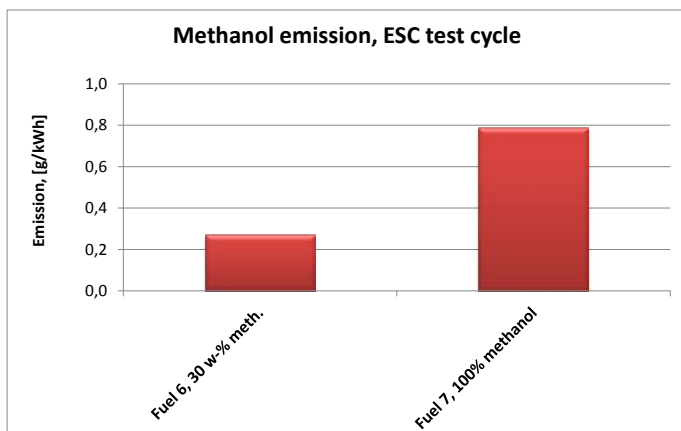
Fig. 14 Methanol emission
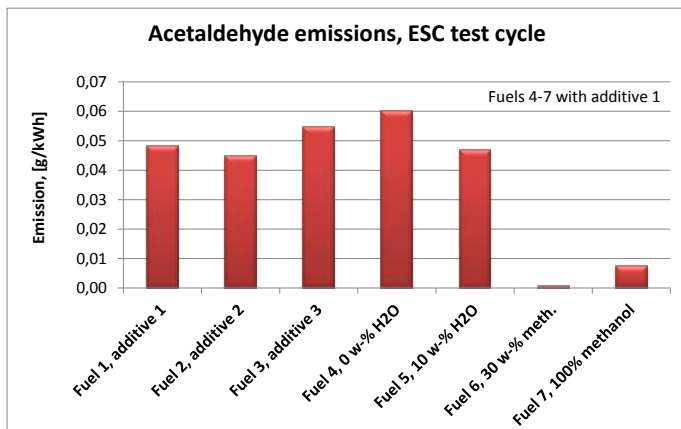
Fig. 15 Acetaldehyde emission

The normal dosing of ignition improver additive is sufficient for stable engine operation in all conditions. The cylinder pressure analyser system, in addition to recording cylinder pressure traces, calculates heat release. Start of injection is electronically controlled, and doesn't in this case vary with fuel. The crank position for 10 % accumulated heat release was used as a gauge for ignition properties. This position should be retarded using a fuel with poor ignition quality. Fig 16 shows crank position for 10 % of accumulated heat release in two load points, 1800 rpm at 10 (left) and 100 % (right) load.

1812 / 105

1812 / 1046

Fig. 16 Crank position (after top dead centre) for 10 % accumulated heat release,

1800 rpm at 10 and 100 % load

At 10 % load, differences between fuels 1 – 6 are negligible, only neat methanol (fuel 7) can be distinguished for retarded combustion, and even this difference is relatively small. At full load, even fuel 7 performed normally regarding heat release. Fuel 1 was tested both at the beginning and at the end of the programme, and Fig. 16 confirms that repeatability was good (first and last bars).

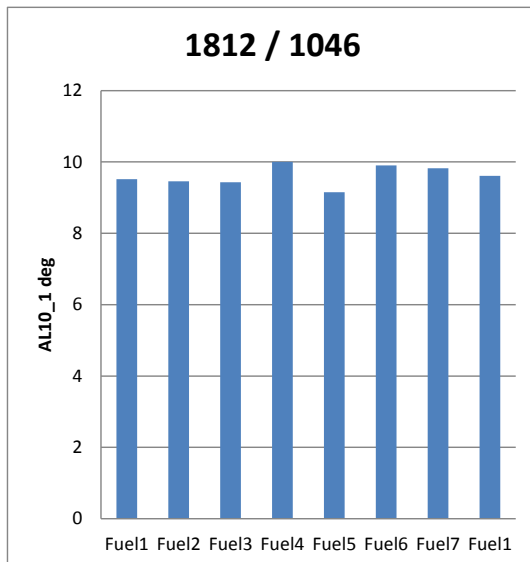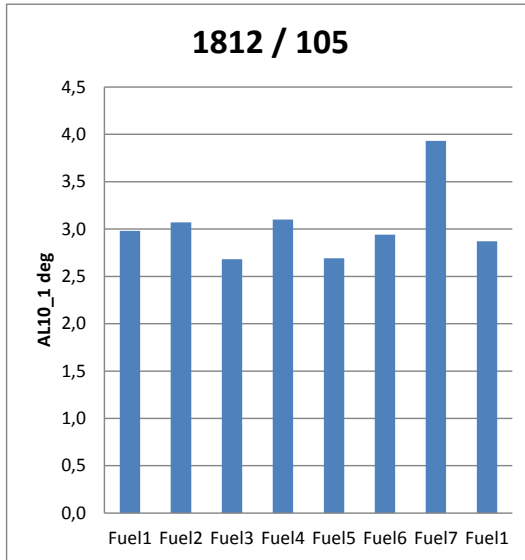The notion that additive dosing is more than sufficient, at least with a warmed-up engine, was accentuated when reducing fuel 1 additive concentration. With 50 % additive dosing the engine still operated normally in all load points. Only when reducing additive concentration to 25 %, abnormalities could be found in the cylinder pressure traces at low load.

Fig. 17 shows cylinder pressure traces using 25 % additive dosing. Load point is 1800 rpm and 25 % load. On the left is cylinder pressure trace without and on the right with intake manifold injection. On the left, one can see a discontinuity in the pressure trace, induced by extended ignition delay. On the right the pressure trace is smooth, corresponding to operation on full additive dosing.

Fig. 18 shows crank position for 10 % accumulated heat release as a function of additive dosing with and without intake manifold injection. The base case (5 % additive) corresponds to the left part of Pic. 16, with 10 % heat release accumulated at some 3 $^\circ$CA after top dead centre. When reducing additive dosing to 25 %, location of the point of 10 % heat release is retarded to some 5 – 6 $^\circ$CA after top dead centre without intake manifold injection. With intake manifold injection, the location position of 10 % heat release is stable at around 1 – 2 $^\circ$CA. The amount of fuel injected into the intake was

some 25 % of the total amount of fuel. The system and the calibration were by no means optimised, so the intake manifold resulted in a fuel consumption penalty of some 10 %
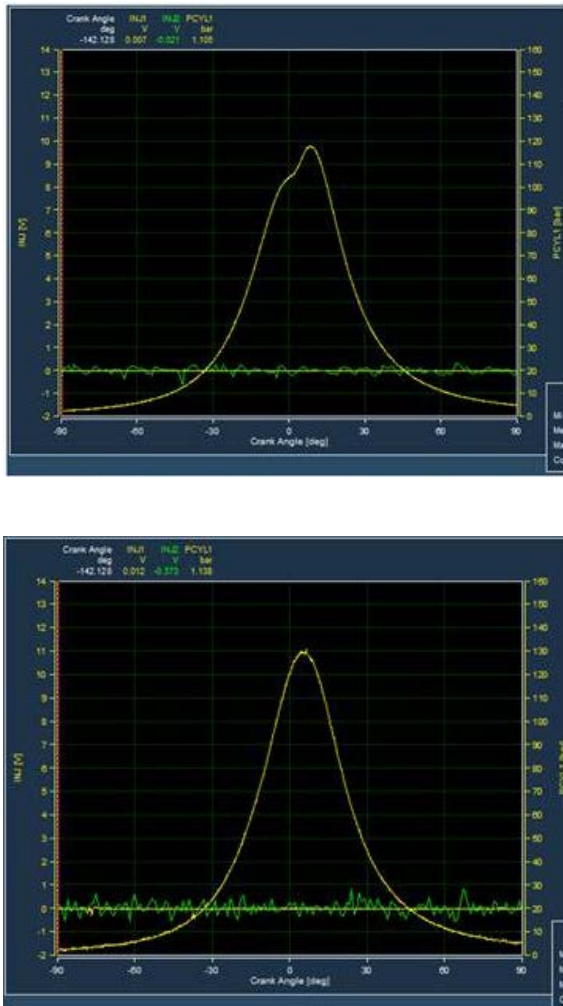
Fig. 17 Cylinder pressure traces without (left) and with (right) intake manifold injection.

Load point is 1800 rpm and 25 % load

31

Crank angle location for 10% heat release value

—◆— with add. fuel inj   —■— without add. fuel inj.

Loadpoint: 1812 rpm/105Nm (10% load)
Amount of additional injected fuel : 25% of total amount
   -> 10% increase on fuel consumption

Crank angle ATDC, deg

5% additive   2.5% additive   1.2% additive

Fig. 18 Crank position for 10 % accumulated heat release as a function of additive dosing

with and without intake manifold injection

5. Discussion

Scania's ethanol engine delivers what it is supposed to deliver, diesel-like efficiency (42.5 %) and a NO_x level of 2.0 g/kWh, when running on fuels that the engine is designed for.

In addition, the testing demonstrated that the direct injection ethanol engine concept has some built-in multi-fuel capabilities. Even with standard calibration the engine operated quite well with additive treated neat methanol. However, with neat methanol the engine didn't reach full power, and the injection periods on partial load were prolonged compared to ethanol operation. Methanol tends to increase PM emissions. The result must be an indication of semi volatile components or artifacts with methanol, as methanol contains no soot forming carbon-to-carbon bonds. The standard oxidation catalyst most probably would have reduced PM mass with methanol.

A blend of 70 % ethanol and 30 % methanol delivered lower CO, HC and NO_x emissions compared to the baseline ethanol fuel. However, it must be stated that Scania doesn't approve the use of methanol fuels.
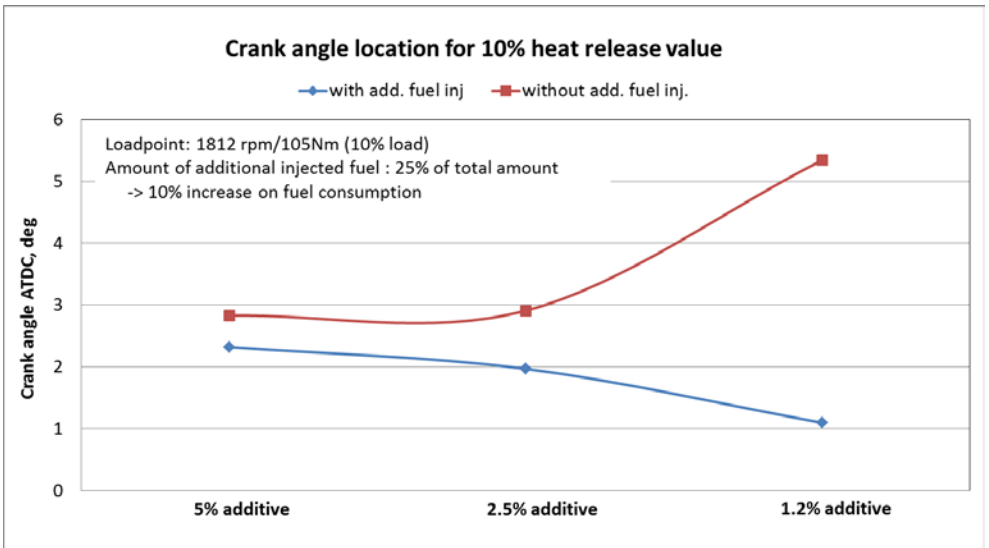
The engine was tested on three different additive packages for ethanol. Additive had negligible effects on engine operation. The emissions, cylinder pressure and fuel consumption results were equal with all three fuels. Some minor differences in PM and CO emissions were detected. Those differences can be considered insignificant when taking into account the statistical deviation from test to test. The results of the cylinder pressure measurements (e.g. start of heat release, pressure gradient, maximum pressure) were consistent for each fuel. The testing demonstrated that the regular additive dosing is more than sufficient, the warmed-up well on half

of the regular dosing. However, a significant reduction of additive dosing might jeopardize cold starting.

The effects of fuel water content were what could be expected. In comparison with the baseline hydrous fuel, leaving out the water increases both CO and NO_x emissions, whereas adding water reduces both these emissions marginally. With the introduction of Euro VI emission regulations and selective catalytic reduction (SCR) for NO_x control the role water for NO_x suppression will diminish.

The need for ignition improved additive can be reduced by injecting part on the fuel into the intake manifold. The premixed fuel facilitates ignition. Using intake manifold injection normal combustion (ignition delay and rate of heat release) could be achieved with only 25 % of the standard additive dosing. However, intake manifold injection increased overall fuel consumption. Further testing to optimize, e.g., amount of pilot fuel and timing of main fuel injection, is needed to really show the potential of the concept. One way to take advantage of this concept would be to keep the dosing of additive as it is, but to reduce the compression ratio, thus lowering both mechanical stresses and costs for the engine.

The Scania ethanol engine used for the testing was equipped with unit injectors. A common-rail fuel system would enable pre-injection without any additional hardware.

6. Acknowledgements

support from North European Oil Trade NEOT and St1.

References:

[1] Hådell, O. (2012). Alternative energy and new technology for heavy-duty vehicles. TransEco seminar "Advanced bus technologies and alternative fuels". Helsinki, 6 February 2012.
http://www.transeco.fi/files/534/Alternative_Energy_and_New_Technology_for_Heavy-duty_Vehicles_Olle_Hadell.pdf

[2] Technology Roadmap: Biofuels for Transport. International Energy Agency. Paris, April   2011.
http://www.iea.org/publications/freepublications/publication/Biofuels_Roadmap_WEB.pdf

[3] IEA-Advanced Motor Fuels. Annual Report 2011.
http://www.iea-amf.org/app/webroot/files/file/Annual%20Reports/Annual%20Report%202011%20Final_.pdf

[4]   Toepel, R., Bennethum, J., and Heruth, R., "Development of Detroit Diesel Allison 6V-92TA Methanol Fueled Coach Engine," SAE Technical Paper 831744, 1983, doi:10.4271/831744.

[5] Scania continues renewable fuel drive. New highly efficient diesel-ethanol engine – ready to cut fossil CO2 emissions by 90%. Press Info 21.5.2007.
http://www.scania.com/Images/P07503EN%20New%20ethanol%20engine_tcm10-163550_81325.pdf

[6] Scania Buses & Coaches. Engine range. Euro 6.
http://www.scania.com/products-services/buses-coaches/Engine-range/Euro%206.aspx

[7] Nylund, N.-O., Laurikko, J., Laine, P., Suominen, J. & Anttonen, M.P.A. (2012). Benchmarking heavy-duty ethanol vehicles against diesel and CNG vehicles. Biomass Conv. Bioref. DOI 10.1007/s13399-012-0049-z

[8] Tunér, M. (2014). State of the art of Partially Premixed Combustion (PPC). CRC Workshop on Advanced Fuel and Engine Efficiency. Hyatt Regency Baltimore. Baltimore, Maryland, February 24-26, 2014

Part B:

Detailed investigations of alcohol fuels in diesel-type combustion

Report from the Technical University of Denmark (DTU), presented at "The 21ˢᵗ International Symposium on Alcohol Fuels – 21ˢᵗ ISAF"

Authors: Jesper Schramm, Mads Carsten Jørgensen

Abstract

The International Energy Agency Implementing Agreement on Advanced Motor Fuels has initiated an activity, Annex 46, on alcohols fuels for diesel engines. The goal is to report the best possibilities for implementation of alcohols in diesel engines. One of the main objectives of the project is to secure the supply of fuels for diesel engines, in this way by focusing on sustainable biofuels in the form of alcohols. Two research partners, DTU of Denmark and VTT of Finland teamed up in this activity, with technical support from Scania. DTU and VTT have their respective research agendas. DTU is carrying out work with an experimental engine, whereas VTT has conducted work using a commercial Scania heavy-duty ethanol engine. Accordingly, two separate papers will be presented. The paper at hand describes the activities at DTU.

   At DTU, new ethanol fuel formulations and additive options has been studied in a two cylinder research engine equipped with pressure transducers to characterize the combustion processes, and equipment for

gaseous emission measurements. One of the cylinders was fueled by ordinary diesel fuel while the other cylinder was fueled by the fuels to be investigated. The test fuel matrix covered commercial diesel fuel, commercial E95 ethanol fuel for diesel engines and pure ethanol as reference fuels. Furthermore, two alternative additive added ethanol fuels were tested.

The investigation included analysis of the obtained experimental data in order to optimize the fuel conversion with respect to fuel economy and emissions. To characterize the best possible operation area, it is essential to characterize and understand the spray formation and heat release pattern during engine operation for different fuels/additives. This paper gives an evaluation of the possibilities and limitations of operating a diesel engine on ethanol-based fuels based on classical engine parameter analysis.

Objective

The Advanced Motor Fuels Implementing Agreement under the international Energy Agency initiated in 2013 a project with the purpose to investigate the possibilities to apply alcohols for compression ignition engines (diesel engines). The main purpose was to test a commercially available fuel additive package together with two new additive packages and carry out a comparison of the performance of these additives. The additives are designed for application together with neat ethanol in order to operate in a conventional diesel engine. The project was split up into two activities: experimental work with a commercial Scania engine, which took place at VTT, Finland and experimental work on a research engine at DTU, Denmark. The results of the findings at VTT are reported elsewhere [1]. This paper describes the result of the work at DTU.

Introduction

In Europe, Directive 2009/28/EC on renewable energy sets the transport bioenergy obligation in 2020 at a minimum of 10% of the transport energy use. Modern spark-ignition (SI) vehicles are compatible with 10% ethanol in gasoline (E10), which represents 6% of the bioenergy content. Since a higher ethanol content can be used only with a limited car population, ethanol use is limited — even if higher amounts are commercially available (the so-called "blending wall").

Fuel economy is an increasingly important current issue. An obvious goal is to achieve efficiencies similar to diesel engines with the alcohol applications. However, the application of alcohols in a diesel engine requires a fuel additive to ignite the unburned mixture. An option is to use ethanol together with additives in heavy-duty ethanol diesel engines, which are now manufactured by Scania. These engines run on a fuel that consists of 95% hydrous ethanol together with an ignition improver, a corrosion inhibitor, and denaturants .

With this concept, relatively small modifications are required in the engine. The compression ratio is increased, and the fuel system is modified. The exhaust catalyst is developed to prevent excessive aldehyde emissions. This concept, for example, is used in buses in Stockholm.

Particularly in Europe the need for middle distillates is increasing. This calls for new diesel fuels, and contribution of alcohols to the engines that are normally fueled by diesel is one of the options that seem possible to explore.

The most interesting option would be a "flex fuel" diesel engine that can run with both ethanol and diesel fuel without pilot injection technology. Engines that can use only ethanol would be suitable only in restricted areas, where availability of fuel can be controlled. Diesel engines and their control technologies have advanced considerably in recent years. For example, the common-rail system enables fine adjustments of injection. Ethanol diesel engines could be used in road transport, machinery, and the marine sector — especially in countries where ethanol is produced on a large scale.

Alcohols represent superior fuels for the SI engine with respect to key properties, such as octane number and latent heat of vaporization. Basically, alcohols can withstand high pressures and temperatures without igniting uncontrollably. In many parts of the world, ethanol is widely used in low concentration blends with gasoline, and it has a more limited use in high concentration blends. In the case of low-ethanol blends (E5-10), it is possible to produce fuels with a slightly higher or similar octane number compared to that for regular gasoline. Most modern cars are able to regulate the ignition timing and advance the timing to a degree that increases engine efficiency by a few percentage points. A high share of ethanol, up to 85%, can be used in special SI flexible-fuel vehicles.

Ethanol application in diesel engines has been studied seriously since the 1980s.The application can be divided into three main categories [2]:

1. Ethanol and diesel oil blends, emulsions and solutions
2. Neat ethanol using spark ignition, glow plug or ethane improving additives

3. Separate ethanol injection, dual fuel injection or fumigation

Ethanol diesel blends (E-diesel)

Most commonly fuels tested are blends containing 10-20% ethanol. Recent investigations indicate that this percentage can be increased when applying biodiesel together with conventional diesel oil [2]. Looking at the literature there seems to be a definite potential for improving the efficiency of CI engines by blending ethanol into diesel fuel [3-9]. The fuel system needs to be designed to a higher fuel flow due to the lower calorific value of ethanol. Power and torque, however, seems to be increasing when the fuel system is redesigned [10]. Emissions of PM can be reduced significantly with E-diesel [5, 8-9], whereas other regulated emissions seems to be unchanged [11].

Neat Ethanol

Compression ignition of neat ethanol requires a very high compression ratio, unless heat and/or an ignition improver is added to the fuel. Application of ignition improvers has been demonstrated to work properly in buses in Sweden [12]. These busses apply Scania engine technology. The advantage with this form of application is that the engine technology is very close to existing diesel engine technology. The main changes needed lies in the additive package of the fuel and the adjustment of the fuel system to the lower calorific value and some material compatibility problems. In recent years new additive packages have entered the market, and the price of the additives are therefore expected to decrease. The present paper describes the results of applying three different additive packages in a dedicated Scania production engine.

Another obvious advantage is the possibility to run the engine stoichiometric so that a three-way catalyst can be applied to lower the emissions of CO, HC and NOx. PM emissions are low from the spark ignited combustion itself [2].

Dual Fuel

Although the concept of having two systems handling two different fuels can seem unrealistic for widespread commercial use, it still offers advantages over less complicated solutions. The one major disadvantage for the end user is the need to fill two separate fuel tanks. However, a relatively lower fuel price on ethanol could probably motivate car users to use it anyway, especially if the result is also increased fuel efficiency. Lubrication additives and/or improved materials might be needed for this technique. The main advantages are high engine efficiencies, high displacement of fossil diesel, and low NOx and PM emissions.

Alcohols, particularly those produced from biomass, are the obvious fuels for more intense combustion engine applications in the near future. Therefore, it is relevant to study on the best way to produce alcohols from biomass. The goal is to combine good fuel economy with low emissions. Many countries are interested in fuel/additive development. In addition, there are ideas for new combustion schemes (e.g., reactant controlled combustion), which could eliminate the need for the dedicated ignition improver additive [2].

The Test Engine

The engine used to test the different fuels was a 4-stroke diesel engine with two cylinders. One of the cylinders was modified to work on ethanol with

ignition improvers. The compression ratio was thus increased from 18 to 24,5. The compression ratio was chosen as high as possible in order to relate the results to the experimental work with the highly compressed Scania engine design at VTT [1]. The fuel injection system was modified to work on ethanol. The engine was installed with a piezo electric fuel nozzle, and the fuel pressure was supplied from a nitrogen pressurized fuel tank.

The engine was equipped with a pressure pick-up in order to measure the in-cylinder pressure during an engine cycle. Heat release analysis was carried out from heat release calculations based on the pressure traces from the usual rewriting of the first law of thermodynamics:

$$\dot{Q}_{net} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} \qquad [1]$$

The intake system was heated electrically in order to keep a constant intake temperature (80°C), high enough to ensure proper ignition of pure ethanol without additive packages. The emission measurements were carried out with an FTIR analyzer.

The engine data are summarized in Table 1, and the engine setup is shown in Figure 1.

Table 1. Test engine specifications

BUKH DV24 Test Engine
One standard diesel cylinder:
Compression ratio   18
Displacement volume 482 cm^3
One test cylinder:
Compression ratio
Geometric   24.5
Effective   21.3
Displacement volume 482 cm^3
Piezo electric injector
Injection pressure   150 Bar
Up to 5 injection pr. cycle
Changeable piston top
Intake air heater
Oil heater
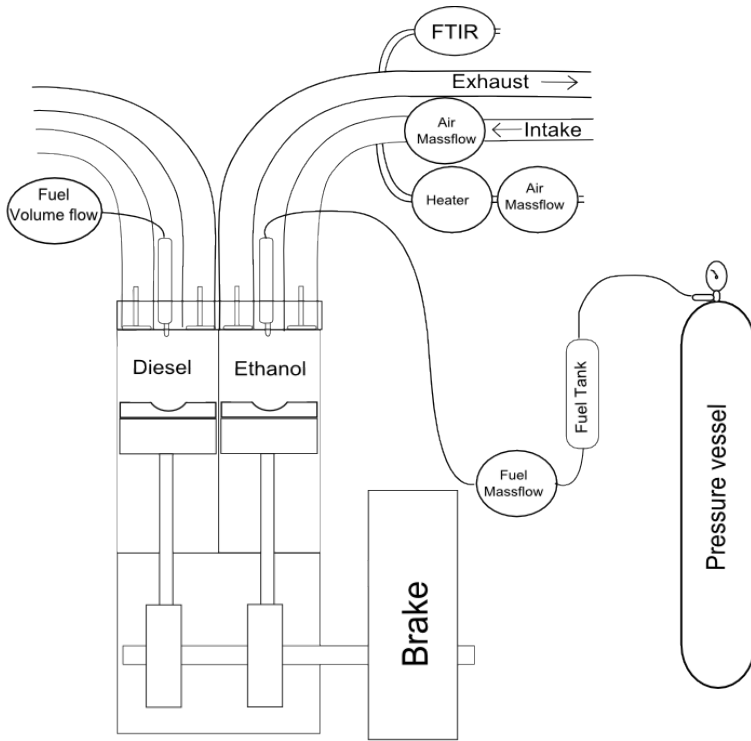
Figure 1. Test engine setup

Fuels and Test Conditions

The fuels and the test conditions are summarized in Table 2. Additive 1 is an additive package that is known to work well in a diesel engine. Additive 2 and 3 are alternative additives. The engine was operated at steady state low load conditions. Two different combustion principles were achieved: HCCI (homogeneous charge compression ignition) by injecting the fuel very early, allowing the fuel and air to mix completely before ignition, and PPCI (partially premixed compression ignition) where injection is started

relatively late during compression, allowing only a part of the mixture to mix homogeneously before ignition.

Table 2. Fuels and test conditions

	Engine operating conditions:
	Excess air ratio (λ):
	3.5 (low load)
	Engine speed:
	1200 rpm
	Intake temperature:
	80 ˚C
	Start of injection (SOI):
	360 CAD BTDC (HCCI) and 24 CAD
	BTDC until misfire (PPCI)
Fuel 0	Neat ethanol with 5,5% water
Fuel A	Ethanol + additive 1 and 5,5% water
Fuel B	Ethanol + additive 2 and 5,5% water
Fuel C	Ethanol + additive 3 and 5,5% water

Results

The heat release analysis is based on equation [1]. In this equation the term $\dot{Q}_{net}$ is the heat release from the fuel corrected for heat losses like losses to the walls and through fuel blow-by past the piston. The term *apparent heat release* , $\dot{Q}_{app}$, is then defined from the equation:

46

$$\dot{Q}_{net} = \dot{Q}_{app} + \dot{Q}_{loss} \qquad\qquad [2]$$

The apparent heat release then represents the actual heat, or energy, released from the fuel during the combustion. In the following the rate of net heat release (NRHR) refers to the net heat release as defined in equation [2].

PPCI operation

The NRHR for different injection timings are shown in Figure 2. The accumulated net rate of heat release values, ANRHR, are shown in Figure 3, relative to the total energy contained in the fuel per cycle, Q_{cycle}. This net efficiency is thus defined as:

$$\eta_{net} = \frac{ANRHR}{Q_{cycle}} \qquad\qquad [3]$$

With this definition the efficiency becomes a function of the CAD. The final value $\eta_{net, final}$ is then the net efficiency of the whole cycle.
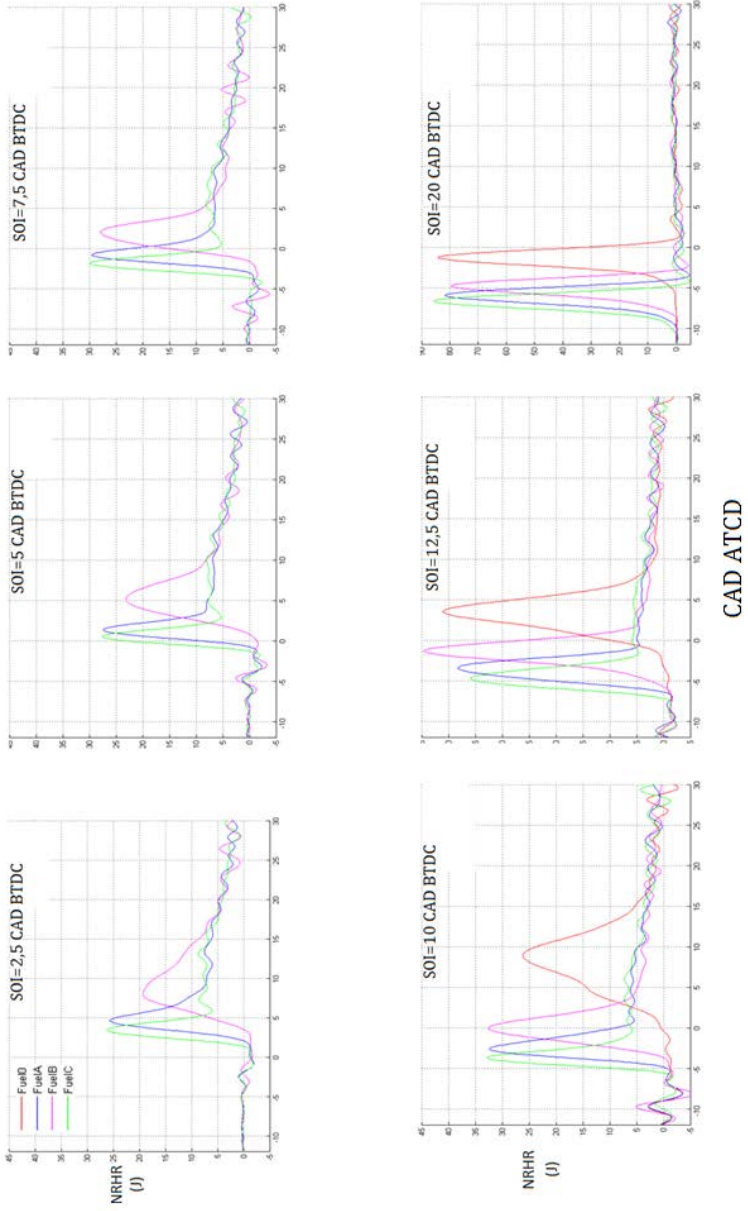
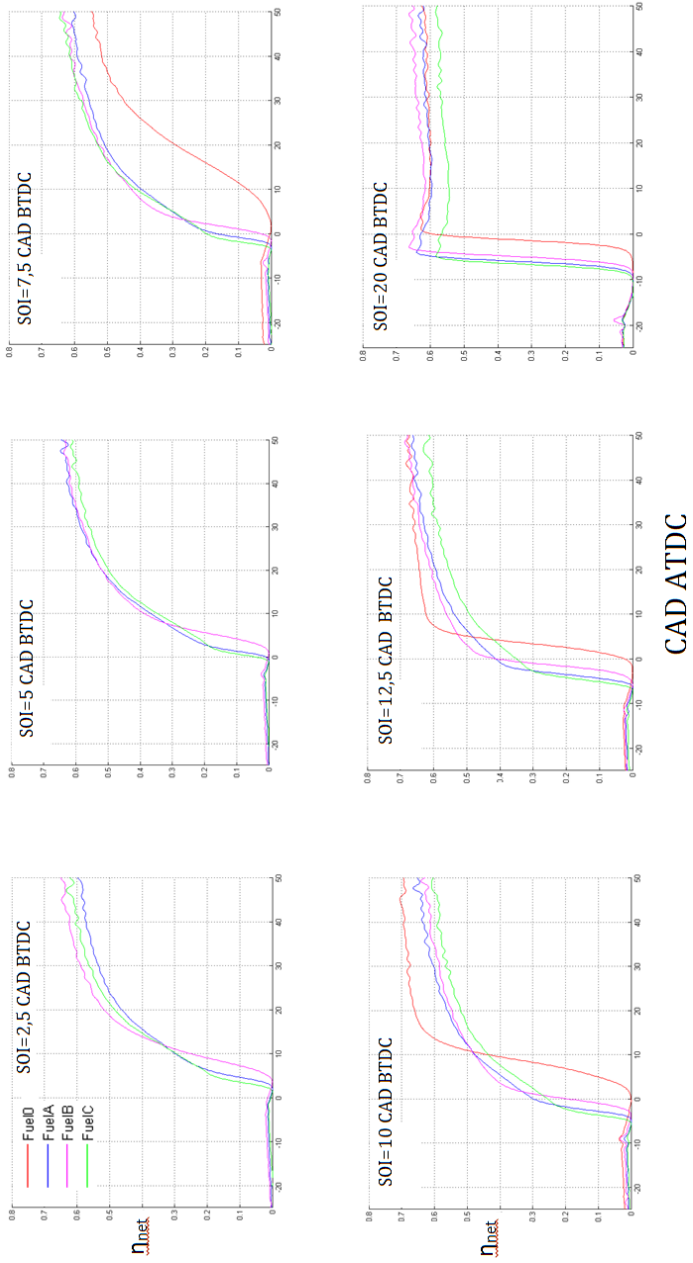**Figure 2. Net rate of heat release at different SOI.**

48

Figure 3. Net efficiency at different SOI.

49

From Figure 2 it is clearly seen that fuel C ignites faster than the rest of the fuels. The order of ignition delay is: fuel C< fuels A< fuel B< neat ethanol. This is the case for al SOI. The ignition delay is shown in Figure 4 as a function of SOI. From Figure 3 it is seen that neat ethanol does not burn properly at SOI later than 7.5 CAD BTDC. At SOI 5 CAD BTDC and later neat ethanol does not burn at all. It is also obvious, as expected, that earlier SOI is followed by a larger portion of pre-mixed combustion. This has the effect of increasing maximum temperatures and therefore also increasing of NOx emissions as seen in Figure 5.

From Figure 3 it is also seen that the best value of η_{net} is found with neat ethanol and SOI 10 CAD BTDC. One should, however, remember that the effect of blow-by influences these results. The engine was not designed for the high compression ratio applied and the results are not representative for the efficiencies of the combustion process. From the values of the maximum pressures in Figure 6 it is seen that the blow-by from neat ethanol at SOI of 10 CAD BTDC is expected to be much lower due to the lower cylinder pressure. The combustion efficiencies can be evaluated better from the carbon dioxide concentration which reflects the conversion of fuel carbon to exhaust carbon dioxide. These values are shown in Figure 7. Figure 7 indicates, as expected from the NRHR values, that fuel C gives the most complete combustion since the CO_2 concentrations are highest with this fuel for all SOI. The order of CO_2 concentration is: fuel C > fuel A > fuel B > neat ethanol for SOI values where the non pre-mixed combustion is important. As the injection is advanced more than 25 CAD BTDC the pre-mixed combustion dominates and the effect of additives disappears as described in the HCCI section. In Figure 8 the concentrations of unburned ethanol in the exhaust are shown. The values support the order of combustion efficiencies

for the different fuels. The ethanol values support that at late SOI neat ethanol fails to burn completely. This also seems to be the case with fuel B at SOI close to TDC.
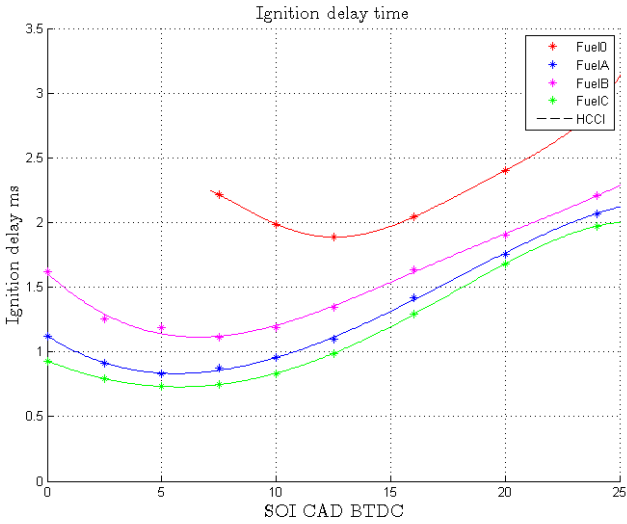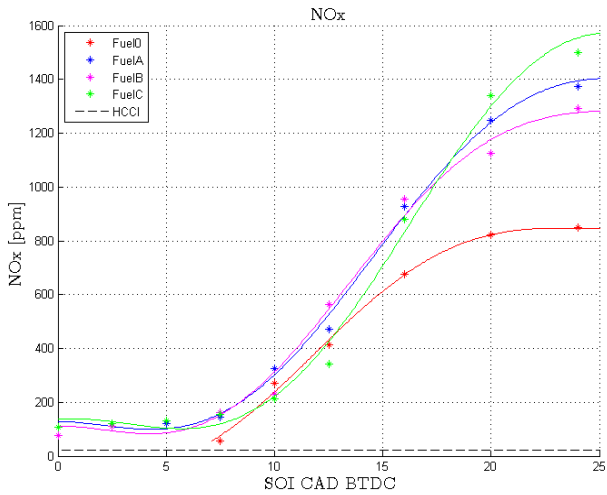

Figure 4. Ignition delay at different SOI.

Figure 5. NOx concentrations in exhaust at different SOI.
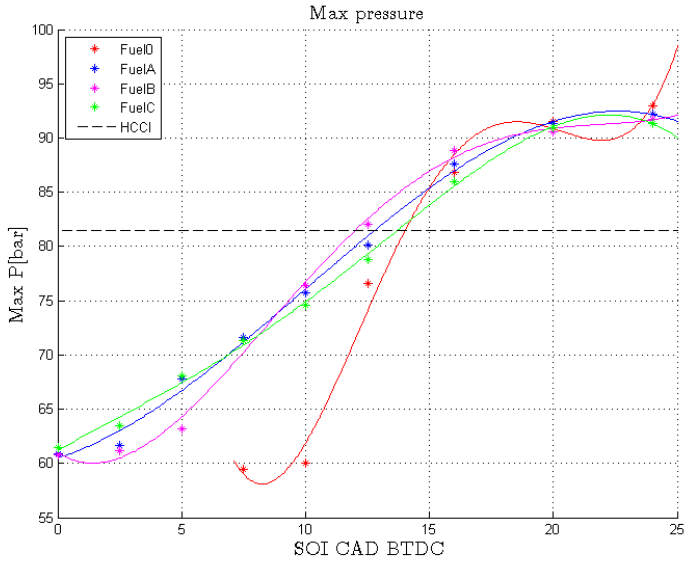
Figure 6. Maximum pressures for different SOI.
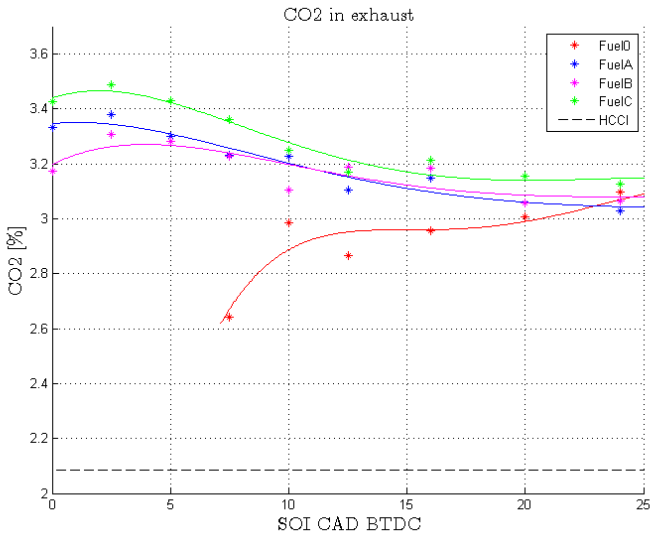
Figure 7. Carbon dioxide concentrations in exhaust at different SOI.
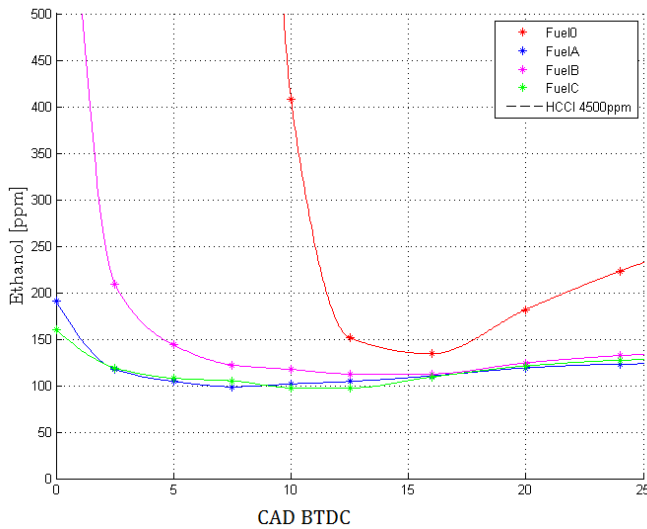
Figure 8. Unburned ethanol concentrations in exhaust at different SOI.

HCCI operation

In order to ensure HCCI operation the fuel was injected 360 CAD BTDC (SOI). With this concept the fuel will have adequate time to mix completely with the intake air. The values of NRHR and η_{net} are shown in Figure 9 and 10. It is seen that there is practically no influence of the additives on the start of ignition, and the maximum pressures for all fuels very within a few percent (not shown here). Actually neat ethanol gives the highest net efficiency.

It is, however, also seen that the HCCI principle in all cases gives a poor combustion efficiency, which can be seen from the CO_2 emissions in Figure 7 and the emissions of unburned ethanol in Figure 8. An indication of this is also seen from the lower net efficiency values in Figure 10. Finally it was clear that HCCI operation gave much higher emissions of unburned ethanol as seen in Figure 8.

Discussion

The high compression ratio of 24.5:1 was not applicable in the experimental engine used in this investigation without excess blow-by. However, since the purpose with the investigation was to compare three different ethanol fuel additives for HCCI and PPCI diesel engine operation, it was found to be of less relevance. The important part was to investigate the ignitability and the heat release/combustion pattern for the fuels in question. The engine design must be optimized for ethanol operation in "real life", like in the case with the Scania engine that has been tested in parallel to this investigation [1]. The Scania engine tests showed that all three fuel additives can be applied in the production engine with only minor performance differences [1]. This investigation digs deeper into the combustion characteristics of the three fuels. The present experiments showed that there are differences in the ignitability of the fuels during PCCI operation. The ignition delay of the fuels can be distinguished clearly in the following order: neat ethanol > fuel B (additive 2) > fuel A (additive 1) > fuel C (additive 3). This trend was actually also detected in the Scania engine. The crank position for 10 % accumulated heat release at low load (10%) is shown in Figure 11, where fuel 1, 2 and 3 corresponds to additive 1, 2 and 3 in the present investigation. The order of ignition is identical.

The additives seemed to have no influence on the ignitability during HCCI combustion. In all fuel cases this engine was not able to burn the fuel properly during HCCI operation.

Conclusions

Ethanol has been tested in a research diesel engine at low load, applying three different additives/ignition improvers, of which additive 1 is proven to work well in practice. As a reference neat ethanol was tested as well. The

intake air was preheated to 80°C and the compression ratio was increased to 24.5:1 in order to ignite all fuels properly within normal fuel injection ranges. In a parallel investigation the same fuels were tested in a Scania production engine with a compression ratio of 28:1.

The experiments with the Scania engine showed that there were only minor differences in performance of the three additives with ethanol. The results with the research diesel engine applied in this investigation also showed that the engine worked well in most PPCI modes and did show clear differences in the behavior of the fuels/additives. One of the two alternative additives burned more readily and efficiently, compared to additive 1, whereas the other alternative burned less readily and efficiently.
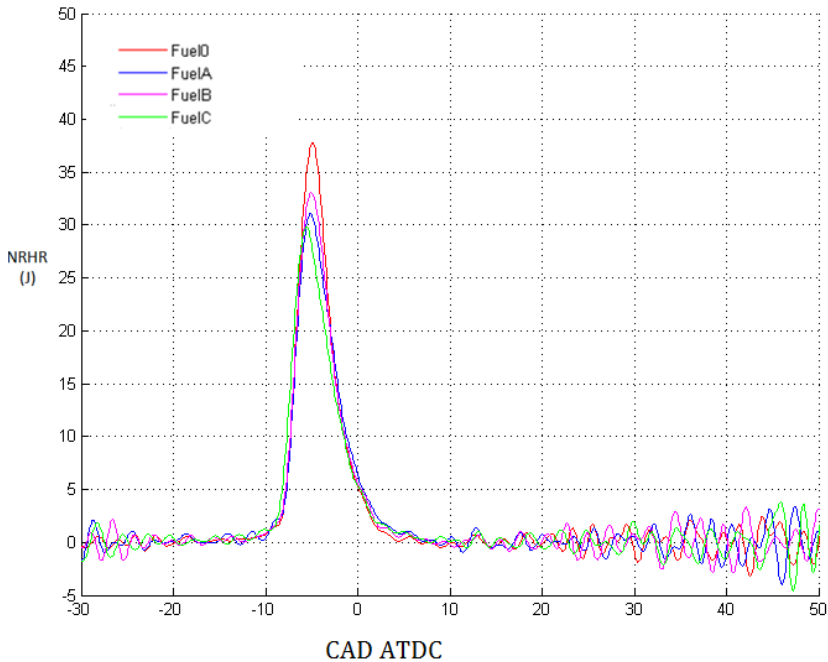

Figure 9. Net rate of heat release for HCCI operation.
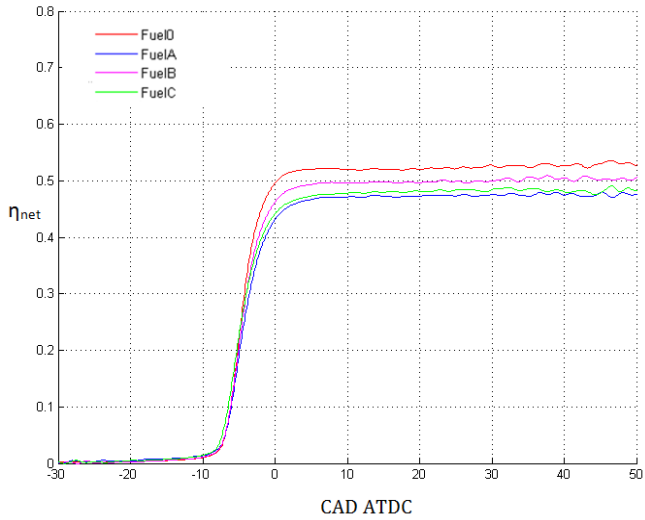
Figure 10. Net efficiency for HCCI operation.
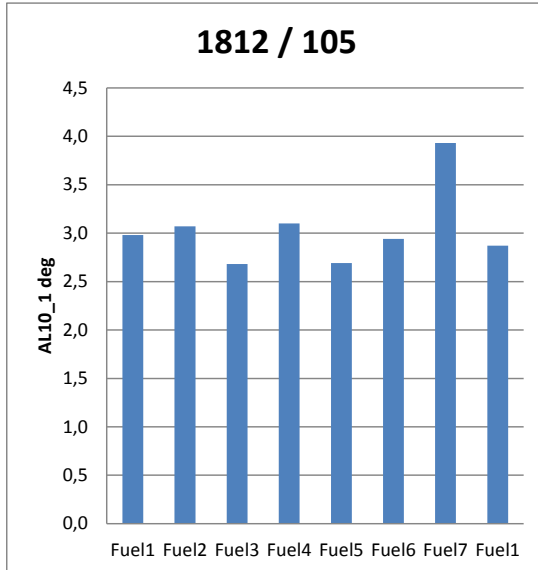
Figure 11. Result from a production engine with a compression ratio of 28:1, showing the crank position ATDC for 10 % accumulated heat release at low load [1].

Two different combustion principles were applied, PPCI and HCCI respectively. With HCCI operation the additives had no influence on the engine operation. The engine operated badly with this principle, and the combustion efficiencies were very low compared to PPCI operation. The emissions of unburned ethanol were extremely high with all fuels operating in HCCI mode.

The engine failed to operate on neat ethanol in PPCI mode when SOI was later than 7.5 CAD BTDC. The measurements of unburned ethanol in the exhaust indicated that the additive that burned poorer than the additive 1 also tended to fail at very late SOI.

References

[1]   Nylund, NO et. Al. "Testing of various fuel and additive options in a compression-ignite heavy-duty alcohol engine". The 21st International Symposium on Alcohol Fuels, Gwangju, South Korea, 2015.

[2]   Larsen, U et. al.   "Ethanol as a fuel for road transportation" Report of Annex 35, subtask 1, IEA Advanced Motor Fuels IA, May 2009.

[3]   ARIC Atmosphere, Climate & Environment Information Programme Web   site. http://www.ace.mmu.ac.uk/eae/Sustainability/Older/Brundtland_Report.ht ml.

[4]   Brusstar, M.J., U.S. EPA (Environmental Protection Agency), et al., "High Efficiency and Low Emissions From a Port-Injected Engine With Neat Alcohol Fuels," SAE 2002-01-2743, 2002.

[5]   He, B., Wang, J., Yan, X., Tsing Hua University, "Homogeneous Charge

Combustion and Emissions of Ethanol Ignited By Pilot Diesel on Diesel Engines," SAE 2004-01-0094, 2004.

[6]   Ubong, E.U., Helsinki University of Technology, "Development of an Ethanol Di Spark Assisted Diesel Engine (Sade)," SAE 901567, 1990.

[7]   Ajav, E.A., Singh, B., and Bhattacharya, T.K., "Thermal balance of a single cylinder diesel engine operating on alternative fuels". Energy Conversion & Management 41 pp 1533-1541, 2000.

[8]   Xing-cai, L., Jian-guang, Y., Wu-gao, Z., and Zhen, H., "Effect of cetane number improver on heat release rate and emissions of high speed diesel engine fueled with ethanol–diesel blend fuel,". Fuel; 83, pp 2013-2020, 2004.

[9]   Rakopoulos , C.D., Antonopoulos, K.A., and Rakopoulos, D.C., "Experimental heat release analysis and emissions of a HSDI diesel engine fueled with ethanol–diesel fuel blends," Energy, vol. 32, issue 10, pages 1791-1808, 2007.

[10]   Ashok, M.P., and Saravanan, C.G., "The performance and emission characteristics of emulsified fuel in a direct injection diesel engine,". Proc. IMechE Vol. 221 Part D: J. Automobile Engineering, 2006.

[11]   Corkwell, K.C., Jackson, M.M., and Daly, D.T., The Lubrizol Corp., "Review of Exhaust Emissions of Compression Ignition Engines Operating on E Diesel Fuel Blends," SAE 2003-01-3283, 2003.

[12]   Scania, Web site. www.scania.com.

Abbreviations

ANRHR   Accumulated Net Rate of Heat Release

ATDC   After Top Dead Center

BTDC   Before Top Dead Center

CAD   Crank Angle Degrees

CI   Compression Ignition

CO   Carbon Monoxide

DTU   The Technical University of Denmark

FTIR   Fourier Transform Infrared Spectroscopy

HC   Unburned Hydro Carbons

HCCI   Homogeneous Charge Compression Ignition

NOx   Nitrogen Oxides

NRHR   Net Rate of Heat Release

$\dot{Q}_{net}$ Net heat release from the fuel in the engine cylinder

$\dot{Q}_{app}$ Apparent heat release from the fuel in the engine cylinder

$\dot{Q}_{loss}$ Losses of heat from the engine cylinder

Q_{cycle} Total heat contained in the fuel lead to the cylinder in one engine cycle

P   The pressure in the cylinder at a given CAD

PM   Particulate Matter

PPCI   Partly Premixed Compression Ignition

SOI   Start Of fuel Injection

V   The volume of the cylinder at a given CAD

VTT	VTT Technical Research Centre of Finland
η_{net}	Net efficiency
ϒ	The specific heat ratio
Ɵ	Crank angle degree

Acknowledgment

Part C:

Testing of diesel fuel blends with methanol ethanol and butanol.

Report from the Technological Institute, Denmark (TI).

Authors: Troels Dyhr Pedersen

Summary

This report describes a comparative test of fuel blends with diesel and 10-30 % of methanol, ethanol and butanol. The engine has been run at 1500 rpm at 0, 25 and 50 % of its rated load.

In all tests, the injection timing was adjusted to maintain the same combustion timing for the fuel blends as the diesel reference, in order to minimize the effect of ignition delay on the formation of particulates. Since alcohols increase the ignition delay, neglecting this effect does not only result in increased smoke due to late combustion, but also substantially increased fuel consumption as well as difficulties with starting and running the engine unloaded.

Measurements on the particulate emissions have shown that the particulate number is reduced with increasing quantities of alcohol. The simplest alcohols ethanol and methanol gave the largest reductions, which where proportional to the fraction of alcohol in the blend. The best result was obtained with 20 % methanol and   10 % butanol as cosolvent. This blend reduced the particulate number up to 75 % in loaded operation, compared

to the diesel reference. With 30 % ethanol, the reduction was up to 60 % in loaded operation. With 30 % butanol, a reduction of up to 40 % was found.

The gaseous emissions were found to be increasing with alcohol blends, but only in idle condition. The alcohol containing blends were found to increase emission of specific aldehydes, carbon monoxide and hydrocarbons. The higher cylinder and exhaust temperatures in loaded operation mean that the combustion is less sensitive to the fuel composition than at idle, and therefore only minor differences was found in loaded operation. The emission of nitrogen oxides was also affected, but no consistent increasing or decreasing trends were found.

Accurate measurements of the fuel consumption show that the brake thermal efficiency improves with increasing quantities of ethanol and methanol. Butanol does not appear to improve efficiency as significantly.

When preparing the diesel fuel blends, large differences in the miscibility of the three alcohols with diesel was clearly observable. Butanol could be blended into diesel without any visible separation or even discoloration. Ethanol did not form a homogeneous blend with cold diesel initially. After some circulation and heating of the blend, it eventually became clear even with 30 % ethanol. Methanol did not form a stable blend with diesel even when heated, but with the aid of butanol as cosolvent, a semi-stable blend was obtained, which in combination with a high circulation rate prevented the components from separating.

Background

The motivation for blending alcohol with diesel fuel is primarily to replace diesel as fuel, since this can help to reduce the increasing demand for diesel. An extra benefit is that the addition of oxygenated fuel can lead to a

substantial reduction in the formation of particulate matter. The effect on the other undesired emissions in the exhaust such as nitric oxides, carbon monoxide and hydrocarbons is less well established, and therefore likely more dependent on the specific engine technology and operational conditions.

The use of diesel fuel blends with oxygenated components such as alcohols may be relevant in particular for older engines that do not comply with the newer standards for emissions. These engines are typically not equipped with advanced exhaust after treatment systems, which can reduce emissions of particulate matter and nitric oxides. Engines that are not built or adapted to comply with the strict road vehicle legislation are typically found in e.g. non-road machinery, railway locomotives, all sizes of marine engines and diesel generators. Emissions from these engines are significant and sometimes the major sources of particulate matter and nitric oxides.

In many less developed countries, the legislation for emission control is either non-existing or not effective, due to the majority of vehicles and engines being old and poorly maintained. In these countries, the effect of blends with diesel and alcohols fuel could have a very large impact on the particulate emissions, as well as reducing the import of oil-derived diesel fuel while supporting local production of the alcohol required for the blends.

Methanol and ethanol are both inexpensive fuels that is widely used for blending with and even substituting gasoline. Blending these simple alcohols with diesel fuel is however more complicated, since such blends can separate in certain conditions such as low temperatures and presence of small quantities of water in the fuel. One approach to solving this problem is to produce emulsions of diesel and the alcohol, with the aid of surfactants.

Such emulsions can be more stable at low temperatures and tolerant to water contamination, but can degrade at higher temperatures and may have a limited lifetime during storage. Emulsions are also complicated to produce, since the alcohol droplets are only few micrometers in diameter. Considering the effort required producing emulsions, blends may be a more attractive option as they can be prepared instantly when refueling a vehicle, and individual blend compositions may be available to the consumers depending on the climate, the engine configuration and other factors that may make a particular concentration of alcohol the best solution.

Literature review on alcohol/diesel blends and emulsions

The concept of using blends and emulsions of diesel and simple alcohols became a popular subject of investigation around 1980. Several studies were performed on blends with alcohols. Emulsions with water were also investigated, which led to the idea of emulsifying alcohols in order to overcome the problems with blend separation.

The use of blends or emulsions of diesel and alcohols are a way of substituting diesel fuel. Diesel fuel is relative expensive to make by biological or synthetic processes, so all fuels that can be used to stretch the available resources will help to satisfy demand are interesting from an economic perspective.

Even though the alcohols methanol and ethanol have very poor diesel combustion characteristics, they also offer some unique advantages. The oxygen that is carried with the alcohols increases combustion efficiency, such that less particulate matter is created.

Blends

A blend of alcohol and diesel is a homogeneous mixture, which means that the components are mixed on a molecular level. A surfactant or cosolvent is normally used to increase the miscibility of the two liquids, since alcohols are polar while diesel is non-polar.

Blending of methanol with diesel is very problematic due to the high polarity and compactness of the methanol molecule, as well as its highly hygroscopic nature. Methanol can be blended with diesel in low concentrations under optimal conditions, but the blend will separate if water enters the blend, as the methanol binds the water and becomes completely immiscible. The solubility also decreases with temperature. These challenges are part of the reason that methanol is not suited for blending with normal diesel.

Ethanol is more miscible with diesel than methanol, although it will still separate at low temperatures and if water is present. Therefore most studies and experiments since 1980 has focused on ethanol as blending component, rather than methanol.

Butanol, or more precisely n-butanol, is a promising alcohol that is fully miscible with diesel, since it has a low polarity. It also has a high cetane number of 35, which reduces the requirement for restoring the cetane number of the fuel blend with additives. Butanol has so far been complicated and expensive to produce by fermentation. It has therefore been produced synthetically, but still the cost has been more than twice that of ethanol. The production costs may however be reduced with recent advances in fermentation and distillation technologies. It is quite likely that n-butanol will be playing a prominent role as a blending component for diesel in the near future or even as a pure diesel fuel in moderately modified

diesel engines.

Both methanol and ethanol are fully miscible with biodiesels, since these have ester groups that are polar and hence can bind the methanol through hydrogen bonding. Fuel combinations with both fossil and bio-based diesels and alcohols are therefore a cost-effective way of stabilizing the fuel mixtures, because the use of biodiesel as cosolvent for the alcohols is cheaper than most alternatives.

Emulsions

An emulsion of alcohol in diesel consists of dispersed droplets of alcohol in a continuous phase of diesel. This kind of emulsion is called water-in-oil, or W/O, since the alcohol is the dispersed phase.

Micro- and nanoemulsions

Polar and non-polar liquids can form two kinds of emulsions: micro- or nanoemulsions. As a response to widespread confusion between these two, McClements has provided a very detailed and comprehensible explanation of microemulsions and nanoemulsions. Although his article is focused on oil-in-water (O/W) emulsions, the underlying considerations and theories are of high relevance to alcohol/diesel emulsions as well.

Although the names suggest that emulsions can be distinguished by the droplet size being in a nano or micrometer scale, this is not the case – both emulsion types typically have droplet sizes below 100 nanometers and nanoemulsions typically have larger droplet sizes. The confusion of names is mainly because they been used more or less randomly in the last few decades, for emulsions with droplets smaller than 100 nanometers. Such emulsions are translucent, since light is not scattered by droplets smaller

than approx. one quarter of its wavelength. The two emulsion types can even be prepared from the same components. The type of emulsion resulting from the process is mainly determined by the surfactant type and concentration, as well as the method used to prepare them.

The most important difference between nano- and microemulsions is the stability of the emulsion, which is explained by Gibbs free energy. Nanoemulsions has a higher free energy in the emulsified state than in the separated state, whereas microemulsions has a lower free energy in its emulsified state, than in its separated state. Nanoemulsions are therefore thermodynamically unstable, meaning that the components will tend to revert to their separate states. Microemulsions are, at least in principle, infinitely stable, and will therefore tend to stay in the emulsified state. Stability is increased by keeping the droplet size below approx. 90 nanometers, where Brownian motion dominates the gravitational forces.

Microemulsions can form spontaneously when mixing the components, but often an energy input is required to overcome a small energy barrier. This energy can be applied by agitation or heating. Nanoemulsions typically require much more energy input, both by heating and mechanical mixing, since the level of free energy is higher in the emulsified state.

Advantages of emulsions

Emulsification is relevant, since in most cases, alcohol and diesel are not completely miscible. While pure ethanol and methanol can be blended with diesel, the blends become unstable at low temperatures and/or low humidity levels. It can therefore make sense to emulsify these alcohols, to obtain more stable fuel mixtures than blending can offer. Though emulsions with butanol and higher alcohols is possible, it is less relevant since higher alcohols are perfectly miscible with diesel, and do not separate in changing

conditions.

Stabilization of emulsions

The stabilization of the emulsion is performed with the emulsifier, which is a surfactant that surrounds the alcohol droplets. This prevents the alcohol droplets from collating into larger drops, which eventually causes phase separation. Non-stabilized emulsions of diesel and alcohols can exhibit some short-term stability, which can be adequate if the emulsion is prepared immediately before it is injected into the combustion chamber. This has been utilized in a demonstration of an on-board emulsification process for a vehicle as well as by various researchers [13].

Engine performance changes with alcohol/diesel fuel blends

Emissions

The most important observation when using alcohols in combination with diesel is that the emission of particulate matter (PM) is substantially reduced. This effect is primarily caused by the presence of oxygen in the fuel. A large review study [7] of exhaust emissions of diesel engines in transient operation found that the measured PM is a linear decreasing function of the oxygen content. The effect of 10 % ethanol was found to be comparable to that of 16 % butanol. At this blend ratio, either alcohol decreases the emission of PM with about 25 %. The study also concluded that alcohols are more efficient in reducing PM than biodiesel, which is also an oxygenated fuel.

The study did not find any significant relations with other emissions, other than hydrocarbons, which were found to be increasing with the addition of alcohols. Increased emissions of hydrocarbons is however a minor concern, since these are easily removed by simple oxidation catalysts.

Other studies [11] have found that fuel blends have resulted in either increased or decreased levels of nitrogen oxides (NOx). Nitrogen oxides are generated in the mixing controlled combustion zones, where the temperature is highest. The rate of formation and the final concentration depends mainly on the temperature and residence time. NOx formation is therefore very sensitive to combustion timing, with late timings resulting in lower concentrations. This sensitivity therefore likely the main reason to the reported differences in emissions of NOx when diesel/alcohol fuel blends

are tested, and not an effect of the chemical composition or specific reactions with the fuel.

Efficiency

The brake thermal efficiency of the diesel engine is dependent on the combustion phasing and duration. Since the cetane number decreases when diesel is mixed with alcohols, the combustion phasing is also retarded. If the fuel is to be used in unmodified engines, the cetane number must be restored with cetane improving additives. As an alternative under experimental conditions, the injection timing can be advanced to compensate for the ignition delay, such that the correct combustion phasing is maintained.

The heating values of alcohols are also lower than for diesel fuel. As an example, ethanol has a heating value of approx. 21 MJ/L, whereas diesel has a heating value of approx. 36 MJ/kg. For a given requirement in terms of fuel energy, a larger fuel volume is therefore required when a blended fuel is used instead of pure diesel. This will result in an increased brake specific fuel consumption both in terms of mass and volume, but not necessarily in terms of energy. Several studies [7,11,18] indicate that fuel blends with alcohols can provide a minor improvement in brake specific energy consumption, which is a direct result of an overall more efficient combustion process. The improvement may be caused by a shorter combustion duration (which is thermodynamically more effective), as well as a slightly higher combustion efficiency.

Additives

Surfactants

The surfactants used for blends and emulsions are molecules with a lipophilic (non-polar) carbon chain, which are attached to a hydrophilic (polar) group. This composition means that the molecule can create bonds with both the non-polar diesel fuel molecules and the polar alcohol molecules.

The surfactants have been found to behave differently according to the balance between their hydrophilic and lipophilic parts [20]. This relation is called the hydrophilic/lipophilic balance, HLB:

$$HLB = 20 \, \frac{M_h}{M}$$

With M_h being the molar mass of the hydrophilic part and M being the molar mass of the molecule.

In blends, the role of the surfactant is to ensure that the alcohol molecules are sufficiently miscible with the diesel fuel. The proper surfactants for this purpose are called solubilisers. These molecules have a HLB of 15 to 18, meaning that the hydrophilic part is dominating. They are soluble in water and insoluble in oil.

In emulsions, the surfactant is used to form a membrane around alcohol droplets, called a micelle. The hydrophilic end, which is bonded with the alcohol, is arranged in a sphere around the alcohol droplet, while the lipophilic end stretches into the continuous diesel fuel phase. The surfactants used for emulsions are called emulsifiers. Emulsifiers have a HLB

of 3 to 6, meaning that the lipophilic part is much larger than the hydrophilic. They are therefore soluble in oil and insoluble in water.

Surfactant types and elemental composition

The variants of surfactants that are used for blends and emulsions can be either of the anionic or non-ionic type, which refers to net charge of the hydrophilic end. Since the surfactant is to be used in concentrations above 1 %, they must not contain elements, which can form toxic combustion products, or products that can damage internal engine components or the exhaust after treatment system.

Most non-ionic surfactants contain only oxygen or hydroxyl (OH) as functional groups. In some compounds, the functional groups are nitrogen or amines (NH). The fuel born oxygen and nitrogen do not form any combustion products that can harm the engine or be a concern for humans and environment. These types of surfactants are however typically the weakest interacting, and must be used in larger quantities than e.g. anionic surfactants for the same effect.

Anionic surfactants typically contain elements from the periodic system group II such as sodium, potassium or phosphor. These elements can create salt deposits, which are harmful to both engine and after treatment systems. It is therefore unlikely that these can be used. Another common element in anionic surfactants is sulphur, which is not harmful to the engine, but the combustion product (sulphur dioxide) decreases activity of diesel oxidation catalysts, and is also a precursor for soot. Sulphur containing surfactants should therefore only be considered for diesel engines without oxidation catalysts or catalysts integrated in the diesel particulate filter. It should also be remembered that the limit for sulphur in diesel fuel for road use is currently 10 ppm, so the concentration of surfactant must be very low if it

contains sulphur.

Cationic surfactants typically contain fluorine, chlorine and bromine, which are group VII elements. These elements are toxic in low concentrations and form toxic gases in combustion, and the use of cationic surfactants should therefore not be considered.

Cetane improvers

The cetane number of the fuel blend or emulsion de with the addition of alcohol. In order to meet the EN590 standard for diesel, which requires a cetane number of at least 51, an additive that restores the cetane number is required. Since diesel fuel is a complex product of varying quality and cetane number, cetane improvers are commercially available from chemical suppliers such as Akzo Nobel and Lubrizol.

Corrosion inhibitors

Alcohols are weak acids, which can cause corrosion of some metals and alloys. Aluminum and zinc are susceptible to corrosion by ethanol and methanol. Corrosion inhibitors that has been developed for gasoline/ethanol blends are commercially available and usable for diesel/alcohol blends.

Outline of the experiment

The purpose of the test series presented in this report is to make a comparison of the emissions and combustion characteristics of the three alcohols methanol, ethanol and n-butanol, when used as minor fraction blending components with diesel. Standard diesel is also measured and used as reference.

The experiment was intended to test blends, which were prepared in

batches with 10, 20 and 30 % alcohol by weight. This was performed with ethanol and butanol, while methanol was tested in combination with butanol at 10 % and 20 % concentrations, due to problems with obtaining a stable blend. A 30 % methanol blend was not tested, since it was clear that the fuel ignitability would become a problem during starting and idling.

The testing of the fuel blends was performed with a heavy-duty truck engine coupled to an engine dynamometer. The bench setup was operated at 1500 rpm at 0 %, 25 % and 50 % loads. Emissions and fuel consumption was measured when the engine had reached steady state.

The concentrations of gaseous and particulate emissions were measured with FTIR spectroscopy and a particle counter, respectively. The fuel consumption was measured with a precision scale.

Experimental setup

Fuel blends

	Fuel composition (in weight %)								
Diesel	90	80	70	90	80	70	90	85	70
Ethanol	10	20	30						
Butanol				10	20	30		5	10
Methanol							10	10	20
Name of blend	E10	E20	E30	B10	B20	B30	M10	M10 + B5	M20 + B10

Table 1: Composition of fuels in the test

Engine

The engine is a Scania model DC09 071A. It is a four stroke DI diesel with specifications as in table 1. The engine is equipped with turbocharger and intercooler.

Number of cylinder	5 in-line
Displacement	9.3 liters
Bore x stroke	130 x 140 mm
Compression ratio	16:1
Rated power and torque	226 kW and 1440 Nm @ 1500 RPM
Emission compliance	EU stage III a

Table 2: Engine specifications

The engine model is adapted for power generation at fixed speeds of 1500 or 1800 RPM. Due to the programming of the ECU being optimized for 1500 RPM, this speed was used for the test series.

The engine is connected to an AC dynamometer with a frequency controller, which controls the engine speed or torque as desired. It also measures the torque and power produced by the engine. Control of shaft torque was handled by the dynamometer, while the engine was operated in speed control mode. Since the dynamometer is limited to absorbing a torque of 700 Nm, the engine could only be tested at approximately half of the rated power.

ECU parameter control

The engine has previously been modified to run on experimental fuels. The original engine control unit has been replaced with a programmable MEDC 17 FLEX ECU unit from BOSCH, which allows full programmability and

control of the parameters for injection. The ECU is monitored by a software named INCA on a remote laptop, from which injection parameters can be adjusted online.

The control for the EGR valve has not been implemented in the programmable ECU, so although present, the EGR is not used. The lack of EGR means that the intended suppression of NOx formation at idle and low load operation is not possible, and thus there is some potential for improving the emission performance obtained with the alcohol blends. It may also be possible to increase the inlet air temperature, such that the ignition delay is reduced. EGR is however only fitted on this particular engine model, whereas all models with higher power output, including the 6 and 8 cylinder versions, does not have EGR.

The control for the pneumatic waste gate control on the turbocharger waste gate has not been implemented in the ECU either. It is however not needed, since at medium load, the turbocharger does not reach the pressure at which the waste gate should be activated.

Injector technology

The engine is equipped with unit injectors (PDE). These injectors combine a pressure pump with a pressure activated injector nozzle in one unit. The injector pumps are driven by the camshaft in parallel with the valve train. Control of the nozzle opening is performed by closing a fuel bypass with a solenoid valve, thereby forcing the fuel through the injector nozzle.

The PDE injector technology, although a very reliable and cost-effective solution, may be considered outdated in a market where most road engines today are equipped with high-pressure common rail systems. It is however also engines with older injection technology and with little or no exhaust after treatment, that can benefit the most from new fuel formulations

which can reduce emissions of particulate matter. In addition, older injector technologies is still being used in many applications where the precise control provided by the common rail is not required.

Injection timing

The timing of the injection is fully adjustable and can be changed while the engine is running. This was used to compensate for the longer ignition delay of the fuel blends with alcohol, such that the combustion took place at the same CAD for all fuels. After testing with diesel, it was decided to use the positions of peak heat release for premixed combustion in table 3.

	Idle	25 % load	50 % load
Start of injection CAD BTDC	11	12	12
Peak of premixed heat release CAD ATDC	10	5	3

Table 3: Injection timing and position of premixed heat release peak with standard diesel fuel

The injection angles that were initially chosen may be considered late for diesel, but the concern was that injection angles would become too advanced with the low cetane blends, particularly those containing methanol. Table 4 displays the values for Start of Injection (SOI) that was used to compensate for the ignition delay.

SOI CAD BTDC	Diesel	B10	B20	B30	E10	E20	E30	M10	M10 + B5	M20 + B10
Idle	11	12	12	13,5	11	15	18	11	13	20
25 %	12	12	13	15	12	14	17,5	12	14	17

| 50 % | 12 | 12 | 12 | 13 | 12 | 13 | 14,5 | 12 | 13 | 14 |

Table 4: Start of injection (SOI) with the various alcohol diesel blends.

By maintaining a fixed position of the premixed combustion, some effects on the emissions related to retardation of the combustion was avoided, such as increased particulate formation and reduced nitrogen oxide formation. A constant combustion timing also makes a comparison of the brake thermal efficiency more reasonable.

Fuel system

The original fuel system on the engine was used in combination with a 20-liter tank, which was placed on a precision scale.

The fuel system feeding the unit injectors is comprised of a low-pressure pump driven by the crankshaft, a fuel filter and a fuel rail that distributes fuel to the cylinder heads. A backpressure valve opening at 4.5 – 5 Bars directs fuel into the return line and back into the fuel tank. This system turned out to function well, since the fuel was subjected to both powerful circulation and heating, which helps to keep the alcohol from separating from the diesel.

Cylinder pressure measurements

Cylinder pressure is measured on one cylinder. This produces accurate information about the ignition delay as well as the shape and peak of the heat release curve. The peak of the premixed combustion is used to adjust timing, since this peak is the most consistent and visible part of the heat release curve which is displayed.

The pressure measurements contain valuable information regarding the timing and duration of the combustion. The influence of alcohols on the

ignition delay is of particular importance in this study, as well as the influence on the premixed combustion.

Test procedure

Test scheme

All tests were performed at 1500 RPM. The engine loads tested were 0, 25 and 50 % for each fuel combination. In order to ensure that all measurements were performed consistently and without using the fuel batch, a time scheme was set up for the testing of each fuel combination.

Load [%]	Brake torque [Nm]	Brake power [kW]	Fuel cons. est. [kg/h]	Time at steady state [min]	Fuel cons. exp. [kg]
Initial warmup	0	0	5	30	2.5
0	0	0	4.4	15	1.1
25	350	55	15.6	10	2.6
50	700	110	22.6	10	3.8

Table 5: Time scheme for testing

All testing of each alcohol fuel was completed in one session. The engine was first started and warmed up with standard diesel. The sequence in table 5 was then completed with standard diesel to make a reference measurement. After completing the sequence, the engine was stopped and the first fuel blend with 10 % concentration of alcohol was prepared. The sequence was then run again with the fuel blend. This was repeated with increasing concentrations of 20 % and 30 % of the same alcohol, except for

methanol where butanol was used as cosolvent.

After each run of the load sequence shown in table 5, the engine was allowed to idle for 5 minutes to cool the turbocharger, pistons and engine oil, before it was stopped. The engine was further cooled in the time it took for refueling to a new concentration, such that each measurement could start with conditioning the engine to the same temperature before testing.

The required quantities for the refilling of the fuel tank to 10 %, 20 % and 30 % concentrations were calculated in a spreadsheet, based on the remaining fuel quantity in the engines fuel system, connecting lines and the fuel tank.

Measurements of emissions and fuel consumption were performed when concentration levels of NO and NO2 had stabilized.   The temperatures of cooling water, engine oil, inlet/exhaust and the turbocharger pressure were monitored and logged manually. The log files confirmed that oil and cooling water temperatures were increasing equally with the change in load levels in all the tests, which means that the engine friction should be constant as well.

Fuel measurement setup

Fuel consumption was measured on a digital scale during intervals of 6 minutes with constant load. As the scale output is in kg with two decimals, care had to be taken in the measurement. The time was started on the change of the last digit on the scale, and stopped again on the first change of the last digit occurring after 6 minutes. The uncertainty when using this method is an estimated 5 seconds or about 1.4 % for the consumption of the given fuel quantity measured at idle. With the engine loaded at 25 % and 50 %, the fuel consumption is much higher and therefore the uncertainty is reduced to about 3 seconds, which is 0.8 %.

In order to get stable readings from the scale, it was necessary to isolate it from floor vibrations by placing it on a heavy metal plate on top of a tall foam matt. Vibrations in the fuel forward and return lines were removed by fixing the lines with rubber-insulated pipe fixtures.

Emission measurement setup

The gaseous emissions were measured with an ANTARIS IGS FTIR from Thermo Scientific. The emissions were measured non-diluted and non-condensed through a sample line at 180 °C. The species included in the calibration of the instrument are listed in table 6.

Component	Name	Cal. range
Water	H_2O	0.5 – 25 %
Carbon dioxide	CO_2	0.1 – 20 %
Carbon monoxide	CO	1 - 10,000 ppm
Nitric oxide	NO	1 - 10,000 ppm
Methane	CH_4	1 - 10,000 ppm
Ammonia	NH_3	1 - 1000 ppm
Nitrogen dioxide	NO_2	1 - 100 ppm
Nitrous oxide	N_2O	1 - 100 ppm
Sulfur dioxide	SO_2	1 - 100 ppm
Acetylene	C_2H_2	1 - 100 ppm
Ethene	C_2H_4	1 - 100 ppm
Ethane	C_2H_6	1 - 100 ppm
Propene	C_3H_6	1 - 100 ppm

Propane	C3H8	1 - 100 ppm
1,3 butadiene	C4H6	1 - 100 ppm
Iso-butylene	C4H8	1 - 100 ppm
N-Pentane	C5H12	1 - 100 ppm
Benzene	C6H6	1 - 100 ppm
Iso-octane	C8H18	1 - 100 ppm
Formaldehyde	CH2O	1 - 100 ppm
Acetaldehyde	CH3CHO	1 - 100 ppm
Total hydrocarbons	THC	FID equivalent[2]
Non-methane hydrocarbons	NMHC	FID equivalent
Total NOx	NOx	NO + NO2

Table 6: List of measured species

The emission of particulate matter was measured with an AVL 489 Advanced Particle Counter. The instrument is a CPC type counter. The functional principle is to dilute the raw gas, dry the particles in a heated chamber, dilute the hot sample (which also cools it) and then condense evaporated n-butanol onto the particles. The particles are then illuminated by laser and counted by photo detection. The instrument is capable of detecting particle sizes down to a mean diameter of 23 nm. Particles of VOC are not measured, since they will be evaporated and diluted before the detection chamber.
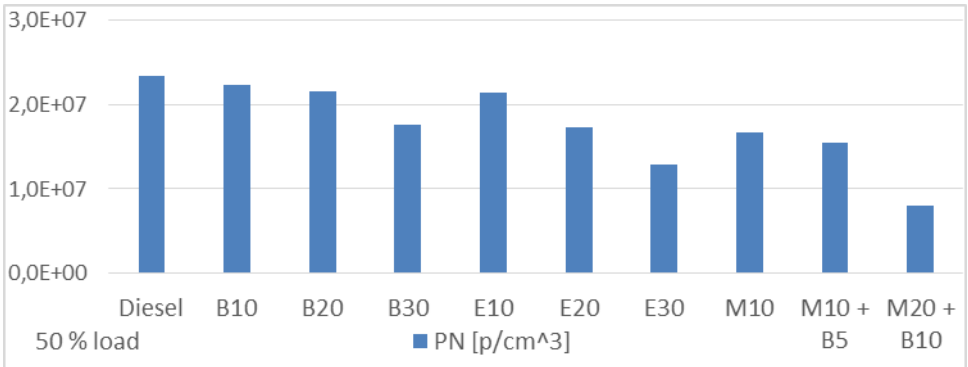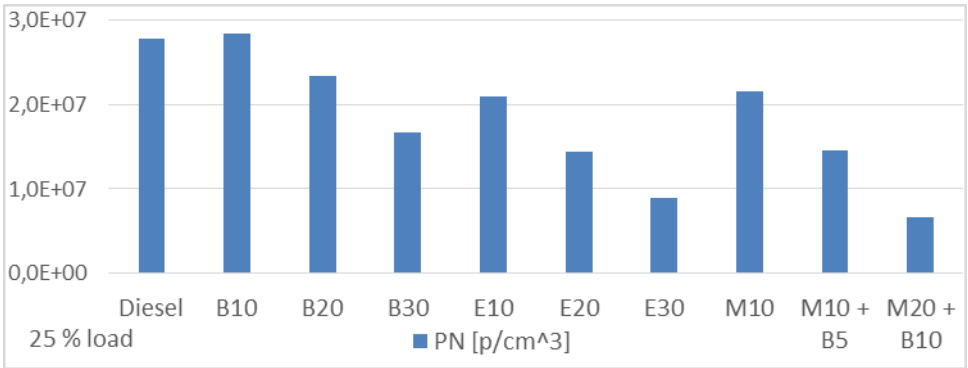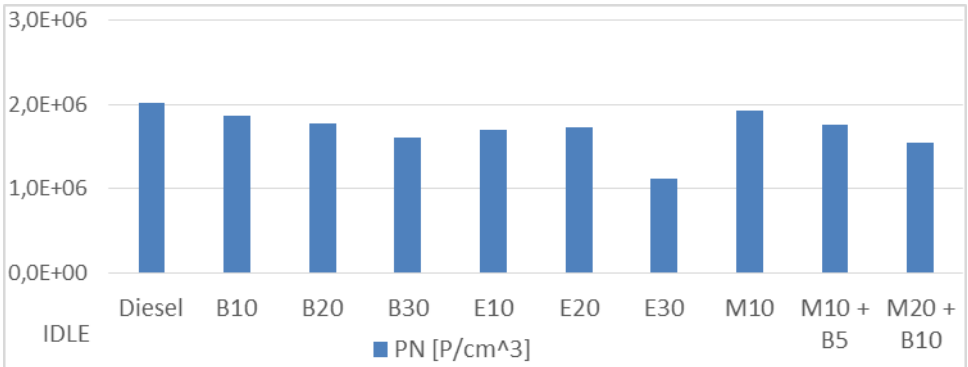
---

[2] The hydrocarbons in the table are used to calculate a value that is equivalent to the response of a standard FID instrument, which has output in propane ppm.
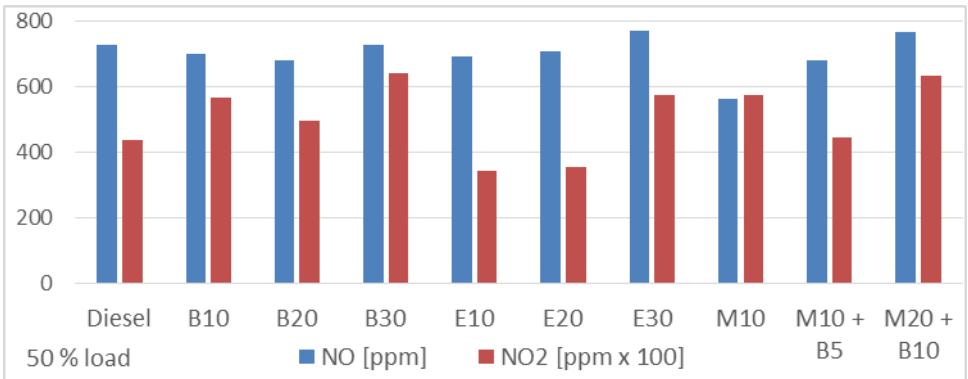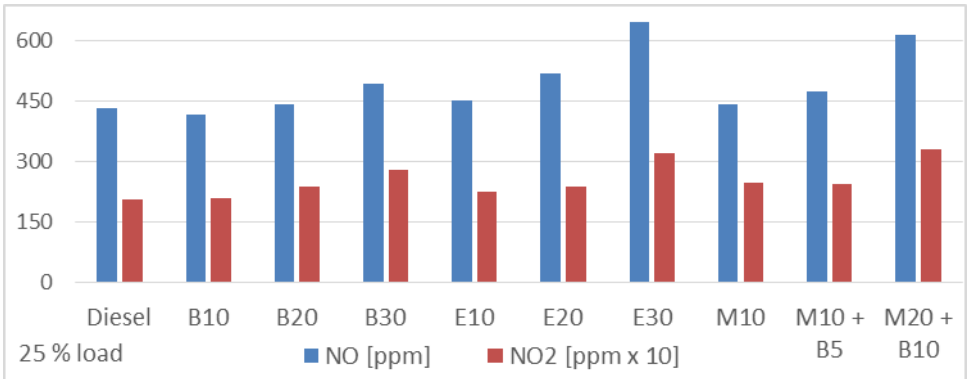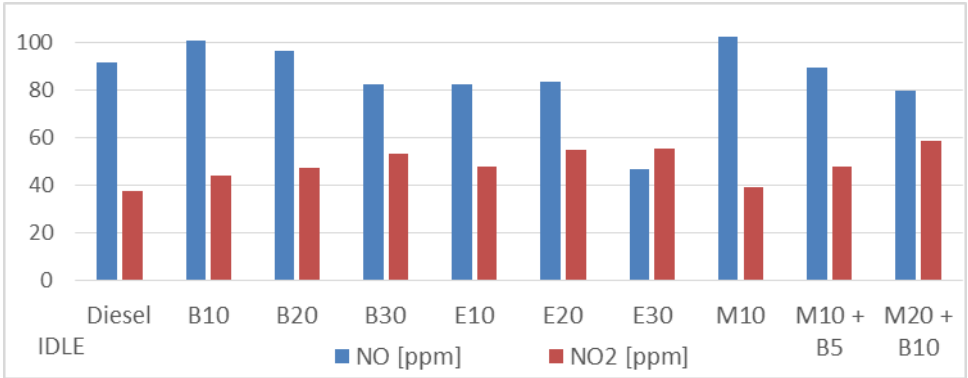
Measurements

Particulate number

The formation of particulates at idle appear to be only moderately affected by the addition of alcohols. However, the reference level is already quite low in idle operation, less than 10 % of the levels in loaded operation. With 30 % E, the PN is reduced to about half of the reference level, which is quite significant considering that the NOx formation is reduced as well (next page).

At 25 % and 50 % load, the particulate concentration increases approx. 10-15 times compared to idle. The formation of particulates is greatly reduced with addition of alcohols. The reduction is most effective with methanol and ethanol. The addition of 5 % butanol to the 10 % methanol blend appears to improve the reduction obtained with 10 % methanol only. Since butanol only has a minor effect on particulate formation at 10 % concentration, the combined effect of methanol and butanol on particulate formation could be caused by a more homogeneous fuel mixture.
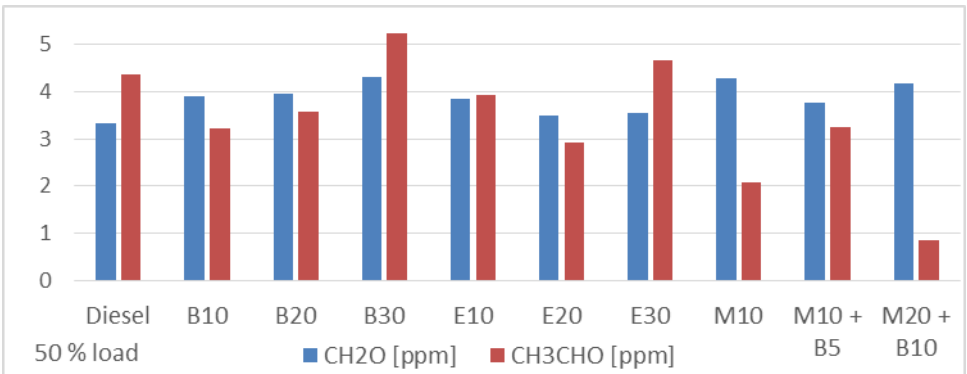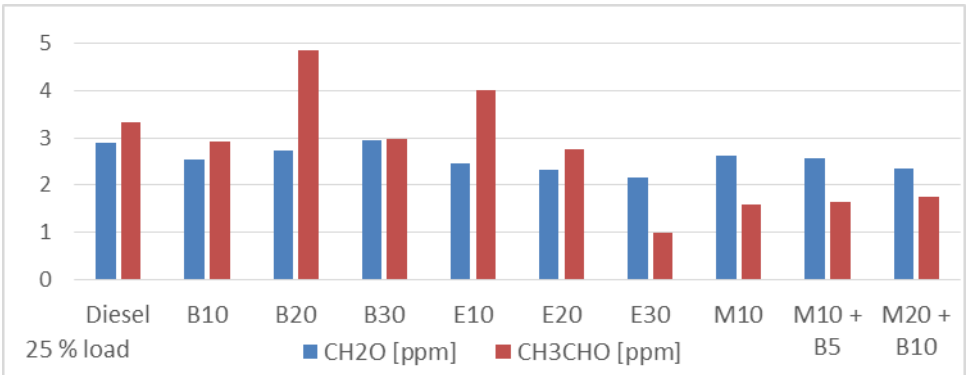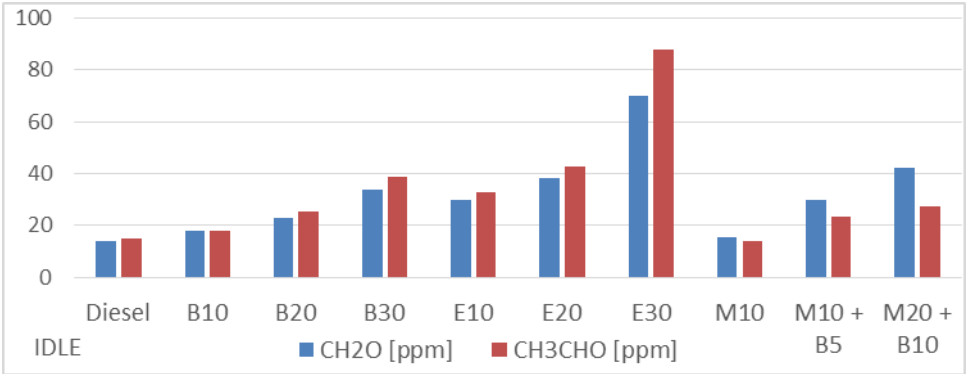
IDLE
PN [P/cm^3]

25 % load
PN [p/cm^3]

50 % load
PN [p/cm^3]

Nitrogen oxides

IDLE
NO [ppm]   NO2 [ppm]

25 % load
NO [ppm]   NO2 [ppm x 10]

50 % load
NO [ppm]   NO2 [ppm x 100]

In idle, the emissions of NO and NO2 are moderately affected by the addition of alcohols. The level of total NOx is very constant, except with 30 % E, in which test the premixed combustion was suppressed. It appears that the balance between NO and NO2 is changed towards NO2 with increasing percentage of alcohol, which indicates that the combustion products are being cooled more when alcohol is present in the fuel.

At 25 % load, NO2 levels are reduced to approx. 5 % of the total NOx emission. Nitrogen oxides are moderately increasing with increasing amounts of alcohol. The increase is stronger with methanol and ethanol. At 50 % load, the NO2 concentration is reduced to less than 1 % of the total NOx emission. There is a weak effect on NO emission with ethanol and a somewhat stronger effect with methanol, where a low concentration of methanol is lowering the NO and the higher concentration is increasing it again. A possible explanation for the effect of alcohol at 25 % load and, to some extend 50 % as well, may be that the heat release rate in the premixed and mixing controlled combustion phases are increasing with increased concentration of alcohol, due to a higher volatility of the fuel combined with a longer ignition delay. The increased heat release rate results in an elevation of the combustion temperature and hence the NO formation.

Aldehydes
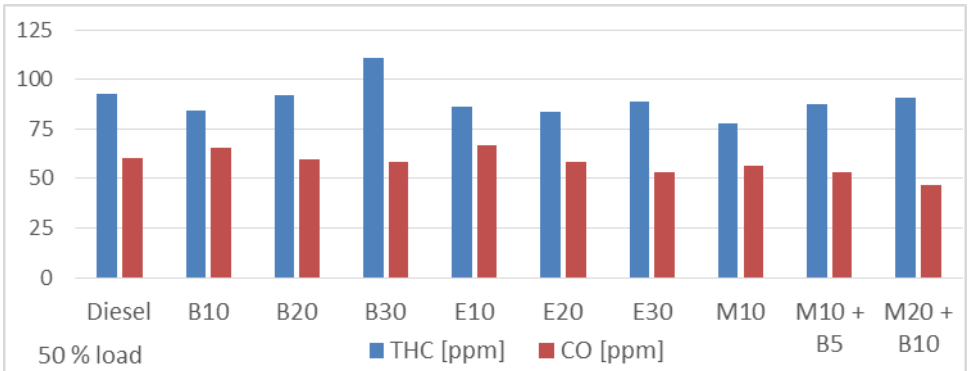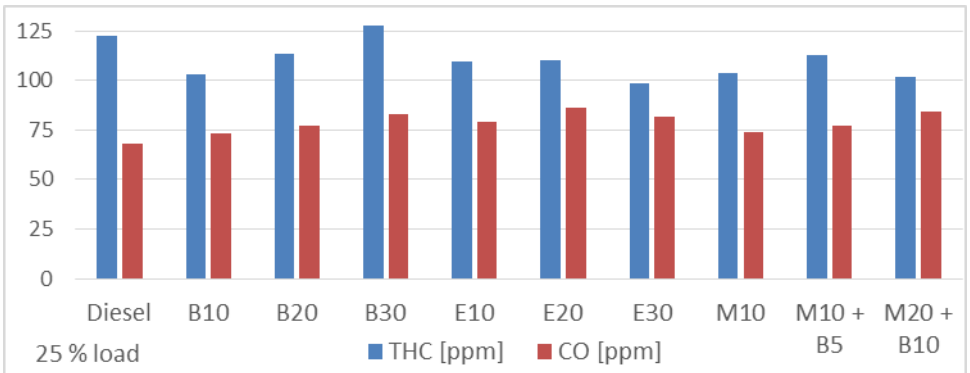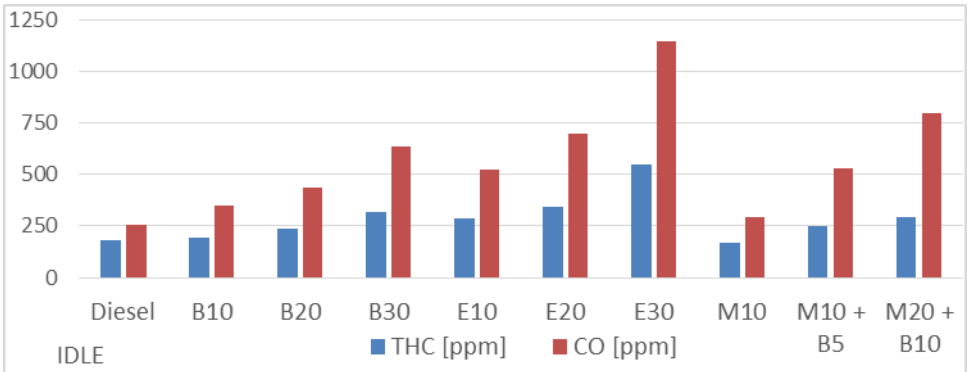

IDLE

25 % load

50 % load

In idle, the formation of formaldehyde and acetaldehyde are significantly increasing with the addition of alcohols. An increase in formaldehydes is also known from SI engines operating on ethanol and methanol. The largest formaldehyde increase is seen with the 30 % E blend, which had poor premixed combustion. Butanol and methanol result in less formaldehyde formation than ethanol.

Aldehydes are generally much lower in loaded operation, due to the high temperatures in the combustion chamber. The emission levels of aldehydes do not appear to be increasing by alcohol addition, as was observed in idle condition. The aldehyde emission at 50 % load appears higher than that observed at 25 % load. However, as the levels are still all below 5 ppm, these emissions are of minor importance.

Hydrocarbons and carbon monoxide

In idle condition, the emission of carbon monoxide and hydrocarbons are increasing significantly with the addition of alcohols. The large increase in CO indicates that the combustion is being less efficient with increasing amounts of alcohol in the blend.

The emissions of hydrocarbons and carbon monoxide are generally low in loaded operation, which is a consequence of the higher combustion temperatures. There is no obvious effect on the emissions by the addition of alcohols when the engine is running loaded.

IDLE · THC [ppm] · CO [ppm]

25 % load · THC [ppm] · CO [ppm]

50 % load · THC [ppm] · CO [ppm]

Brake thermal efficiency

Brake thermal efficiency - 25 % load

Categories (left to right): Diesel, B10, B20, B30, E10, E20, E30, M10, M10+B5, M20+B10

Y-axis: 34,0% to 36,5%

Brake thermal efficiency - 50 % load

Categories (left to right): Diesel, B10, B20, B30, E10, E20, E30, M10, M10+B5, M20+B10

Y-axis: 37,5% to 40,0%
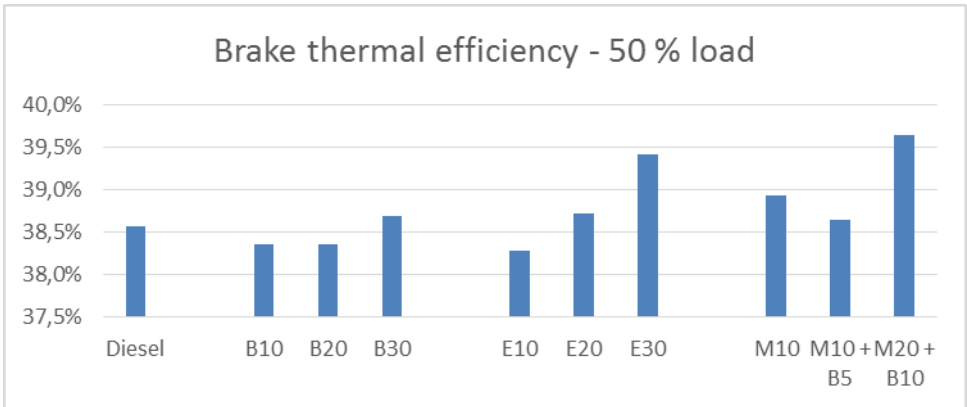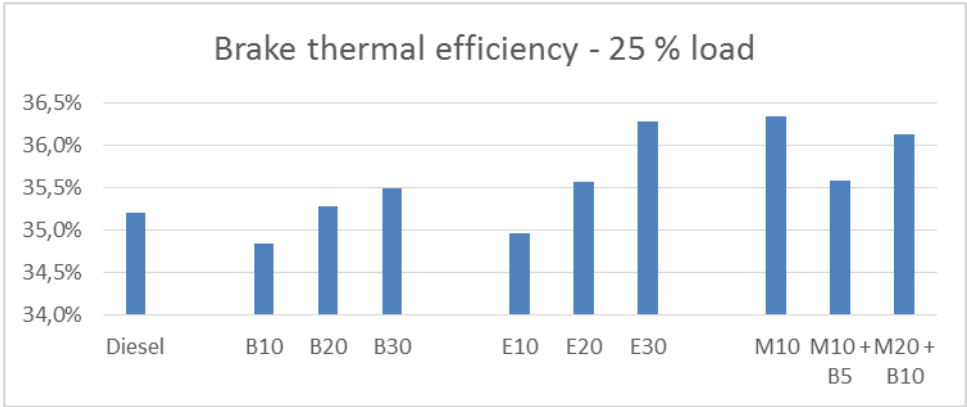
The calculation of brake thermal efficiencies in loaded condition is clearly demonstrating that the efficiency is affected with the alcohol blends.

Butanol does not appear to have as strong an effect as ethanol or methanol, but it is also the alcohol with the lowest ratio of oxygen to carbon, and more similar to a straight chain alkane than the two simpler alcohols.

Ethanol provides a large increase in thermal efficiency with 20 % and 30 % concentration in the blend, but the efficiency is actually decreasing when only 10 % ethanol is used.

Methanol also provides a large increase in thermal efficiency. The trend is strong when looking at the difference between M10 + B5 and M20 + B10 alone. The result with pure methanol compared to the combination with 5 % butanol is however indicating that butanol has a strong influence on the way that methanol behaves in the combustion of the fuel blend.

It must be remembered however, that although care was taken in adjusting the combustion timing with the purpose of comparing the emissions of the fuel blends, there may remain a large potential for optimizing fuel consumption, not only by proper injection timing, but also by the use of fuel additives to restore the cetane number.

Conclusion

Blending of the simple alcohols methanol, ethanol and butanol with diesel can contribute to a significant reduction in particulate matter formation. This may be used to reduce emissions of particulate matter from engines that are not fitted with particulate filters or other means of capturing the particulate matter.

The most effective alcohols for reducing particulate formation are methanol and ethanol. Butanol does not have as great potential as a single blending component, but it may be an important cosolvent for methanol and thereby assist in a more efficient particulate reduction.

Emission of other harmful components such as aldehydes, carbon monoxide and unburned hydrocarbons are mainly increasing when the engine is not under load. These emissions may need to be handled by oxidizing catalysts,

if improvements to the fuel composition and engine tuning are not enough to reduce these emissions to an acceptable level. The same emissions are however not increasing from the reference diesel at higher loads, which indicates that the engine is not very sensitive to the presence of alcohol in the fuel, when the combustion temperature is higher.

The formation of nitrogen oxide is slightly increasing at 25 % load with the use of alcohol blends, possibly due to an increased fraction of the fuel burning in premixed combustion and therefore an overall faster combustion at elevated temperatures. At 50 % load however, it appears that there is little difference between the diesel reference and the alcohol blends. Alcohol blends can therefore not contribute to emissions of nitrogen oxides.

The lower heating value of the fuel is reduced with alcohol blends, and the brake specific fuel consumption is therefore increasing. By measuring the fuel consumption and calculating the lower heating value of the fuels, it was however found that the brake thermal efficiency of the engine is actually increasing significantly when using alcohol blends at 25 and 50 % load. This effect may be due to faster combustion and possibly a reduction in heat losses. The thermal efficiency is however strongly dependent on the timing of the combustion, which in this experiment was monitored and advanced to match the diesel reference measurements. If alcohol blends are used without proper adjustment of injection timing, it is more likely that the thermal efficiency will decrease due to the prolonged ignition delay.

References

1. An experimental investigation on effects of methanol blended diesel fuels to engine performance and emissions of a diesel engine.

Murat Ciniviz, Hüsein Köse, Eyüb Canli and Özgür Solmaz. Selcuk University, Turkey.

2. Formulation of stable mixtures of diesel fuels with methanol. Azev V.S., Gerasimova G.N, Luneva V.V. Chemistry and Technology of Fuels and Oils. (1985)

3. Ethanol-diesel fuel blends – a review. Alan C. Hansen, Qin Zhang, Peter W.L. Lyne. Bioresource Technology vol. 96 pp. 277-285. (2004)

4. Advances in diesel-alcohol blends and their effects on performance and emissions of diesel engines. Satish Kumar, Jae Hyun Cho, Jaedeuk Park, Il Moon. Elsevier – renewable and sustainable energy reviews vol. 22 pp 46-72. (2013)

5. Properties of diesel-alcohol blends. L. Hajba, Z. Eller, E. Nagy, J. Hancsok. Hungarian Journal of Industrial Chemistry Vol. 39 pp. 349-352 (2011).

6. Stability of diesel-bioethanol blends for use in diesel engines. Magin Lapuerta, Octavio Armas, Reyes Garcia-Contreras. Fuel vol. 86, pp. 1351-1357. (2007).

7. Exhaust emissions with ethanol or n-butanol diesel fuel blends during transient operation: A review. Evangelos G. Giakoumis, Constantine D. Rakapoulos, Athanasios M. Dimaratos, Dimitrios C.

Rakapoulos. Renewable and Sustainable Energy Reviews vol. 17, pp. 170-190. (2013).

8. Review Paper: Ethanol-diesel fuel blends – a review. Alan C. Hansen, Qin Zhang, Peter W.L. Lyne. Elsevier - Bioresource Technology pp 277-285. (2004).

9. E-diesel – Demonstration test in Denmark. Urban Löfvenberg, Akzo Nobel Surface Chemistry. 2002

10. Emission measurement and calculations of two trucks – one with diesel and one with a diesel ethanol mixture. Ken Friis Hansen, Flemming Bak, Michael Grouleff Jensen. Danish technological Institute, 2002.

11. A comprehensive review on the application of emulsions as an alternative fuel for diesel engines. Biplab K. Depnath, Ujjwal K. Saha, Niranjan Sahoo. Renewable and Sustainable Energy Rewievs vol. 42, pp. 196-211. (2013).

12. Nanoemulsions: formation, structure, and physical properties. Mason TG, Wilking JN, Meleson K, Chang CB, Graves SM, Journal of Physics: Condensed Matter, 2006, 18(41): R635-R666

13. A novel emulsifier for ethanol-diesel blends and its effect on performance and emissions of diesel engine. Jilin Lei, Lizhong Shen, Yuhua Bi, Hong Chen. Fuel, vol. 93, pp. 305-311. (2012)

14. Multicylinder C.I. Diesel engine tests with unstabilized emulsion of water and ethanol in diesel fuel. Angelo de Vita. SAE 890450. (1989)

15. Use of alcohol-in-diesel fuel emulsions and solutions in a medium-speed diesel engine. Quentin A. Baker, Southwest Research Institute. SAE 810254 (1981)

16. Methanol as an alternative transportation fuel in the US: Options for sustainable and /or energy-secure transportation. L. Bromberg and W.K. Cheng. Sloan Automotive Laboratory, Massachusetts Institute of Technology. (2010).

17. Progress in the production and application of n-butanol as a biofuel. Chao Jin, Mingfa Yao, Haifeng Liu, Chia-fon F. Leed, Jing Ji. Renewable and Sustainable Energy Reviews 15 4080–4106. (2011)

18. Combustion process investigation in a high speed diesel engine fuelled with n-butanol diesel blend by conventional methods and optical diagnostics. S.S. Merola*, C. Tornatore, S.E. Iannuzzi, L. Marchitto, G. Valentino. Renewable Energy vol. 64 pp. 225-237 (2014)

19. Biomass Energy Data Book. Stacy C. Davis and Susan W. Diegel (Oak Ridge National Laboratory), Robert G. Boundy (Roltek, Inc.). (2014)

20. Classification of Surface-Active Agents by 'HLB'. Griffin, William C. Journal of the Society of Cosmetic Chemists 1 (5): 311-26 (1949)

Part D:

Report/summary of common workshop for Advanced Motor Fuels IA and Combustion IA

Authors: Jesper Schramm

Purpose

The IEA Implementing Agreements for Advanced Motor Fuels (IEA AMF) and Combustion (IEA Combustion) respectively have common interests in studying application of alcohols for diesel type engines. Therefore a common seminar/workshop was arranged for participants from both IA's in order to exchange experiences and to approach mutual coordinated progress in development of the area. The IA participants further invited relevant people in order to cover an, as broad as possible, aspect of the theme. The participants were:

Johan Danbratt	Wärtsilä
Eelco Dekker	Methanol Institute
Troels Dyhr Pedersen	Teknologisk Institut
Lennart Haraldson	Wärtsilä
Youmin Haug	Chalmers
Timo Huthisaari	NEOT
Anders Johansson	Swedish Energy Agency
Bengt Johansson	Lund University

Mads Jørgensen	DTU
Per Sune Koustrup	Statoil
Martti Larmi	Aalto University
Nils-Olof Nylund	VTT
Bengt Ramne	Chalmers
Mette Rasmussen	DTU
Teemu Sarjovaara	Aalto University
Jesper Schramm	DTU
Johan Sjöholm	MAN Diesel & Turbo
Thomas Stenhede	Wärtsilä
Toni Stojcevski	Wärtsilä
Martin Tunér	Lund University
Gerardo Valentino	Istituto Motori, Napoli

Agenda

The agenda for the meeting was as follows:

Application of Alcohols in Diesel Engines
Common seminar for IEA Combustion IA and IEA Advanced Motor Fuels IA

Date: February 26, 2015

Venue: FIRST HOTEL KONG FREDERIK
                        Vester Voldgade 25
                        DK-1552 Copenhagen

Time	
9.00-9.15	**Check-in, welcome**
9.15-9.30	**Purpose of meeting** Jesper Schramm, DTU Denmark
9.30-10.00	**Scania engine performance on ethanol with 3 different additives** Timo Huthisaari, NEOT
10.00-10.30	**Research engine performance on ethanol with 3 different additives** Mads Jørgensen, DTU Denmark
10.30-10.45	**Coffee**
10.45-11.15	**Dual fuel ethanol/diesel engine experiments** Martti Larmi, Aalto University, Finland
11.15-11.30	**Optical measurements on butanol/diesel blends in a CI engine** Jesper Schramm, DTU Denmark For Gerardo Valentino, Instituti Motori, Italy
11.30-12.00	**Compression Ignition of neat alcohols, the Partially Premixed Combustion** Martin Turnér, Lund University, Sweden
12.00-13.00	**Lunch**
13.00-14.00	**How can we best implement alcohols in diesel engines?** **(based on present knowledge)** Common discussion
14.00-15.00	**Which information do we need to go further?** **(should we initiate new projects?)** Common discussion
15.00-15.30	**Coffee**
15.30-16.00	**Preparing the final report** Common discussion

Engine Technology

Relevant alcohols discussed in relation to diesel engine application where methanol (M), ethanol (E) and higher alcohols – in this case butanol (BU) was discussed through the activities at Instituto Motori (IM), Italy and Technological Institute (TI), Denmark.

M and E applications are known through different engine operation principles:

- As fuel blends together with diesel
- Neat M and E with ignition improver
- Together with diesel in a Dual Fuel (DF) engine

Finally neat M and E or M and E as drop-in fuels with gasoline may also be applied in Spark Ignition (SI) engines, if the goal simply is to replace diesel fuels.

As fuel blends together with diesel

The experience from TI, Denmark showed that blending of the simple alcohols M, E and BU with diesel can contribute to a significant reduction in particulate matter formation. This may be used to reduce emissions of particulate matter from engines that are not fitted with particulate filters or other means of capturing the particulate matter. The most effective alcohols for reducing particulate formation are M and E. BU does not have as great potential as a single blend component, but it may be an important co solvent for M and thereby assist in a more efficient particulate reduction. IM, however, concluded that as an overall conclusion, it may be stated that the lower cetane number of the butanol-diesel blends extend the ignition delay, allowing more time for mixing, which in turn would reduce NOx and

particulate emissions simultaneously.

Neat M and E with ignition improver

This technology has been applied in Scania engines since the 1970'ties. Ethanol with additives has been demonstrated intensively in busses in Sweden. The main problem with the technology has been the price of the fuel/fuel additive. In recent years new additives have entered the market, and the fuel price is therefore expected to decrease.

VTT, Finland and DTU, Denmark presented experimental results from application of 3 different additives in E using a Scania production engine and a research engine respectively.

Scania's ethanol engine delivers what it is supposed to deliver, diesel-like efficiency (42.5 %) and a NO_x level of 2.0 g/kWh, when running on fuels that the engine is designed for. In addition, the testing demonstrated that the direct injection ethanol engine concept has some built-in multi-fuel capabilities. However, with neat methanol the engine didn't reach full power, and the injection periods on partial load were prolonged compared to ethanol operation. No real differences between additive packages could be found.

The experiments with the experimental engine at DTU confirmed the measurements on the Scania engine, but a more detailed picture of the heat release pattern did show differences between the three fuels/additives which calls for adjustments of injection strategy to the individual fuels.

Together with diesel in a Dual Fuel (DF) engine

ML presented results from Aalto University in Finland where a 7,4 litre and an 8,4 litre off-road diesel engine had been converted to DF ethanol operation and showed good engine operability. The results indicated that a

very simple conversion is possible if no serious concern to emissions needs to be taken. The development was aimed at the Brazilian market where emission regulations are not as stringent as in Europe. The second generation development for more stringent markets included more refined fuel injection strategy and other incentives like variations in: charge air temperature, lambda, compression ratio and oxidation catalyst. The results obtained showed that ethanol and E85 have potential as DF engine main fuel. There were, however, huge variations between engine operation points. NOx emissions decreased but THC and CO emissions increased significantly. The engine will need after treatment if existing European emissions legislation were to be fulfilled. Finally diesel injection timing should be adjusted to fully exploit the fuel.

M and E as a drop-in fuel with gasoline
M and E are already available in standard gasoline fuels. In amounts up to 10 %, no engine modifications are necessary.

PPC Engine Technology
MT presented the opportunities in applying PPC (Partially Premixed Combustion) technology in vehicle engines. This technique shows excellent opportunities for applying gasoline like fuels in diesel like engines. Lund University has studied this technology intensively, and ethanol has been shown to have the highest efficiency of all fuels in this context. Many fuels are suited for this type of engine, which makes it possible to "change the fuel on the fly". Most likely future cars will be able to run on multiple fuels.
Different fuels/efficiencies for HD engines are shown in Figure 1, which was presented by MT.
In the same figure it is seen that alternative fuels and also ethanol/PPC
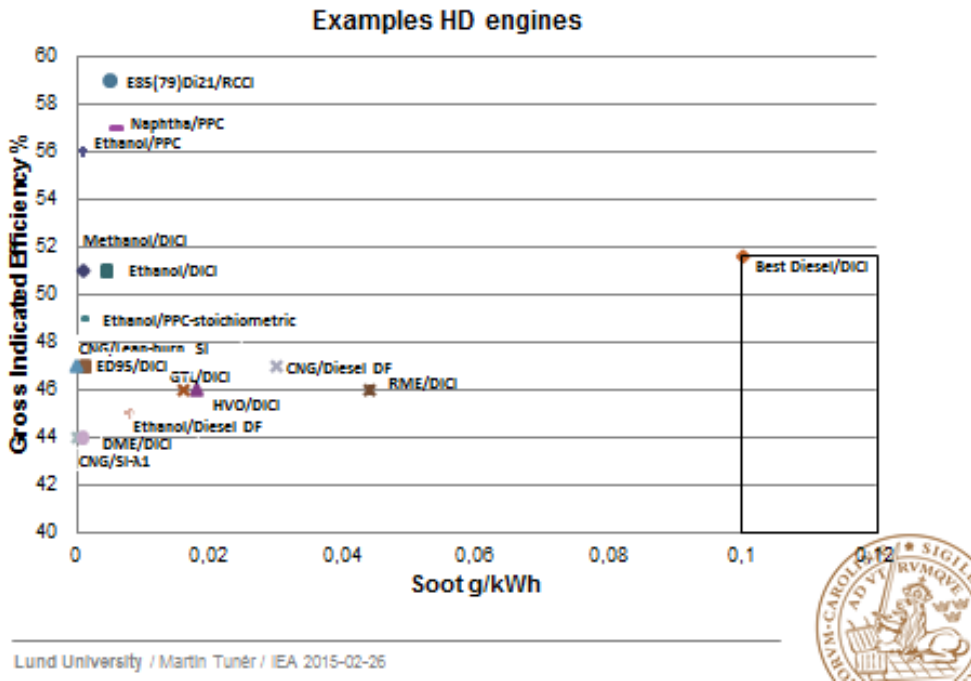
shows superior particulate emissions performance.

Figure 1. Efficiencies vs soot emissions, applying PPC and other engine technology

Evaluation of meeting

General

The meeting consisted of participants from the two implementing agreements and persons invited by representatives of the two IA's. It was remarkable that many of the participants were very interested in M. This indicated that the M society find the time relevant to look at the

opportunities to introduce M in larger scale, again reflecting that M might be a relevant fuel as a substitution for diesel to a larger extend in the future.

BJ mentioned that emission reduction doesn't seem to be a viable argument for introducing alternative fuels in road vehicle anymore since the engine technology is so developed that many experts see cars as a cleaning plant rather than a nuisance for the environment. CO_2 emissions being an exception in this context. This view was supported by several participants.

TH and others mentioned that in order to get political support for introduction of alcohols, it is necessary to show solutions (demonstration projects etc.).

Glycerol should be considered as a potential fuel for diesel engines. The availability of this fuels is, however, limited (BJ).

Methanol (M)

LNG is a very popular fuel in Europe, and the M society will have to fight for the introduction of M on markets.

New rules for sulfur content in marine fuels makes M an obvious alternative in the marine sector (TS and others). LNG seems to be a very likely alternative fuel in the marine sector, however, M will protrude if the problems with unburned methane are not solved (ML).

WTW efficiency for M is very high, and biomass based M has one of the highest carbon dioxide savings potential.

Converting a diesel engine to M calls for a redesign of the fuel system, due to the lower LHV of the fuel (JS and others).

M is the cheapest fuel on the market today and it is peculiar that this is not well known (information distribution is needed). However, people have invested otherwise, therefore, it is rather a political issue which is the

barrier for further M progress than the technical issues (ED).

M is a good fuel to optimize the engine for, because it is a well defined fuel with known characteristics (BJ).

M/E as drop-in fuel in SI engines is an obvious way to start introduction of these alcohols into an infrastructure (ED and others).

<u>Ethanol (E)</u>

ED mentioned a Horizon2020 project where application of gasoline engine fuels in diesel engines is investigated. This is connected to the information under "PPC Engine Technology" and E application is an option in this context. Application of E in PPC engines could very likely be the future technology applied, maybe in connection with a multi tank concept where the engine will run on several fuels. The issue here is the thorough combustion/engine control, which is needed. However, HEV's seems to be an obvious way to progress as well. In this type of vehicle engine the steady state engine operation makes engine control an easier task to solve.

TH mentioned that E is the cheapest fuel of all to pump. Production of E from biomass results in production of useful bi-products as well.

M/E as drop-in fuel in SI engines is an obvious way to start introduction of these alcohols into an infrastructure (ED and others).

Like with M the LHV is a problem, which needs to be addressed, although this is more a tank capacity issue than a fuel flow issue.