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Effect of charge air temperature on E85 dual-fuel diesel combustion

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ABSTRACT

The scope of this experimental engine study was to explore the effect of charge air temperature on E85 ethanol/gasoline blend dual-fuel combustion. The E85 was injected in the intake manifold, while diesel fuel was direct injected into the cylinder with a common-rail injection system. The study focused on medium and high load conditions at 1500 rpm. The diesel injection timing parameters were kept in every test case the same as in the original diesel production engine. The results showed that charge air temperature influenced the ignition delay, the cylinder pressure rise rate (PRR) and the maximum cylinder pressure by altering the E85 combustion phasing, while the changes on the diesel fuel combustion were minor. Lower charge air temperatures allowed higher E85 injection rates without the risk of a too high PRR, especially at high load conditions. The increase of the E85 rate allowed by lower charge air temperature, decreased nitrogen oxide emission, but simultaneously increased carbon monoxide and unburned total hydrocarbon emissions and decreased combustion efficiency.

1. Introduction

Limited crude oil reserves and concerns of global warming have roused an interest in alternative fuels on on-road and off-road vehicles [1–3]. For heavy duty and off-road vehicles the main alternative has been renewable diesel fuels, which would not require significant modifications on existing diesel engine technology. Dual-fuel technology is another option besides renewable diesel fuels, but it requires modifications on the existing diesel engine concept. In the dual-fuel concept the primary fuel is injected into the intake manifold at low pressure, typically below 10 bar. The primary fuel mixes with combustion air to form a homogenous mixture, which is ignited in the cylinder near the top dead center (TDC) by injecting diesel fuel into the cylinder. Diesel injection timing is used to control the primary fuel combustion. Typically dual-fuel technology is associated with natural gas engines, but the technology can be adopted for several other fuel alternatives. Typically these fuels are suitable for spark ignition (SI) engines by having a research octane number (RON) between 95 and 110, while having a low cetane number [4].

In this study, a commercial blend of ethanol and gasoline is used as primary fuel. The blend is commonly known as E85, where the maximum ethanol mass fraction is 85%, while rest of the mixture is gasoline-like light hydrocarbons. The motivation, for selecting the E85 blend instead of neat ethanol, was that E85 has already existing distribution network in many countries, while neat ethanol as transport fuel is only available in few countries. The fuel blend has been developed for SI engines, and the reason for the light hydrocarbon addition is the poor ignition properties of ethanol at cold conditions [5,6]. There are several studies on dual-fuel combustion with different liquid primary fuels, but little knowledge of the capability of the E85 blend on dual-fuel engines [7–15]. E85 would have several advantages in comparison to other more typical fuel alternatives, since it is already available on several markets, it is liquid in atmospheric conditions and it has low carbon-dioxide emission, because it is typically manufactured from waste or crop.

In a previous study by Sarjovaara et al. [16] the focus was on the effects of diesel injection timing parameters on ethanol dual-fuel combustion in a heavy-duty diesel engine equipped with an ethanol port fuel injection (PFI) system. In their study, the diesel fuel injection consisted of two separate injections, which was found to have a significant effect on the cylinder pressure rise rate (PRR) and the maximum ethanol portion. Sarjovaara et al. used the value of 10 bar/CA° as the limit for the maximum acceptable PRR, where CA stands for crank angle. When only a single injection/cycle was used, the ethanol rates were low due to the tendency of high PRR, whereas the injection timing did not have significant effect on it. When two injections/cycle was used, the timing of the later diesel injection was found to have more effect on combustion and on PRR than the first injection. They achieved a maximum ethanol rate of 90% by energy content in their study [16].

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In another dual-fuel study Sarjovaara et al. [17] focused on the E85 blend as a dual-fuel primary fuel. The engine used in their study was a heavy-duty diesel engine equipped with an E85 PFI system. The diesel injection timing parameters were kept similar to a base production diesel engine. They found that the maximum E85 rate of 89% by energy content, was achieved at a medium load at 1500 rpm, while at high and low loads the rates were clearly lower (<50%) as also with 2300 rpm engine speed. At high loads the auto-ignition occurred and limited the maximum E85 rate. In all studied cases, the carbon-monoxide (CO) and total unburned hydro-carbon (THC) emissions increased significantly as E85 was introduced, while in most of the cases the nitrogen-oxide (NO_x) emissions decreased. Soot emissions were low in all cases [17].

In a study by Padala et al. [18], ethanol dual-fuel was studied with a 0.498 l single cylinder diesel engine. They varied the ethanol rate from 0% to 80%, and they had only a single injection/cycle and the start of injection (SOI) was varied from 13 to −1 CA° before TDC (BTDC). In the study, they achieved 10% increase in efficiency with an ethanol rate of 60%, while with higher rates misfiring occurred. The increase in efficiency was mainly due a shorter burn duration and more advantageous combustion phasing with ethanol. NO_x, CO and THC emissions increased as the ethanol share was increased, and with higher rates than 20% the smoke emissions were negligible [18].

Rezende et al. [19] studied ethanol dual-fuel combustion with a heavy-duty diesel engine with an ethanol PFI system. They reported that ethanol substitution of 12–57%, under different load conditions, increased engine efficiency and decreased soot and NO_x emissions. They installed the engine also to a heavy-duty vehicle operating in off-road conditions and achieved almost 38% volume substitution of diesel fuel in their field tests [19].

In his study, Tutak [20] investigated E85 as a fuel for a dual-fuel engine. The engine utilized in the study was a naturally aspired 2.9 l diesel engine equipped with an E85 PFI system. He studied the E85 rates up to 90% (by energy content) and found that at low loads E85 decreased cylinder pressure, while at high loads the cylinder pressures increased as the E85 was introduced. The CO and THC emissions increased considerably, but NO_x emissions decreased as did also brake thermal efficiency (BTE) [20].

Splitter and Reitz [21] studied fuel reactivity and its effects on dual-fuel compression ignition engine. In the study they used a single cylinder 2.44 l test engine and utilized a Reactivity Controlled Compression Ignition (RCCI) combustion strategy. They used E85 as port injected low reactivity fuel, while diesel fuel and 2-ethylhexyl nitrate doped gasoline were the alternatives for direct injected high reactivity fuel. They also studied the effect of charge air temperature and pressure besides of the fuel effects. Their results demonstrate that, by adjusting the engine conditions and fuel reactivity, RCCI strategy can increase engine efficiency. As part of their findings, decreasing the charge air temperature widened the engine operational window making it suitable for RCCI combustion [21].

Papagiannakis [22] studied inlet air preheating and exhaust gas recirculating (EGR) on a dual-fuel engine. He did the study by simulating a single cylinder high speed diesel engine. The simulation model he utilized was phenomenological two-zone model. In the study the inlet temperature was varied between 35 and 80 °C and the EGR rate from 0% to 26%. He found that increasing the inlet temperature decreased the ignition delay and increased the peak cylinder pressure, while the EGR tended to affect in the opposite way [22].

In their study, Park et al. [23] compared bioethanol and gasoline as premixed fuel on a dual-fuel engine and they used a soybean derived biodiesel as the direct injection fuel. The research was carried out with single cylinder 0.373 l engine. They found that dual-fuel combustion decreased ignition delay and NO_x emissions while THC and CO emissions increased. It was interesting that they found significant differences on combustion and emission between gasoline and bioethanol based dual-fuels. With bioethanol ignition delay, fuel consumption and THC emissions were higher than with gasoline, whereas NO_x emissions were significantly lower. The CO emissions were on same level with both premixed fuel alternatives [23].

This study was a continuation of the previous studies made by Sarjovaara et al. [16,17]. The goal of this research was to study how charge air temperature affects E85 dual-fuel combustion at different load conditions. The research focus was on:

- Combustion characteristics.
- Maximum E85 rate by energy.
- Exhaust gas emissions.
- Engine efficiency.

Despite of the many studies made on dual-fuel combustion with different fuel selections and research approaches, there is still a clear lack of basic information concerning dual-fuel combustion fundamentals and this study aims to offer more information to further understand the parameter dependencies of this combustion technique and especially of the usage of E85 as the premixed fuel.

2. Experiments

2.1. Research apparatus and fuels

The research in this study was carried out with a 6-cylinder 7.4 l heavy-duty diesel engine (Table 1). The engine was equipped with a common-rail diesel injection system, similar as in the production base engine, and an intake manifold E85 injection system, which was custom made. The intake manifold was also custom made with an individual intake runner for each six cylinders. The diesel fuel injection pressure and timing values were same as in the production diesel engine and only the main diesel injection duration was decreased when ethanol was applied to keep the engine torque constant. The engine was connected the Schenk W400 eddy-current dynamometer equipped with force transducer. The test engine had a water-to-air charge air cooler and charge air temperature was adjusted by controlling the cooling water flow with a manual valve.

The THC emissions were measured using a J.U.M. VE7 analyzer based on the hydrogen flame ionization detector (FID) method. A sick Sidor multi-component analyzer was used to measure O_2 utilizing a paramagnetic cell, while CO and CO_2 were measured using a Non-Dispersive Infra-Red (NDIR) technique. NO_x emissions were measured with an Ecophysics CLD 880 chemiluminescence analyzer. Smoke emissions were measured using an AVL 415S Smoke Meter. A Kistler 6125 uncooled piezoelectric sensor was installed on one cylinder to measure the cylinder pressure with a 0.25 CA° resolution. Diesel fuel mass flow was measured with an AVL Fuel Meter 733 and the mass flow of E85 was measured with a Rheonik RHM03 coriolis mass flow meter. The experimental setup is described in Fig. 1.

The oxygen containing exhaust gas HC species are challenging measure with FID analyzer, since its response factor for oxygen containing species is lower than for no oxygen containing species. As the premixed fuel in this study contained oxygen (ethanol) and the analyzer calibration was performed with propane (no oxygen), there might be uncertainties in the THC measurement values. The uncertainty of the THC measurements depends on concentration ratio of unburned ethanol and no oxygen containing hydrocarbons, e.g. ethylene, on exhaust gases. Since in these test there was no detailed measurement for HC characterization, exact uncertainty

Table 1
The test engine specifications.

Displaced volume	7.4 l
Number of cylinders	6
Rated power	120 kW
Compression ratio	16.5:1
Number of valves per cylinder	2
Rated speed	2300 rpm
External EGR	No
Exhaust gas after-treatment	No

Table 2
The measurement in-accuracies.

Measurement	In-accuracy
Load	±0.1%
Speed	±0.1%
Temperature	±1.0 °C
CO	±3 ppm
THC	<30 ppm[a]
NO_x	<40 ppm
Smoke	±0.005 FSN
Pressures	±0.1%
Fuel mass flows	±0.12%

[a] For non oxygen containing species.

is difficult to define. The analyzer manufacturer J.U.M. Engineering promises this additional error to be <1.2%. The error can be also considered to be systematical and consistent for all E85 cases. The measurement in-accuracies are presented in Table 2.

The studied fuels were a commercial E85 bio-ethanol/gasoline blend and EN590 diesel fuel. The properties of the fuels used in the study are presented in Table 3.

2.2. Experimental conditions

The effect of charge air temperature was studied at two load conditions (Table 4). In both load cases, the charge air temperature was gradually decreased from no-cooling conditions to the temperature of ca. 40–50° Celsius by adjusting the charge air cooler water flow. For both test cases the diesel injection timing values were kept the same as in the diesel reference tests. In both load cases the diesel reference injection consisted of pilot and main injection.

In the first test phase the E85 injection quantity was kept constant at every load case. The E85 rate was defined for both load cases by the maximum E85 rate with PRR value below 10 bar/CA° at no-cooling charge air conditions. In the second test phase the maximum E85 rates at the lowest charge air temperature conditions were studied.

2.3. Analysis Methods

Cylinder pressure was measured with a 0.25 CA° resolution, and pressure values of 20 consecutive cycles were averaged to filter the cycle-to-cycle fluctuation of cylinder pressure. The average values were then filtered with a zero phase-shift Butterworth low-pass filter, with a cut-off frequency of 2000 Hz. This was done to filter the high frequency noise from the measurement signal. The filtered cylinder pressure data was used for heat release rate (HRR) and cumulative heat-release (HR) analyses, which were based on Eq. (1).

$$\frac{dQ_n}{dt} = \frac{\gamma}{\gamma - 1} p \frac{dV}{dt} + \frac{1}{\gamma - 1} V \frac{dp}{dt}, \quad (1)$$

where Q_n denotes net heat-release, t is time, γ is the specific heat capacities ratio, V is volume and p stands for cylinder pressure [24]. The calculated HRR values were summarized to represent cumulative heat release. The HRR analysis method utilized in this study did not take into account E85 evaporation during the compression, blow-by or heat transfer. The ignition delays in this study were defined as a relative to main injection start and the CA° value, when HR reached 0 J after the diesel fuel evaporation phase, was used to indicate the ignition.

The specific exhaust gas emissions were calculated from the measurement results based on the ISO 8178-1 emissions measurement standard [25]. The combustion efficiency was derived from the emission measurement results of THC and CO as presented by Heywood [24]. The brake thermal efficiency has been calculated as ratio of brake output power and thermal energy of the supplied fuels.

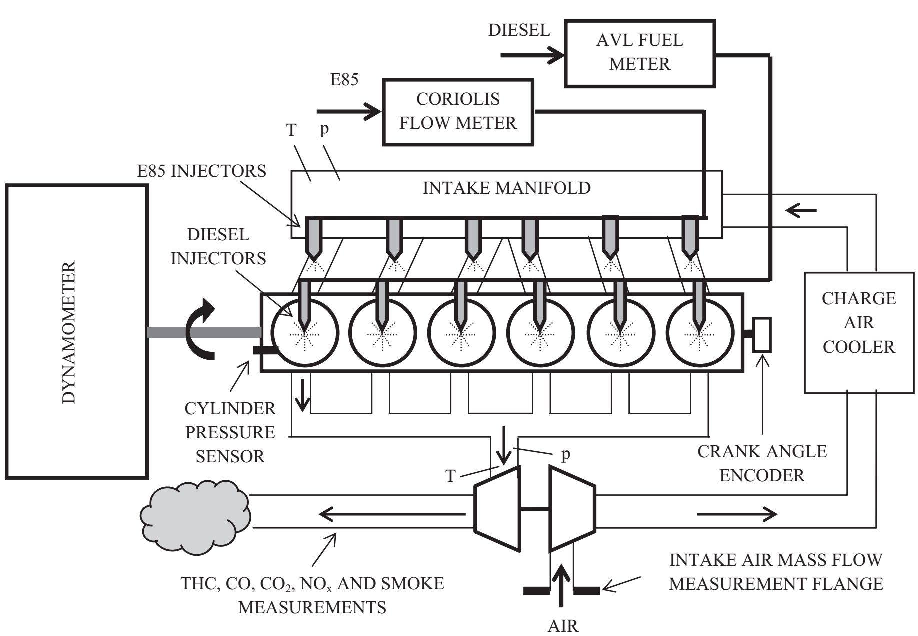

Fig. 1. The experimental setup utilized in this study.

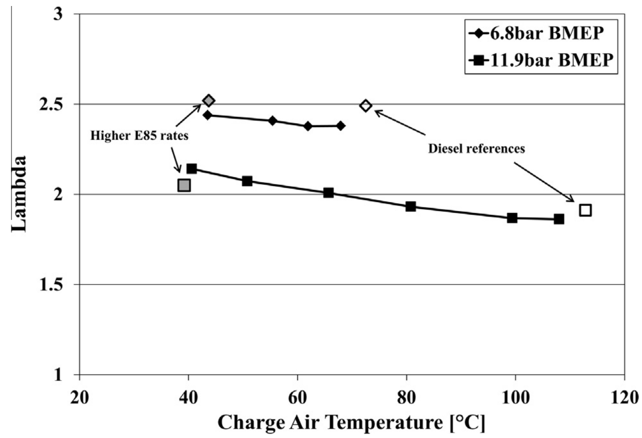

Fig. 2. Lambda values at different charge air temperatures. The curves with black symbols present the constant E85 rate cases. The symbols with gray fill present the value at higher E85 rate, while the symbols with no fill present the diesel reference case.

Table 3
The fuel properties.

		Diesel EN590	E85
Density	kg/m^3	840	785
Cetane number	–	53	~11[a] (ethanol) 20-25[a] (gasoline)
Octane number	–	–	105
Carbon	wt-%	85.8	55.4
Hydrogen	wt-%	13.6	13.2
Oxygen	wt-%	–	31.4
Stoichiometric air-fuel ratio	–	14.5	9.5
95% evaporated	°C	340	78[b] (ethanol boiling point) 190[b] (gasoline)
Lower heating value	MJ/kg	43.2	28.7
Heat of evaporization	kJ/kg	260[a]	780[a]
Autoignition temperature	°C	230[a]	420[a] (ethanol) 300[a] (gasoline)

[a] Values are based on fuel analysis, except [20].
[b] Values are based on fuel analysis, except [5].

3. Results and discussion

Table 5 presents the boundary conditions of each test case. As the charge air temperature was lowered in the load points the charge air pressures and the exhaust gas pressures (pre-turbo) remained almost constant, whereas the exhaust gas temperature decreased significantly. SOI values in Table 5 are based on the start of the electrical injection pulse.

The lambda (λ) values, defined by intake air and fuel mass flows, are shown in Fig. 2. The values were rather constant in both load cases. A minor change at the highest charge air temperatures, when the E85 was introduced, was seen. Also at the lowest temperatures, when the E85 rate was increased, the λ changed slightly. When the charge air temperature was lowered the λ seemed to have a slightly increasing trend, but the differences were below 0.2.

3.1. Cylinder pressure and heat release rates

The cylinder pressures and HRR values at 1500 rpm and 6.8 bar BMEP are shown in Fig. 3. The compression pressures were slightly lower with E85, most likely due to fuel evaporation during the compression stroke. There were significant differences in maximum cylinder pressures between different charge air temperatures. At the highest temperatures the pressure peak value (95 bar) was close to the diesel reference value, but as the temperature was lowered to 55° and below, the peak value (87 bar) decreased significantly. Increasing the E85 rate at the lowest charge air temperature further decreased the peak pressure to 85 bar.

Table 4
The load conditions.

n (rpm)	Load (N m)	BMEP (bar)	Power (kW)
1500	400	6.8	63
1500	700	11.9	110

The HRR results in Fig. 3 were well in line with the corresponding cylinder pressures. The differences on HRR and the effect of the charge air temperature on E85 combustion after the TDC was even more obvious than on the pressure curves. There was clear trend, where HRR peak was lower and occurred later as the charge air temperature was lowered. It is interesting that charge air temperature and E85 rate at low temperatures seem to affect the flame front combustion of E85 combustion after the main diesel combustion more than the ignition delay or diesel fuel combustion.

The cumulative heat release at 1500 rpm and 6.8 bar BMEP are presented in Fig. 4. It was noteworthy that the ignition delay increased with E85, and that charge air temperature affected significantly the combustion after the 4 CA° ATDC. At this load, as the E85 was introduced, the ignition delay increased from 2.5 CA° of diesel reference case to 3.9 CA° of 68 °C E85 case and further increased to 5.0 CA° as the charge air temperature was decreased from 68 °C to 44 °C. As the E85 rate was increased from 68% to 75% at 44 °C, the ignition delay did not change. The late phase of combustion (after the 30 CA° ATDC) was very similar between the cases and there were no differences in cumulative heat releases between the E85 cases. The diesel reference had a higher HR level, while the slope was similar to the E85 cases.

Fig. 5 presents cylinder pressures and HRRs at 1500 rpm and 11.9 bar BMEP. The charge air temperature had significant effect on combustion at this load, especially since there was an auto-ignition tendency of E85 at the highest temperatures. The charge air temperatures below 99 °C changed the combustion behavior and at these temperatures there was no auto-ignition detected. At this load the HHR peak values occurred later as the charge air temperature decreased, but unlike at the 6.8 bar load, the HRR peak values increased at the same time. This could be explained by increased ignition delay of pre-mixed E85 as temperature decreased, and it also seemed that at the lowest temperatures the high peaks were caused by simultaneous main diesel fuel HRR and E85 HRR peaks. In some cases there was clear three stage combustion: pre-injection diesel, main injection diesel and pre-mixed E85.This feature was evident especially with higher E85 rates at the lowest charge air temperature.

The cumulative heat releases at the 11.9 bar load are presented in Fig. 6. The cumulative heat release indicated clearly the auto-ignition tendency of E85 at high charge air temperatures, as ignition delay was even negative (−0.2 CA°) for 108 °C case. In the E85 cases where auto-ignition did not occur, the ignition did not seem to significantly dependent on charge air temperature, as the ignition delay increased only 0.5 CA° when the temperature decreased from 81 °C to 41 °C. Further increase of the E85 rate increased the ignition delay to 3.5 CA°. Though the effect of the charge air temperature on the ignition delay was relatively small, it affected significantly the HR slope after the ignition due to the effect on the ignition and combustion of E85.

Table 5
The boundary conditions of the test cases. The pressures are gauge values.

n (rpm)	BMEP (bar)	E85 rate (energy) (%)	Charge temp. (°C)	Charge pressure (bar)	Exhaust gas temp. (°C)	Exhaust gas press (bar)	Pre. inj. SOI (CA°) BTDC	Main inj. SOI (CA°) BTDC
1500	6.8	0	73	0.5	385	0.5	15.5	7.5
	6.8	68	68	0.4	351	0.4	15.5	7.5
	6.8	68	62	0.5	351	0.4	15.5	7.5
	6.8	68	55	0.5	343	0.5	15.5	7.5
	6.8	68	44	0.5	335	0.5	15.5	7.5
	6.8	75	44	0.5	341	0.5	15.5	7.5
1500	11.9	0	113	1.0	530	0.7	15.4	6.5
	11.9	36	108	0.9	494	0.7	15.4	6.5
	11.9	36	99	0.9	485	0.7	15.4	6.5
	11.9	36	81	0.9	464	0.7	15.4	6.5
	11.9	36	66	0.9	450	0.7	15.4	6.5
	11.9	36	51	0.9	437	0.8	15.4	6.5
	11.9	36	41	0.9	428	0.8	15.4	6.5
	11.9	74	39	0.8	412	0.7	15.4	6.5

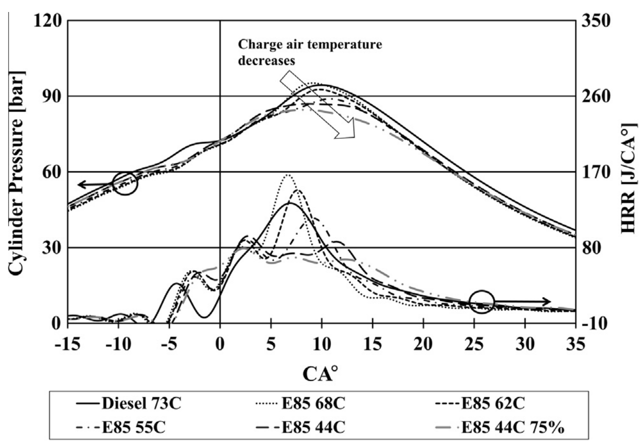

Fig. 3. The cylinder pressures and HRR at 1500 rpm and 6.8 bar BMEP.

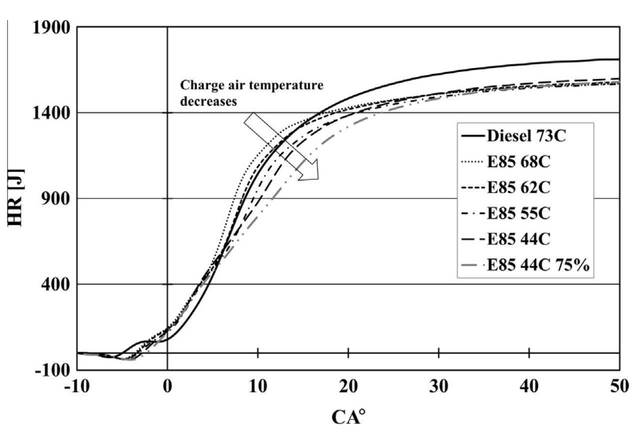

Fig. 4. The cumulative heat release rates at 1500 rpm and 6.8 bar BMEP.

The combustion in the E85 cases seemed to have three parts: pre-injection diesel fuel combustion, main injection diesel fuel combustion and E85 combustion. This three phase feature was evident in both loads. When excluding the cases with auto-ignition of E85, the pre-injection fuel ignition delay increased as E85 was introduced and as the charge air temperature was lowered. The HRR peak BTDC values were higher in these cases with E85, but the differences on cumulative HR were lower between the diesel reference and the E85 cases. This indicated that only a small amount of E85 burned during the pre-injection combustion and the difference on HRR was caused mainly by increased ignition

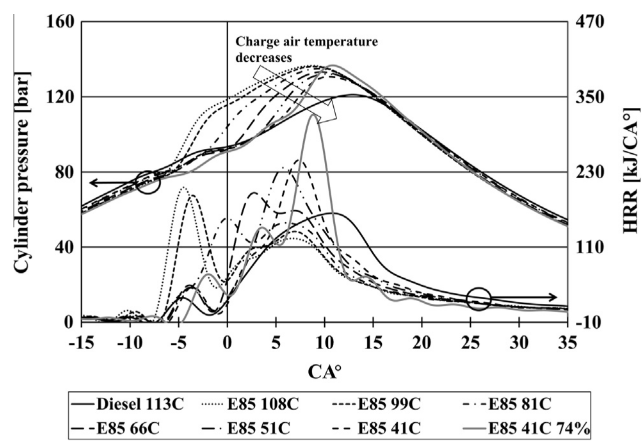

Fig. 5. The cylinder pressures and HRR at 1500 rpm and 11.9 bar BMEP.

delay. The analysis of the main injection diesel combustion was harder, since E85 combustion occurred during the same period. At the low load cases, where E85 combustion occurred after the HRR peak of the main diesel fuel combustion, the analysis was possible and showed that the ignition of the main diesel fuel was not notably influenced by E85 or the charge temperature. At high load cases the analysis was more challenging, since the E85 combustion occurred before the main diesel fuel combustion at charge air temperature higher than 51 °C, whereas at the 51 °C and 41 °C cases the E85 combustion occurred simultaneous with main diesel fuel combustion. The HRR was higher and had a higher slope in the E85 cases, indicating the high combustion velocity of premixed fuel and that part of the E85 burned simultaneous with diesel fuel. In most of the cases the three peak HRR structure was evident, while the charge air temperature had the biggest effect on E85 combustion. The reason for higher temperature dependency of E85 combustion was most likely the lower reactivity, i.e. low cetane number and high autoignition temperature, when compared to diesel fuel (Table 3).

3.2. Emissions

The THC and CO emissions at the tested cases are presented in Fig. 7. At both loads the THC emission were close to zero in the diesel reference cases. At the high load case the decrease of the charge air temperature increased the THC emissions slightly and the trend was linear. In the lower load case the effect of E85 addition was higher than at the high load case. Most likely this was due to the higher E85 rate at this lower load. At the lower load case the charge

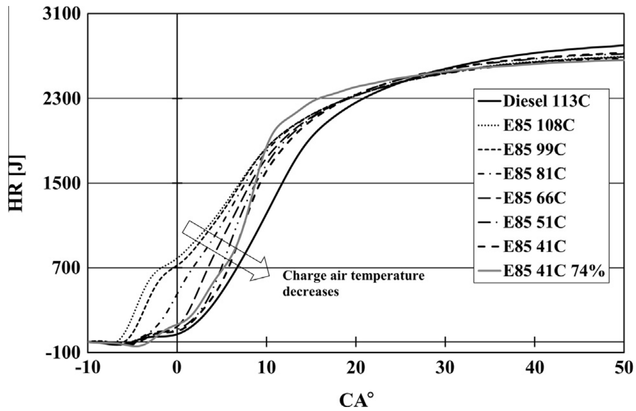

Fig. 6. The cumulative heat release rates at 1500 rpm and 11.9 bar BMEP.

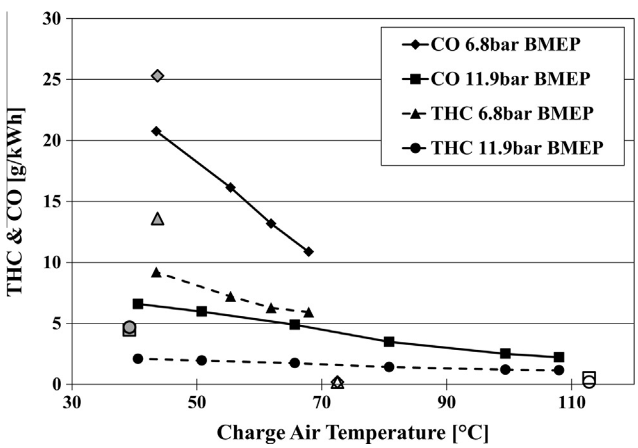

Fig. 7. The THC and the CO emissions. The symbols with gray fill present the value at the higher E85 rate, while the symbols with no fill present the diesel reference case.

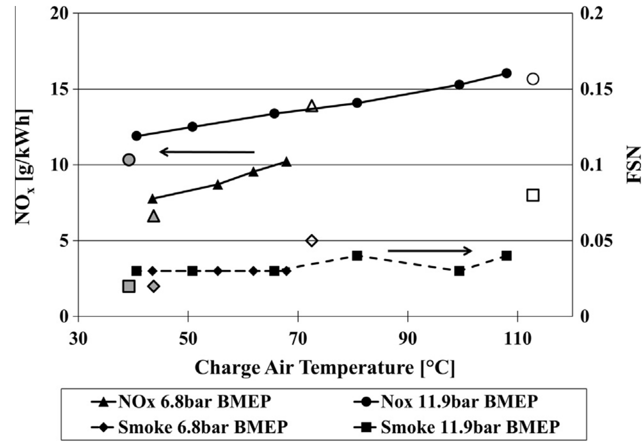

Fig. 8. The NO_x and smoke emissions. The symbols with gray fill present the value at the higher E85 rate, while the symbols with no fill present the diesel reference case.

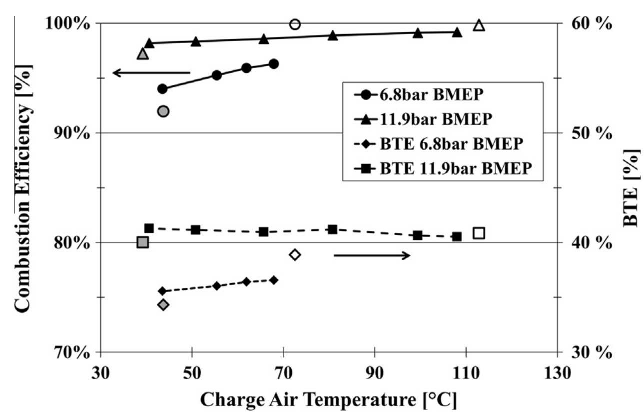

Fig. 9. The combustion and break thermal efficiencies. The symbols with gray fill present the value at the higher E85 rate, while the symbols with no fill present the diesel reference case.

air temperature had a strong effect on THC, as the THC emission increased clearly when the charge air temperature was lowered. As the E85 rate was increased at the lowest charge air temperature at both loads, THC emissions increased indicating a high rate of unburned fuel.

As the THC emissions were considered to indicate the rate of unburned fuel, the CO emissions indicated the rate of incomplete combustion. The main trends were rather similar as in THC emission (Fig. 7), but some clear differences were also found. The charge air temperature had a stronger and more linear effect on CO values and increasing the E85 rate had smaller effect than on THC values, while in the 11.9 bar BMEP case the E85 addition even decreased the CO emissions slightly. Both, CO and THC emissions, increased significantly in all E85 cases and a decrease of the charge air temperature tended to increase the emissions, especially in the low load case. With the E85, the CO and the THC emission values were so high that exhaust gas after-treatment would be most likely to be required, if any emission legislation is to be fulfilled.

The NO_x emissions (Fig. 8) had an opposite trend when compared to the THC and CO emissions. In the low load case the E85 addition at high charge air temperature decreased NO_x emission, while in the high load case NO_x emission increased slightly, where the behavior can be explained by the auto-ignition tendency. As the charge air temperature was lowered the NO_x emissions further decreased with very linear and equal trend. The increase of the E85 rate at the lowest charge temperature decreased the NO_x emission even more at both loads. In both load cases the smoke emissions were rather low (FSN < 0.1) and charge air temperature did not have significant effect on it. At the high charge air temperatures, the E85 affected the smoke emissions by decreasing them. E85 rate addition at the lowest charge air temperature cases had a consistent decreasing effect.

3.3. Efficiency

Fig. 9 presents both break thermal and combustion efficiencies. The combustion efficiency in both load cases had a consistent decreasing trend, whereas the BTE decreased in the low load case and in the high load cases it remained almost constant. In the high load case the difference between the two efficiencies can be explained by short combustion duration and more beneficial combustion phasing at lower charge air temperatures in the E85 cases, which compensated the decrease of combustion efficiency. Though the effect of the initial E85 addition and the decrease of the charge air temperature on the combustion efficiency was compensated by more beneficial combustion, the increase in the E85 rate at the lowest temperature point decreased the BTE.

4. Conclusions

In this study the effect of charge air temperature on E85 dual-fuel combustion was investigated. The engine tests were carried out at two loads, 6.8 and 11.9 bar BMEP, at 1500 rpm engine speed. In the tests the E85 share for both load cases was defined as the maximum E85 share in no charge air cooling conditions. The limiting factor for the maximum E85 rate in this study was the maximum acceptable pressure rise rate (PRR) of 10 bar/CA°. At both loads also a significantly higher E85 rate was studied at the lowest charge air temperature. All the E85 rates presented in this study were based on energy content.

The main conclusions of the study were:

- As the charge air temperature was lowered the maximum acceptable E85 rates increased and at the lowest temperatures E85 rates were approximately 75% at both tested loads.
- The introduction of E85 increased ignition delay, except at the higher load where auto-ignition of E85 occurred at high charge temperatures.
- The E85 dual fuel combustion could be divided to three parts. (1) Pre-injection diesel fuel combustion, (2) main injection diesel fuel combustion and (3) E85 combustion. The charge air temperature had a significant effect on E85 combustion by delaying it as the temperature decreased.
- The NO_x emissions decreased as the E85 was introduced at lower load but the trend was opposite at higher load. At both loads, as the charge temperature was lowered, the NO_x emissions decreased.
- The THC and the CO emission had similar increasing trend on both loads.
- The smoke emission was relatively low at every tested case (FSN < 0.1), but the E85 seemed to have a decreasing effect on it.
- The E85 and the decreasing of the charge temperature had a negative effect on the combustion efficiency at both loads. This was especially evident at the lower 6.8 bar BMEP load, where it decreased 8%.
- The effect on the brake thermal efficiency (BTE) varied between the two loads. At both loads the introduction of E85 had a negative effect on BTE, but at the high load the decrease of the charge air temperature slightly increased the BTE, which compensated the negative effect of the E85. This was probably due to the more advantageous combustion phasing of E85 at low temperatures.

The drawbacks of the CO and THC emissions as well as the loss in engine efficiency could be overcome, at least to some extent, with diesel injection parameter optimization, but still most likely exhaust gas after treatment would be required for the CO and THC emission to be at an acceptable level. This would require more research work and the authors will continue their work in this direction in the future.

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