1 **Effect of Charge Air Temperature on E85 Dual-Fuel Diesel Combustion**

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3 **Authors:**

4 Corresponding author:

5

6 Sarjovaara, Teemu ^{Aalto University}

7 Department of Energy Technology

8 Aalto University

9 00076 AALTO

10 Finland

11 teemu.sarjovaara@aalto.fi

12 +358503012039

13

14 Larmi, Martti ^{Aalto University}

15 Department of Energy Technology

16 Aalto University

17 00076 AALTO

18 Finland

19 martti.larmi@aalto.fi

20 +358505695625

21

22 Vuorinen, Ville ^{Aalto University}

23 Department of Energy Technology

24 Aalto University

25 00076 AALTO

26 Finland

27 ville.vuorinen@aalto.fi

28 +358503611471

29 **Abstract**

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

40 **Keywords**

41 Dual-fuel; ethanol; E85; charge air temperature

42 **1. Introduction**

43 Limited crude oil reserves and concerns of global warming have roused an interest in alternative fuels on on-road and
44 off-road vehicles. [1,2,3] For heavy duty and off-road vehicles the main alternative has been renewable diesel fuels,
45 which would not require significant modifications on existing diesel engine technology. Dual-fuel technology is another
46 option besides renewable diesel fuels, but it requires modifications on the existing diesel engine concept. In the dual-
47 fuel concept the primary fuel is injected into the intake manifold at low pressure, typically below 10 bar. The primary
48 fuel mixes with combustion air to form a homogenous mixture, which is ignited in the cylinder near the top dead centre
49 (TDC) by injecting diesel fuel into the cylinder. Diesel injection timing is used to control the primary fuel combustion.
50 Typically dual-fuel technology is associated with natural gas engines, but the technology can be adopted for several
51 other fuel alternatives. Typically these fuels are suitable for spark ignition (SI) engines by having a research octane
52 number (RON) between 95 and 110, while having a low cetane number. [4]

53 In this study, a commercial blend of ethanol and gasoline is used as primary fuel. The blend is commonly known as
54 E85, where the maximum ethanol mass fraction is 85%, while rest of the mixture is gasoline-like light hydrocarbons.
55 The motivation, for selecting the E85 blend instead of neat ethanol, was that E85 has already existing distribution
56 network in many countries, while neat ethanol as transport fuel is only available in few countries. The fuel blend has

57 been developed for SI engines, and the reason for the light hydrocarbon addition is the poor ignition properties of
58 ethanol at cold conditions. [5,6] There are several studies on dual-fuel combustion with different liquid primary fuels,
59 but little knowledge of the capability of the E85 blend on dual-fuel engines. [7-15] E85 would have several advantages
60 in comparison to other more typical fuel alternatives, since it is already available on several markets, it is liquid in
61 atmospheric conditions and it has low carbon-dioxide emission, because it is typically manufactured from waste or
62 crop.

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

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

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

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

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

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

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

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

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

- 117 • Combustion characteristics
- 118 • Maximum E85 rate by energy
- 119 • Exhaust gas emissions
- 120 • Engine efficiency

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

125 **2. Experiments**

126 **2.1 Research Apparatus and Fuels**

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

135 The THC emissions were measured using a J.U.M. VE7 analyser based on the hydrogen flame ionization detector (FID)
136 method. A Sick Sidor multi-component analyser was used to measure O₂ utilizing a paramagnetic cell, while CO and
137 CO₂ were measured using a Non-Dispersive Infra-Red (NDIR) technique. NO_x emissions were measured with an
138 Ecophysics CLD 880 chemiluminescence analyser. Smoke emissions were measured using an AVL 415S Smoke Meter.
139 A Kistler 6125 uncooled piezoelectric sensor was installed on one cylinder to measure the cylinder pressure with a 0.25
140 CA° resolution. Diesel fuel mass flow was measured with an AVL Fuel Meter 733 and the mass flow of E85 was
141 measured with a Rheonik RHM03 coriolis mass flow meter. The experimental setup is described in figure 1.

142 The oxygen containing exhaust gas HC species are challenging measure with FID analyzer, since its response factor for
 143 oxygen containing species is lower than for no oxygen containing species. As the premixed fuel in this study contained
 144 oxygen (ethanol) and the analyzer calibration was performed with propane (no oxygen), there might be uncertainties in
 145 the THC measurement values. The uncertainty of the THC measurements depends on concentration ratio of unburned
 146 ethanol and no oxygen containing hydrocarbons, e.g. ethylene, on exhaust gases. Since in these test there was no
 147 detailed measurement for HC characterization, exact uncertainty is difficult to define. The analyzer manufacturer
 148 J.U.M. Engineering promises this additional error to be <1.2%. The error can be also considered to be systematical and
 149 consistent for all E85 cases. The measurement in-accuracies are presented in figure 2. The studied fuels were a
 150 commercial E85 bio-ethanol/gasoline blend and EN590 diesel fuel. The properties of the fuels used in the study are
 151 presented in table 3.

152 **2.2 Experimental Conditions**

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

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

160 **2.3 Analysis Methods**

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

$$166 \quad \frac{dQ_n}{dt} = \frac{\gamma}{\gamma-1} p \frac{dV}{dt} + \frac{1}{\gamma-1} V \frac{dp}{dt} \quad , \quad (1)$$

167 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
 168 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 0J 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.

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 figure 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 1500rpm and 6.8bar BMEP are shown in figure 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 (95bar) was close to the diesel reference value, but as the temperature was lowered to 55° and below, the peak value (87bar) decreased significantly. Increasing the E85 rate at the lowest charge air temperature further decreased the peak pressure to 85bar.

The HRR results in figure 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.

196 The cumulative heat release at 1500 rpm and 6.8bar BMEP are presented in figure 4. It was noteworthy that the ignition
197 delay increased with E85, and that charge air temperature affected significantly the combustion after the 4CA° ATDC.
198 At this load, as the E85 was introduced, the ignition delay increased from 2.5CA° of diesel reference case to 3.9CA° of
199 68°C E85 case and further increased to 5.0CA° as the charge air temperature was decreased from 68°C to 44°. As the
200 E85 rate was increased from 68% to 75% at 44°C, the ignition delay did not change. The late phase of combustion
201 (after the 30CA° ATDC) was very similar between the cases and there were no differences in cumulative heat releases
202 between the E85 cases. The diesel reference had a higher HR level, while the slope was similar to the E85 cases.

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

212 The cumulative heat releases at the 11.9bar load are presented in figure 6. The cumulative heat release indicated clearly
213 the auto-ignition tendency of E85 at high charge air temperatures, as ignition delay was even negative (-0.2CA°) for
214 108°C case. In the E85 cases where auto-ignition did not occur, the ignition did not seem to significantly dependent on
215 charge air temperature, as the ignition delay increased only 0.5CA° when the temperature decreased from 81°C to 41°C.
216 Further increase of the E85 rate increased the ignition delay to 3.5 CA°. Though the effect of the charge air temperature
217 on the ignition delay was relatively small, it affected significantly the HR slope after the ignition due to the effect on the
218 ignition and combustion of E85.

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

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

235 **3.2 Emissions**

236 The THC and CO emissions at the tested cases are presented in figure 7. At both loads the THC emission were close to
237 zero in the diesel reference cases. At the high load case the decrease of the charge air temperature increased the THC
238 emissions slightly and the trend was linear. In the lower load case the effect of E85 addition was higher than at the high
239 load case. Most likely this was due to the higher E85 rate at this lower load. At the lower load case the charge air
240 temperature had a strong effect on THC, as the THC emission increased clearly when the charge air temperature was
241 lowered. As the E85 rate was increased at the lowest charge air temperature at both loads, THC emissions increased
242 indicating a high rate of unburned fuel.

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

251 The NO_x emissions (Figure 8) had an opposite trend when compared to the THC and CO emissions. In the low load
252 case the E85 addition at high charge air temperature decreased NO_x emission, while in the high load case NO_x emission
253 increased slightly, where the behaviour 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.

3.4 Efficiency

Figure 9 presents both brake 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 1500rpm 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 6.8bar 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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350

Table 1

Displaced volume	7.4 Litres
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

Measurement	In-accuracy
Load	±0.1%
Speed	±0.1%
Temperature	±1.0°C
CO	±3ppm
THC	<30ppm ⁽¹⁾
NO _x	<40ppm
Smoke	±0.005FSN
Pressures	±0.1%
Fuel mass flows	±0.12%

⁽¹⁾ for non oxygen containing species

Table 3

		Diesel EN590	E85
Density	kg/m ³	840	785
Cetane number	-	53	~11 ⁽²⁾ (ethanol) 20-25 ⁽²⁾ (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 ⁽³⁾ (ethanol boiling point) 190 ⁽³⁾ (gasoline)
Lower heating value	MJ/kg	43.2	28.7
Heat of evaporation	kJ/kg	260 ⁽²⁾	780 ⁽²⁾
Autoignition temperature	°C	230 ⁽²⁾	420 ⁽²⁾ (ethanol) 300 ⁽²⁾ (gasoline)

Values are based on fuel analysis, except ⁽²⁾ [20] and ⁽³⁾ [5]

Table 4

<i>n</i>	Load	BMEP	Power
rpm	Nm	bar	kW
1500	400	6.8	63
1500	700	11.9	110

Table 5

<i>n</i>	BMEP	E85 rate (Energy)	Charge temp.	Charge pressure	Exhaust gas temp.	Exhaust gas press.	Pre. Inj. SOI	Main inj. SOI
rpm	bar	%	°C	bar	°C	bar	CA° BTDC	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

Figure 1

Figure 2
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