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# Splash Blended Ethanol in a Spark Ignition Engine – Effect of RON, Octane Sensitivity and Charge Cooling

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**Abstract:** Downsized spark ignition engines have the benefit of high thermal efficiency; however, severe engine knock is a challenge. Ethanol, a renewable gasoline alternative, has a much higher octane rating and heat of vaporization than conventional gasoline, therefore, ethanol fuels are one of the options to prevent knock in downsized engines. However, the performance of ethanol blends in modern downsized engines, and the contributions of the research octane number (RON), octane sensitivity (defined as RON-MON) and charge cooling to suppressing engine knock are not fully understood. In this study, eight fuels were designed and tested, including four splash blended ethanol fuels (10 vol.%, 20 vol.%, 30 vol.% and 85 vol.% ethanol, referred to as E10, E20, E30 and E85), one match blended fuel (E0-MB) with no ethanol content but the same octane rating as E30, and three fuels (F1-F3) with different combinations of RON and octane sensitivity. The experiments were conducted in a single-cylinder direct-injection spark ignition (DISI) research engine. Load and spark timing sweep tests at 1800 rpm were carried out for E10-E85 to assess the combustion performance of these ethanol blends. In order to investigate the impact of charge cooling on combustion characteristics, the results of the load sweep for E0-MB were compared to those of E30. Load sweep tests were also carried out for F1-F3 to understand the impacts of RON and octane sensitivity on suppressing engine knock. The results showed that at the knock-limited engine loads, splash blended ethanol fuels with a higher ethanol percentage enabled higher engine thermal efficiency through allowing more advanced combustion phasing and less fuel enrichment for limiting the exhaust gas temperature under the upper limit of 850 °C, which was due to the synergic effects of higher RON and octane sensitivity, as well as better charge cooling. In comparison with octane sensitivity, RON was a more significant factor in

31 improving engine thermal efficiency. Charge cooling reduced engine knock tendency through lowering the  
32 unburned gas temperature.

33  
34 **Keywords:** Ethanol; Direct Injection; Knocking; Charge Cooling; Octane Sensitivity

35

## 36 **1. INTRODUCTION**

37 The transportation sector is facing pressures of increased light duty mobility demand and more stringent  
38 regulations on greenhouse gas emissions [1]. Even though hybrid and electric vehicles are gaining  
39 significant popularity, conventional vehicles powered by internal combustion engines will still be the main  
40 power source for light duty transportation. Therefore, all CO<sub>2</sub> reduction techniques, including improving  
41 the efficiency of internal combustion engines, are highly relevant in the coming years.

42 A downsized gasoline engine is one of the proven technologies that improves engine thermal efficiency  
43 and thus reduces automotive fleet CO<sub>2</sub> emissions, by as much as 25% [2]. Downsized engines equipped  
44 with turbo- or super-chargers operate at higher engine loads to deliver the same power outputs as larger  
45 engines, thus, downsized engines lead to lower pumping losses and higher efficiency at part load operating  
46 conditions.

47 Ba ãa et al. [2] performed experiments on a 1.4 L downsized turbocharged engine, of which the combustion  
48 system, exhaust system and turbocharger were optimized. The 1.4 L downsized turbocharged engine had  
49 the same peak torque and power output as a 2.4 L NA-engine, but it produced a higher brake thermal  
50 efficiency. The comparative vehicle tests which were conducted using the FTP 75-cycle with pure ethanol  
51 fuel led to an 18% overall fuel consumption improvement. Judez and Sjöberg [3] investigated the  
52 downsizing possibilities of the range extender (RE) of a vehicle by making use of predictive information  
53 of the user's throttle inputs and by using a blended discharging strategy. They found that for the realistic  
54 studied example, the RE can be downsized by 30% without any performance degradation. In downsized  
55 engines, there is a trade-off between the CO<sub>2</sub> reduction and vehicle drive-ability. Bassett et al. [4] solved

56 the drive-ability issue by adding a 48 volt eSupercharger to a downsized 1.2 L 3-cylinder MAHLE engine.  
57 In comparison to the original downsized MAHLE engine, the new downsized engine had a faster transient  
58 response and better drive-ability characteristics, clearly demonstrating eSupercharging as a key technology  
59 for enabling further engine downsizing.

60 However, despite the proven advantages of downsized engines, engine knock, caused by the auto-ignition  
61 of the end gas, is one of the main challenges that stop downsized engines from achieving their full potential  
62 [5]. High octane rating fuels are one of the key solutions for suppressing engine knock [5, 6]. Ethanol, a  
63 widely used renewable gasoline alternative, has a much higher octane rating than conventional gasoline  
64 fuel. Splash blending ethanol into gasoline improves the octane rating of the resulting fuel mixture [7-10].  
65 For example, Stein et al. [9] found that adding 10 vol.% and 20 vol.% ethanol into a RON 82 base gasoline  
66 led to 7 and 13 unit improvements of RON, respectively. The octane boost effect produced by ethanol  
67 addition is more significant for base gasoline with a lower octane rating. Additionally, ethanol has a much  
68 higher heat of vaporization (HOV) than gasoline, which offers an additional benefit of improved charge  
69 cooling when it is used in direct injection (DI) engines. Based on a study of the compression ratio (CR)  
70 distribution of engines models sold in the North American market in 2013 [11], it was found that DI engines  
71 have approximately 1 unit higher CR than PFI engines. The increase of CR in DI engines is mainly due to  
72 the cooling effect.

73 There are many researchers who have studied splash blended ethanol fuels in spark ignition engines and  
74 achieved promising results. For example, Jung et al. [12] studied E10, E20, and E30 fuels in a Ford 3.5 L  
75 EcoBoost DI turbocharged engine with compression ratios (CR) of 10.0:1 and 11.9:1. It was found that a  
76 10 vol.% increase of ethanol in the blends enabled a 2 unit increase of CR without changing the knock  
77 limited combustion phase. The higher ethanol content required less fuel enrichment at high engine speeds  
78 and loads. In comparison with E10 at CR 10:1, E30 at CR 11.9:1 achieved a 7.5% CO<sub>2</sub> emission reduction  
79 when the engine was operated on the US06 Highway cycle, whilst volumetric fuel economy was  
80 approximately the same. Schwaderlapp et al. [13] investigated gasoline, match blended E20, and splash

81 blended E20 in a boosted DISI engine under full load conditions. The match blended E20 with the same  
82 RON as gasoline did not allow an increase in CR, whereas the splash blended E20 enabled a 2.2 unit  
83 increase of CR. At full engine load, due to the higher CR and the reduced fuel enrichment, the engine  
84 thermal efficiency achieved when using splash blended E20 was improved by up to 39% compared with  
85 that achieved when gasoline was used. The potential for CO<sub>2</sub> reduction by using ethanol blends was  
86 investigated using the New European Driving Cycle (NEDC) cycle over various vehicle classes ranging  
87 from mid-sized passenger cars to sport utility vehicles [13]. When the optimised CR was applied to the  
88 engine, CO<sub>2</sub> emission reductions were in the range of 3.9-4.9% for E20 in comparison with gasoline,  
89 depending on the vehicle type.

90 Apart from its high octane rating and high charge cooling effect, ethanol has a high octane sensitivity, which  
91 may play an important role in suppressing engine knock. The octane sensitivity is defined as the difference  
92 between the research octane number (RON) and motor octane number (MON), both of which are measured  
93 in cooperative fuel research (CFR) engines designed 90 years ago [14-16]. However, modern spark ignition  
94 engines, especially turbo-charged downsized engines, tend to operate at relatively lower temperatures than  
95 CFR engines, if the comparison is made with the same intake manifold pressure. This is because of the use  
96 of advanced technologies such as charge intercoolers, cooled exhaust gas recirculation (EGR) and DI [17].  
97 The RON test may partially capture ethanol's charge cooling effect, which is absent in the MON test [18].  
98 To compensate for the disconnection between the CFR engine and modern engines, an octane index (OI)  
99 was proposed as:  $OI = RON + K * (RON - MON)$  [19]. K is a scaling factor depending solely on the in-cylinder  
100 thermal and pressure history experienced by the end-gas prior to the onset of auto-ignition. The literature  
101 shows that, for some engine types at some operating conditions, a fuel with a high octane sensitivity is  
102 beneficial to reduce engine knock tendency [19-23]. For example, Remmert et al. [23] tested seven RON  
103 and MON de-correlated fuels in a prototype "Ultraboost" engine under high boost conditions, and they  
104 found that the K value tended to be negative at boosted conditions; therefore, a high octane sensitivity fuel  
105 was beneficial. Kalghatgi [24] studied 37 spark ignition engines ranging from naturally aspirated to turbo-

106 charged, and 1.2 to 2.4 L. It was found that under high load conditions, some engines experienced less  
107 knocking when high octane sensitivity fuels were used.

108 Currently, ethanol is largely used in low percentage blend forms such as E5 or E10. Higher octane splash  
109 blended ethanol fuels beyond E10 are expected to give better performance in downsized engines, however,  
110 their performance in modern downsized DISI engines, and the contributions of RON, octane sensitivity and  
111 charge cooling to combustion are not fully understood. In this study, eight fuels were designed and tested,  
112 including four splash blended ethanol (10 vol.%, 20 vol.%, 30 vol.% and 85 vol.% ethanol, noted as E10,  
113 E20, E30 and E85), one match blended fuel (E0-MB) with zero ethanol content but the same octane rating  
114 with those of E30, and three fuels (F1-F3) with different combinations of RON and octane sensitivity. The  
115 experiments were conducted in a single-cylinder DISI research engine. Load and spark timing sweep tests  
116 with an engine speed of 1800 rpm, and full load tests were carried out for E10-E85 to assess the combustion  
117 performance of ethanol blends. In order to investigate the effect of charge cooling, the load sweep was  
118 conducted for E0-MB, and the results were compared to those of E30. Load sweep tests were also carried  
119 out for F1-F3, to understand the impacts of RON and octane sensitivity on engine combustion.

120

## 121 **2. EXPERIMENTAL SYSTEMS AND METHODS**

### 122 **2.1. ENGINE AND INSTRUMENTATION**

123 Experiments were conducted in an AVL single-cylinder 4-stroke DISI research engine with 82 mm bore  
124 and 86 mm stroke, the setup of which is presented in Figure 1. Its combustion system features a 4-valve  
125 pent roof cylinder head equipped with variable valve timing (VVT) systems for both intake and exhaust  
126 valves. The cylinder head is equipped with a central-mounted outward opening piezo direct injector. The  
127 spark plug is located at the centre of the combustion chamber slightly tilting towards the exhaust side.  
128 The engine is coupled to an electric dynamometer, which is able to control the engine at a constant speed  
129 ( $\pm 1$  rpm) regardless of the engine power output. The engine is controlled via an IAV FI2RE management  
130 system. An AVL Indicom system is used for real time combustion indication and analysis. A Siemens CATs

131 system is used for signal acquisition and recording, and it communicates with the IAV FI2RE and the AVL  
132 Indicom systems. The Siemens CATs system is also used for controlling air, fuel, coolant and oil  
133 conditioning units, and the emission measurement equipment.

134 A Kistler pressure transducer is used for the in-cylinder pressure measurement, and it is installed in a sleeve  
135 between the intake and exhaust valves. The in-cylinder pressure is collected via a charge amplifier (ETAS  
136 ES630.1) with a resolution of 0.1 crank angles ( $^{\circ}\text{CA}$ ) between  $-30^{\circ}\text{CA}$  and  $70^{\circ}\text{CA}$  after top dead centre  
137 (ATDC), and a resolution of  $1^{\circ}\text{CA}$  at other crank angles. Some key temperature and pressure measurement  
138 locations labelled as 'T' and 'P' in Figure 1.

139 The engine intake system is connected to an external air handling device, capable of delivering up to 3 bar  
140 of boosted air. Air is first filtered and dried, before it is delivered to a conditioning unit. The capacity of  
141 this air conditioning unit is approximately 200 L, in which air pressure and temperature are precisely  
142 controlled using a closed-loop control system. Temperatures of fuel, coolant and oil are controlled by  
143 individual AVL conditioning systems. Fuel consumption is measured by an AVL fuel mass flow meter.

144

## 145 **2.2. FUEL PROPERTIES**

146 Table 1 lists the properties of the fuels in this study. There are three groups of fuels in the fuel matrix.  
147 Group 1 includes E10-E85, which is for the study of engine performance of splash blended ethanol blends.  
148 E10 is a standard EN228 compliant gasoline fuel with a 10 vol.% ethanol content. E20, E30 and E85 were  
149 splash blended fuels produced by adding more ethanol into E10. Group 2 includes E0-MB and E30. E0-  
150 MB had no ethanol content, but it had the same RON and MON as E30. By comparing the engine  
151 performance of E0-MB and E30, it is possible to assess the charge cooling effect. Group 3 includes F1-F3.  
152 F1 and F2 have similar octane sensitivities but 5.6 units difference in RON, and F2 and F3 have similar  
153 RON but 5.5 units difference in octane sensitivity. Therefore, by comparing F1 and F2, and F2 and F3, it  
154 is possible to investigate the effect of RON and octane sensitivity on the engine combustion, respectively.

155

### 156 **2.3. EXPERIMENTAL PROCEDURE**

157 Table 2 lists the test matrix. For E10-E85, engine load and spark timing sweep, and full load performance  
158 tests were conducted for assessing the engine performance of splash blended ethanol fuels. In order to  
159 investigate the effect of charge cooling, RON and octane sensitivity, engine load sweep tests were  
160 conducted for E0-MB and F1-F3.

161 The load sweep was carried out by sweeping the intake manifold pressure from 0.65 to 2 bar at a constant  
162 engine speed of 1800 rpm. The spark timing sweep was conducted by sweeping the spark timing from  
163 KLSA-2 to KLSA+6 at a constant engine speed of 1800 rpm and a constant 1.6 bar intake manifold pressure.  
164 KLSA stands for knock limited spark advance. The engine full load was defined by IMEPs of 15 bar at  
165 1000 rpm, 20 bar at 1800 rpm, 22 bar at 2500 rpm, 21 bar at 3500 rpm, and 20 bar at 3500 rpm.

166 For each fuel at a certain engine operating condition, if the engine was not knock-limited, spark timing was  
167 adjusted by aiming the combustion centre (MFB50) at  $7.5 \pm 0.5$  °aTDC, which was an approximation of the  
168 maximum brake torque (MBT) spark timing. The term ‘MFB50’ stands for the crank angle position where  
169 50% mass fraction of the fuel has been burned. For the remainder of this paper, ‘MFB50’ and ‘combustion  
170 centre’ are used interchangeably.

171 When engine knock occurred, the spark timing was retarded to limit the knock intensity under the maximum  
172 tolerated intensity in order to avoid potential engine damage. In this case, spark timing is referred to as the  
173 KLSA. The same intake and exhaust valve timing, and the same injection timing maps were used for all  
174 fuels; more detailed information can be found in Appendix Table A1 and Table A2.

175 Table 3 lists some key engine boundary conditions. The knock intensity was defined as the maximum  
176 amplitude of in-cylinder pressure oscillation, which was obtained by filtering and rectifying the raw in-  
177 cylinder pressure data using a band-pass filter (3-30 kHz). Since the knock intensity changes significantly  
178 cycle-to-cycle, especially when engine knock occurs, in this study the mean peak knock intensity (MPKI)  
179 over 50 cycles was used as a practical indicator for knocking assessment. KLSA was determined using the  
180 MPKI listed in Table 3.



181 Appendix Table A3 provides some brief summary of the measurement uncertainties of key instrument.

182

### 183 **3. RESULTS AND DISCUSSION**

184 The results and discussion has been split into two sections. In the first section, the effect of splashed blended  
185 ethanol fuels on the engine performance is presented. The benefits of using splash blended ethanol fuels in  
186 GDI engines are related to the high charge cooling effect, high RON and high octane sensitivity of ethanol,  
187 therefore, in the second section, the effect of charge cooling, and the effect of RON and octane sensitivity,  
188 are presented in order to understand their individual contribution to the engine combustion.

189

#### 190 **3.1. SPLASH BLENDED ETHANOL FUELS**

191 Figure 2 shows the results of the engine load sweep for splash blended ethanol fuels (E10-E85) at the engine  
192 speed of 1800 rpm. Seven main combustion indicators, including the engine indicated thermal efficiency,  
193 ignition timing, combustion centre, MFB5-50, coefficient of variation (COV) of IMEP, peak in-cylinder  
194 pressure, and exhaust gas temperature, were selected to illustrate the combustion characteristics of various  
195 ethanol blends. Indicated specific fuel consumption results are also presented in Figure 2.

196 The spark timing was adjusted by aiming the combustion centre at  $7.5 \pm 0.5$  °TDC if the engine was not  
197 knock-limited. At low load ( $< 8$  bar IMEP), the spark timings for all fuels were similar, whilst differences  
198 became clearer when loads higher than 12 bar IMEP were used, with higher percentage ethanol blends  
199 allowing more advanced spark timings. The onset knock-limited IMEP is approximately 8 bar for E10 and  
200 E20, 12 bar for E30, and 16 bar for E85. At 6 bar IMEP, all fuels regardless of the ethanol content had  
201 limited differences in all combustion indicators, largely because the engine was not limited by knocking,  
202 and thus the combustion phasing was optimized for each fuel. The engine thermal efficiency became  
203 differentiated as the engine was operated at knock-limited load: higher percentage ethanol blends achieved  
204 better engine indicated thermal efficiency (defined by the ratio of indicated work produced in a complete  
205 cycle and fuel energy per cycle) compared to E10. For example, in comparison with E10, E85 achieved an

206 approximate improvement of up to 12% in the engine indicated thermal efficiency at the IMEP ranging  
207 from 15 to 20 bar.

208 The early combustion duration, defined as the duration between 5-50% mass fraction burned (MFB5-50),  
209 is used to quantify the burn duration. It can be seen that higher percentage ethanol blends had shorter MFB5-  
210 50, especially at knock-limited load. MFB5-50 is presented because MFB50 was used in this study to locate  
211 the combustion centre. The MFB50 is a more reliable point to extract from the in-cylinder pressure data  
212 than MFB90 or MFB95, which were in a generally flat area of the MFB curve and as such are more  
213 susceptible to noise and cycle to cycle variation [25]. As the engine was operated at knock-limited load,  
214 faster combustion (shorter combustion duration) led to more combustion energy being transferred into  
215 effective work on the piston. The reason for the shorter combustion durations for higher percentage ethanol  
216 blends is because of 1) more advanced spark timing, and 2) faster laminar flame speed of ethanol compared  
217 to that of gasoline. The relevance of this increase in laminar flame speed to combustion in an internal  
218 combustion engine is also related to other factors, such as the mixture turbulence and the influence of the  
219 gas temperature [25].

220 The COV of IMEP shows the cyclic variability in indicated work per cycle. The calculation of COV of  
221 IMEP is:  $COV(IMEP) = \sigma(IMEP) / \mu(IMEP)$ , where  $\sigma(IMEP)$  is the standard deviation of IMEP, and  $\mu$   
222 (IMEP) is the averaged IMEP in the measured cycles. Higher ethanol blends contributed to improved  
223 combustion stability, indicated by the lower COV of IMEP, especially at high load. An advanced  
224 combustion phase and shorter combustion duration both result in higher peak in-cylinder pressure. As more  
225 chemical energy released by fuel combustion was converted to effective work on the piston, the exhaust  
226 gas temperature decreased with ethanol content, especially at high load, contributing to improved engine  
227 thermal efficiency.

228 错误!未找到引用源。 Compared with E10, E85 led to approximately a 40% higher mass-based indicated  
229 specific fuel consumption (ISFC) at knock-free load due to its low calorific value, and the difference was  
230 reduced to 26% at the highest load, resulting from improved indicated thermal efficiency. Similarly, E20

231 and E30 had higher ISFC than E10 at knock-free load. As the engine load increased, the difference started  
232 to reduce or even become completely offset. Because of the higher density of ethanol than gasoline, the  
233 differences between E10 and other higher ethanol blends in volume-based ISFC could be less than these  
234 observed when considering mass-based ISFC.

235 Figure 3 shows the IMEP of splash blended ethanol fuels at various engine intake manifold pressures. It  
236 was observed that higher percentage ethanol blends achieved higher engine loads. Because the engine load  
237 sweep was conducted by sweeping the intake manifold pressure, it is possible to obtain IMEP data at various  
238 intake manifold pressures by interpolating the relevant data. Therefore, the IMEP results presented in Figure  
239 3 are directly linked to the results of the combustion characteristics presented in Figure 2. It was found that  
240 higher ethanol blends achieved higher engine loads. E85 achieved 0.3 bar (3%) and 2.5 bar (14%) higher  
241 IMEP compared to E10 at 1 bar and 2 bar intake manifold pressures, respectively. This is due to more  
242 advanced spark timings, shorter combustion durations, and less exhaust energy losses of the E85  
243 combustion compared to E10. At intake manifold pressures higher than 1.6 bar, the increase of the engine  
244 power output for E10-E30 was almost linear with ethanol content, however, as ethanol content was  
245 increased further to 85 vol.% (E85), the rate of increase in engine power output was reduced. This can be  
246 explained by the octane increase rate with various ethanol additions. The RON of E10, E20, E30 and E85  
247 are 96.5, 99, 101.4 and 107.2, respectively. Therefore, the octane increase rate for E10-E30 is approximately  
248 2.5 units of octane for every 10 vol.% ethanol, however, this rate was reduced to 1.1 units of octane per 10  
249 vol.% ethanol when the ethanol content was increased from 30 vol.% to 85 vol.%. The non-linear increase  
250 of octane rating with ethanol content is a result of the synergistic effect of ethanol with alkanes in  
251 suppressing low temperature heat release. It may also be due to the RON measurement method in which  
252 the charge cooling of ethanol affects the rating [18].

253 Figure 4 presents the results of the spark timing sweep for splash blended ethanol fuels at 1800 rpm and  
254 1.6 bar intake manifold pressure. At this condition, the IMEP was approximately 16 bar; the actual precise  
255 value depended on the spark timing. At this intake manifold pressure, the engine was knock limited for all

256 fuels. The KLSA was 1.4 °CA for E10, -2 °CA for E20, -5.7 °CA for E30, and -11.0 °CA for E85. For each  
257 fuel, the spark timing was swept in the range of KLSA-2 to KLSA+6. In the  $x$  axis of Figure 4, spark retard  
258 (spark-KLSA) represents the number of crank angle degrees that the spark timing is retarded from the  
259 KLSA of each fuel. A positive spark retard means that the spark timing is delayed from KLSA, and a  
260 negative spark retard means that spark timing is advanced from the KLSA.

261 The indicated thermal efficiency and IMEP shown in Figure 4 are normalized from those at the KLSA of  
262 each fuel. The normalization was done for each fuel by dividing the indicated thermal efficiency or IMEP  
263 at one spark timing by that at the KLSA. The normalization of these two parameters enabled a direct  
264 comparison of their responses/sensitivities to spark timing. It is clear that the sensitivities of indicated  
265 efficiency and IMEP to spark retard were fuel dependent. E10 was more sensitive to spark retard than other  
266 higher percentage ethanol blends. For a 2% reduction of IMEP, E10, E20, E30 and E85 allowed 1.5, 2.1, 3  
267 and 5 °CA spark retards, respectively. The combustion centre retard shown in Figure 4 was linear to spark  
268 retard for all fuels. The rate of combustion centre retard was fuel dependent, which were 1.8, 1.6, 1.4 and  
269 1.2 °CA per degree of spark retard for E10, E20, E30 and E85, respectively. The higher rate of combustion  
270 centre retard matched with the higher reduction rate of engine indicated thermal efficiency.

271 The mean peak knock intensity shown in Figure 4 indicated that for E10-E30, spark retards reduced the  
272 knock intensity, and spark timing advances from KLSA significantly increased the knock intensity,  
273 especially for E10 and E20. For E85, a low knock intensity was maintained and it was less sensitive to  
274 spark retard, showing that the CR of the engine fuelled with E85 can be further increased from 11:5:1 to  
275 improve engine thermal efficiency.

276 Figure 5 shows the full load results for splash blended ethanol fuels. The full load power outputs for all  
277 fuels were kept the same, as indicated by the IMEP data. The indicated thermal efficiency orders for all  
278 fuels were: E85>E30>E20>E10. Compared to E10, E20 led to 2.8-7% higher indicated thermal efficiency  
279 at full load, depending on the engine speed, whilst the improvement for E85 was in the range of 8.3-27%.

280

281 High percentage ethanol blends led to higher engine thermal efficiency due to the more advanced phase of  
282 the combustion centre, less fuel enrichment requirement and lower exhaust gas temperature. The exhaust  
283 temperature increased with engine speed due to less heat transfer. Advancing the spark timing reduces the  
284 exhaust gas temperature because the end of combustion is advanced, and more heat energy is converted  
285 into effective work on the engine piston. This explains why between 1000 and 2500 rpm engine speed, high  
286 percentage ethanol blends had lower exhaust gas temperatures. However, advancing the spark timing was  
287 restricted by engine knock. When the exhaust gas temperature exceeded the upper limit of 850 °C, fuel  
288 enrichment was used. For E85, no fuel enrichment was required at any tested engine speed, whilst E10  
289 needed fuel enrichment from 2500 rpm engine speed.

### 291 **3.2. EFFECTS OF RON, OCTANE SENSITIVITY AND CHARGE COOLING**

292 Figure 6 shows the results of the effects of RON and octane sensitivity on engine combustion. It is  
293 noteworthy that F1-F3 all contained 10 vol.% of ethanol, and their heats of vaporization were similar (see  
294 Table 1), therefore, the charge cooling effects of F1-F3 were similar. F2 had almost the same octane  
295 sensitivity as F1, but 5.6 units higher RON, therefore, by comparing the results from F1 and F2, it is possible  
296 to understand the effect of the 5.6 units difference of RON on engine combustion. From Figure 6, it is clear  
297 that at knock-limited engine load, F2 resulted in higher engine thermal efficiency, a more advanced  
298 combustion phasing, shorter combustion duration, higher in-cylinder pressure, and lower exhaust  
299 temperature. The maximum knock-free IMEP for F2 was 9.5 bar, which was approximately 3 bar higher  
300 than that of F1. Due to engine knock and pre-ignition, F1 was only tested up to 1.7 bar intake manifold  
301 pressure, whilst F2 was tested up to 2 bar intake manifold pressure. The engine knock intensity was  
302 monitored in real-time using the AVL Indicom Combustion Analyser. The knock intensity is defined as the  
303 maximum amplitude of in-cylinder pressure oscillation, which was obtained by filtering and rectifying the  
304 raw in-cylinder pressure data using a band-pass filter (3-30 kHz). The definition of a pre-ignition is when  
305 auto-ignition of the fuel/air mixture happens before the spark timing, resulting in significant engine knock

306 and very high in-cylinder pressures. The pre-ignition can be observed from the pressure trace displayed in  
307 the AVL Indicom Combustion Analyser.

308

309 The maximum IMEP for F2 was 3.5 bar higher than that for F1, due to the higher intake manifold pressure  
310 and more advanced combustion centre. F2 also had a lower COV of IMEP at high engine load, resulting  
311 from the less retarded combustion phasing.

312 F3 had almost the same RON as F2, but 5.5 units higher octane sensitivity, therefore, by comparing results  
313 from F2 and F3, it is possible to understand the effect of 5.5 units of octane sensitivity on engine  
314 combustion. It can be seen from Figure 6 that high octane sensitivity led to improved combustion, however,  
315 its impact was much less significant than RON. F3 did not allow a higher knock-free IMEP than F2, whilst  
316 F2 led to a 3 bar higher knock-free IMEP. The maximum IMEP difference between F2 and F3 was 1.5 bar;  
317 considerably less than the 3.5 bar difference between F1 and F2. Similar evidence can also be found in the  
318 COV of IMEP, peak in-cylinder pressure and exhaust gas temperature. In addition, from Table 1 it can be  
319 seen that the increase in octane sensitivity by splash blending ethanol into E10 is less than the increase in  
320 RON. Therefore, it can be expected that, RON would contribute more to the anti-knock quality than the  
321 octane sensitivity for E10-E85.

322 In order to study the effect of charge cooling on engine combustion, E0-MB with no ethanol content but  
323 the same RON and MON as those of E30 was designed and tested. In DISI engines, apart from the octane  
324 rating of fuels, the charge cooling effect is an important contributor to suppressing engine knock. The  
325 charge cooling effect is related to the heat of vaporization; the fuel spray/droplet vaporizes after a direct  
326 injection event by absorbing heat from the compressed air within the cylinder, which reduces the in-cylinder  
327 charge temperature in proportion to the heat of vaporization of the fuel. As a result, compared to port fuel  
328 injection (PFI) engines where fuel spray is vaporized partially by absorbing heat from hot intake valves, DI  
329 engines are usually more knock resistant. Leone et al. [7] suggested that on average, DI engines had 1 unit  
330 higher CR than those of PFI engines.

331 The heat of vaporization of E30 and E0-MB are 551 and 365 kJ/kg, respectively. Due to the existence of  
332 30 vol.% ethanol in E30, the lower calorific value of E30 (38.42 MJ/kg) was 8.7% lower than that of E0-  
333 MB (42.05 MJ/kg), therefore, a higher quantity of E30 was needed for the same amount of energy input  
334 than E0-MB. The collective effects of higher heat of vaporization and reduced lower calorific value resulted  
335 in E30 requiring approximately 65% more heat for vaporization at the same engine load than E0-MB.

336 Figure 7 shows the effect of charge cooling by comparing results from E0-MB and E30. From the indicated  
337 thermal efficiency results, it is clear that E30 was preferred at high load (>15 bar IMEP) where the engine  
338 was knock-limited. The more advanced spark timing and combustion centre, shorter combustion duration,  
339 and higher in-cylinder temperature provides strong evidence that charge cooling contributed to suppressing  
340 engine knock, even though the ethanol content was as low as 30 vol.%. In addition, E30 showed higher  
341 combustion stability, as indicated by a lower COV of IMEP. The higher engine thermal efficiency of E30  
342 was also reflected in the lower exhaust gas temperature compared to that of E0-MB. The maximum IMEP  
343 of E30 was approximately 1.1 bar higher than that of E0-MB, resulting from the charging cooling effect.  
344 Apart from the cooling effect, the faster burning rate of ethanol is also the reason for the better combustion  
345 phasing of E30 in comparison with E0 [25].

346 Figure 8 shows the in-cylinder pressure and unburned zone temperature of E0-MB and E30 at 1800 rpm  
347 engine speed and 2 bar intake manifold pressure. For E30, the in-cylinder pressure rise due to combustion  
348 was more advanced than that for E0-MB, resulting from the more advanced spark timing. The peak pressure  
349 of E30 was approximately 10 bar higher than that of E0-MB. The unburned zone temperature was calculated  
350 by the AVL Concerto software. It showed that due to charge cooling, the unburned gas temperature at top  
351 dead centre (TDC) was approximately 50 K lower for E30 than that for E0-MB. The cooler unburned gas  
352 led to a longer ignition delay, therefore, E30 allowed for a 1.8 °CA more advanced spark timing at this  
353 engine operating condition.

354

#### 355 4. CONCLUSIONS

356 In this study, eight fuels were designed and tested, including four with splash blended ethanol (E10-E85),  
357 one match blended fuel (E0-MB) with zero ethanol content but the same RON and MON as those of E30,  
358 and three fuels (F1-F3) with different combinations of RON and octane sensitivity. The experiments were  
359 conducted in a single-cylinder DISI research engine. The following are conclusions drawn from the results  
360 and discussion.

361 1. Splash blended ethanol has better anti-knock properties than base gasoline, enabling a larger knock-  
362 free engine load range and more advanced combustion phasing when the engine is knock-limited. Other  
363 combustion parameters such as combustion duration, peak pressure and exhaust temperature agreed with  
364 the finding that higher ethanol blends led to better engine indicated thermal efficiency, especially at high  
365 and full load operating conditions. Compared to E10, E20 led to 2.8-7% higher indicated thermal efficiency  
366 at the full load, depending on the engine speed, whilst the improvements for E85 were in the range of 8.3-  
367 27%.

368  
369 2. Compared to E10, at knock-limited engine load, the combustion of higher percentage ethanol blends  
370 were less sensitive to spark timing retard, resulting in less negative impacts on IMEP and indicated thermal  
371 efficiency. At 1.6 bar intake pressure, advances in spark timing from KLSA caused a more severe knock  
372 intensity rise for E10 than for other higher percentage ethanol blends.

373 3. For E30, at knock limited operating conditions, the positive effect of charging cooling was reflected  
374 in the more advanced combustion phasing, higher engine thermal efficiency, and lower unburned gas  
375 temperature at TDC. The high heat of vaporization and low stoichiometric air/fuel ratio of ethanol blends  
376 both contributed to a better charge cooling effect. In addition, the faster burning rate of ethanol also  
377 contributed to this.

378 4. High RON and high octane sensitivity both contributed to improve the fuel's anti-knock quality,  
379 with the impact of RON being more significant than that of octane sensitivity. For ethanol blends, most of  
380 the anti-knock quality improvement was from the RON improvement.



381

## 382 **Acknowledgement**

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386 in fuel blending and engine testing.

## 387 **Definitions, Acronyms and Abbreviations**

|     |       |                                     |
|-----|-------|-------------------------------------|
| 388 |       |                                     |
| 389 | AFR   | Air Fuel Ratio                      |
| 390 | ATDC  | After Top Dead Centre               |
| 391 | BTDC  | Before Top Dead Centre              |
| 392 | °CA   | Crank Angle                         |
| 393 | CAD   | Crank Angle Degree                  |
| 394 | CFR   | Cooperative Fuel Research           |
| 395 | CR    | Compression Ratio                   |
| 396 | COV   | Coefficient of Variation            |
| 397 | DI    | Direct Injection                    |
| 398 | DISI  | Direct Injection Spark Ignition     |
| 399 | EGR   | Exhaust Gas Recirculation           |
| 400 | HOV   | Heat of Vaporization                |
| 401 | KLSA  | Knock Limited Spark Advance         |
| 402 | LHV   | Lower Heating Value                 |
| 403 | IMEP  | Indicated Mean Effective Pressure   |
| 404 | ISFC  | Indicated Specific Fuel Consumption |
| 405 | MFB   | Mass Fraction Burned                |
| 406 | MPKI  | Mean Peak Knock Intensity           |
| 407 | MON   | Motor Octane Number                 |
| 408 | NEDC  | New European Driving Cycle          |
| 409 | OI    | Octane Index                        |
| 410 | PFI   | Port Fuel Injection                 |
| 411 | RPM   | Revolutions Per Minute              |
| 412 | RON   | Research Octane Number              |
| 413 | SI    | Spark Ignition                      |
| 414 | TDC   | Top Dead Centre                     |
| 415 | VOL.% | Volumetric Percentage               |
| 416 | VVT   | Variable Valve Timing               |

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420 **Figure 3:** Engine load for splash blended ethanol fuels at various intake manifold pressures

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422 intensity was below the maximum limit in all the tested spark timing, therefore, the optimised spark timing  
423 for E85 was MBT, instead of KLSA for other fuels)

424 **Figure 5:** Full load results for splash blended ethanol fuels

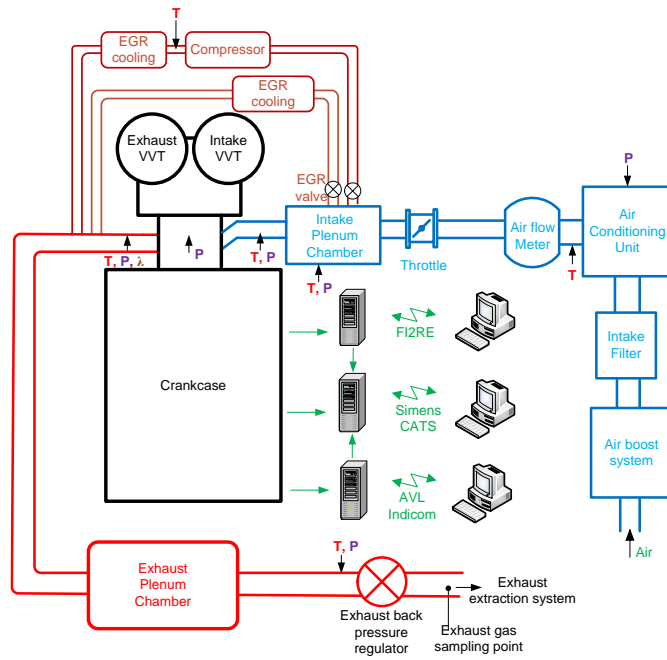
425 **Figure 6:** Results for RON and octane sensitivity effect

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428 MB and E30 at 1800 rpm engine speed and 2 bar intake manifold pressure

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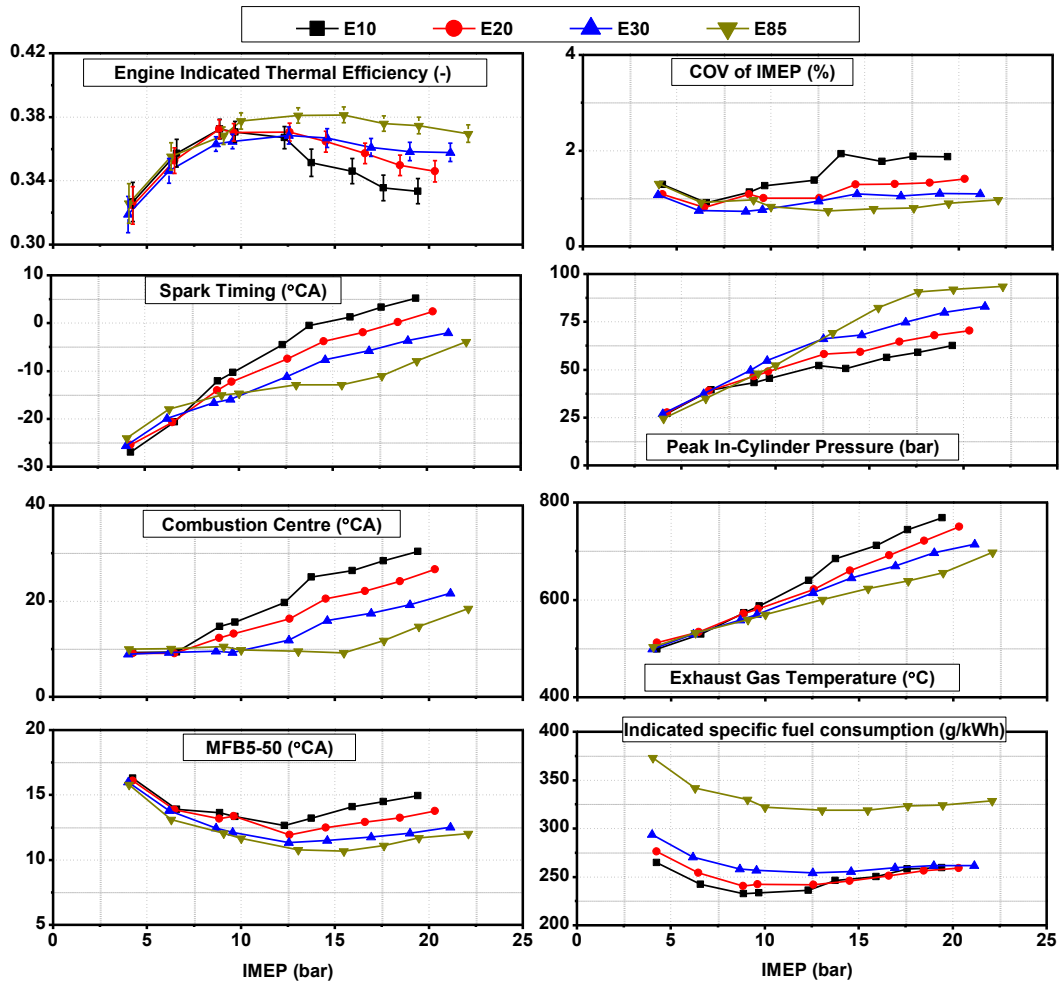
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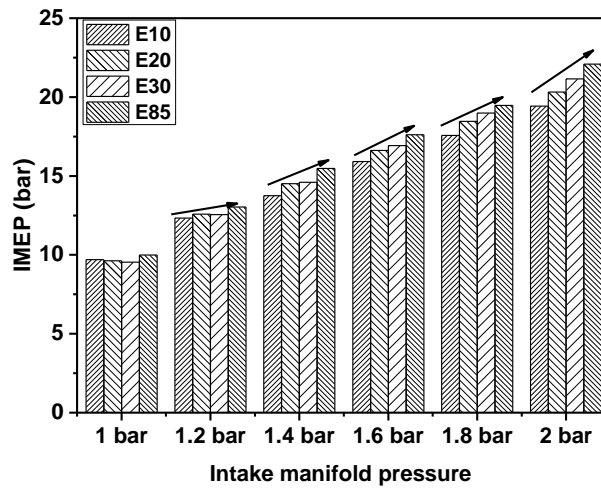
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**Figure 1:** Engine setup

**Results of splash blended ethanol fuels**  
 Engine speed:1800 rpm; Spark timing: MBT/ fixed knock intensity  
 Same valve timing and injection maps for all fuels; Lambda=1; CR=11.5:1



**Figure 2:** Results of engine load sweep for splash blended ethanol fuels



**Figure 3:** Engine load for splash blended ethanol fuels at various intake manifold pressures

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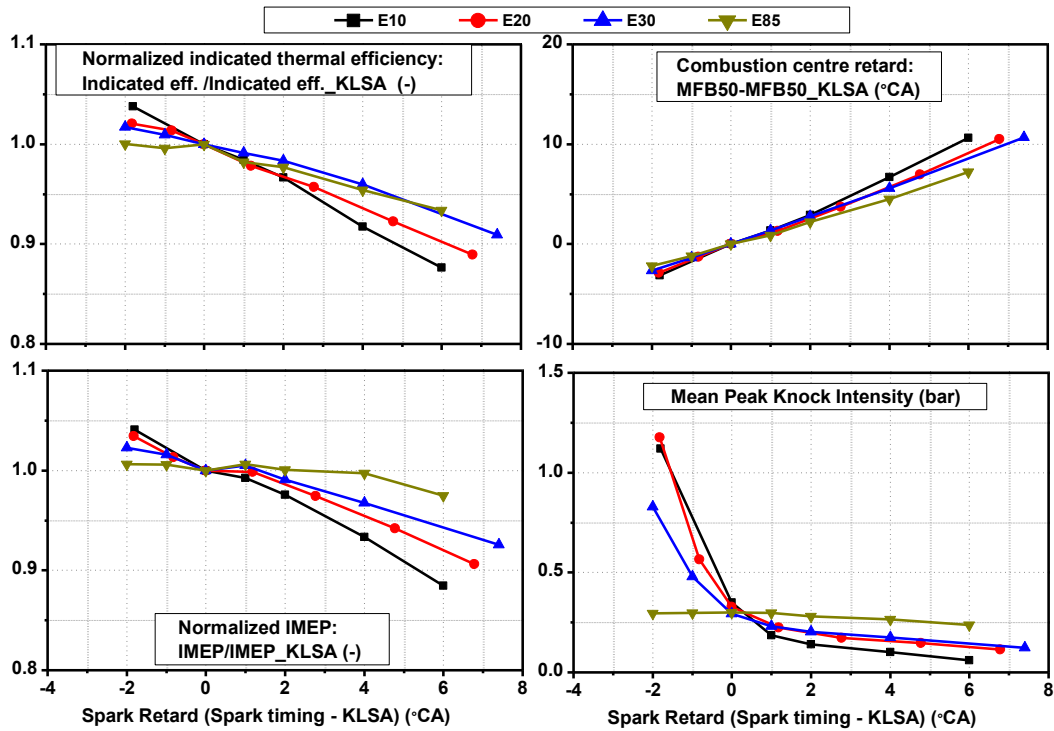
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### Results of spark timing sweep for splash blended ethanol fuels

Same valve timing and injection maps for all fuels; Lambda=1; CR=11.5:1

Engine speed:1800 rpm; Intake manifold pressure: 1,6 bar

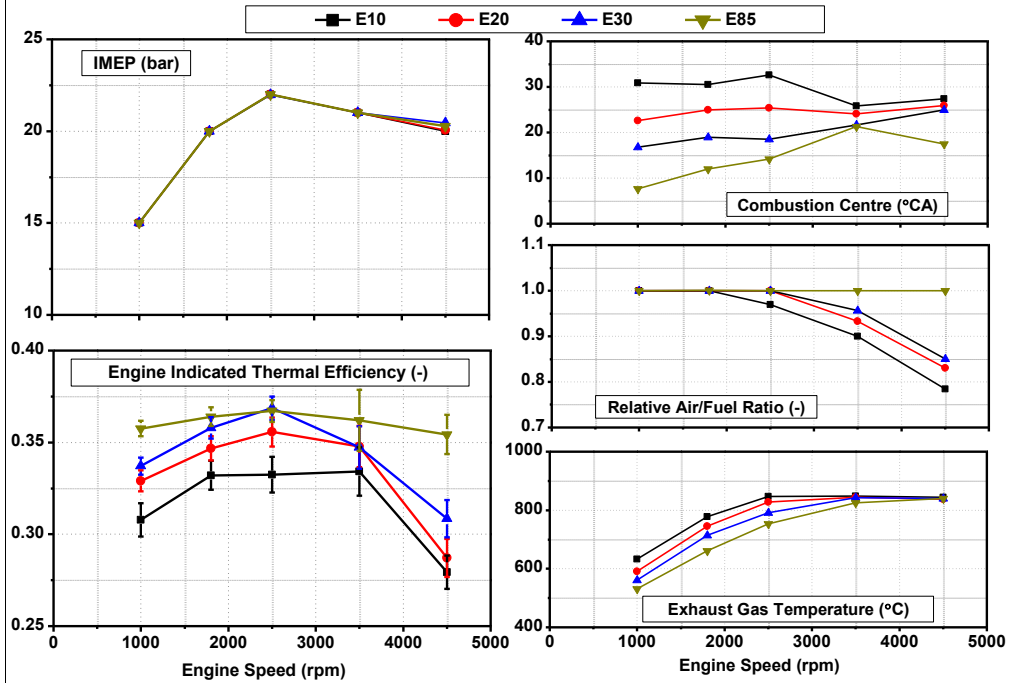
KLSA: 1.4 °CA for E10; -2 °CA for E20; -5.7 °CA for E30; -11.0 °CA for E85



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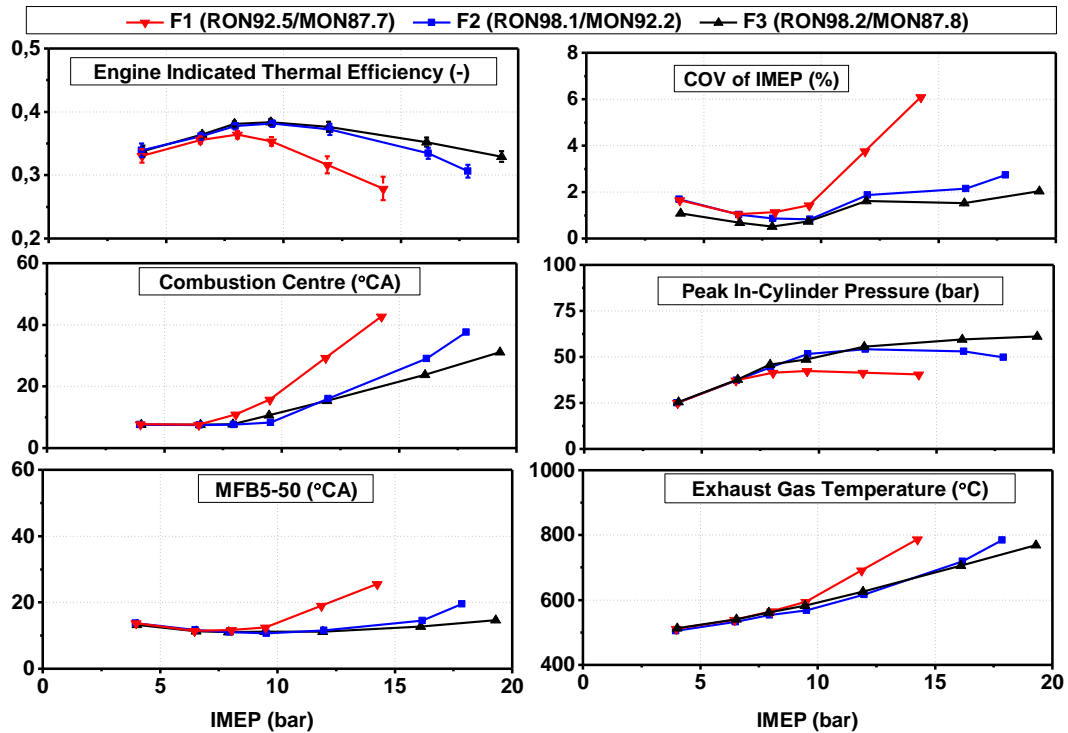
442 **Figure 4:** Results of spark timing sweep for splash blended ethanol fuels (Note: for E85, the knock  
 443 intensity was below the maximum limit in all the tested spark timing, therefore, the optimised spark  
 444 timing for E85 was MBT, instead of KLSA for other fuels)  
 445

**Results of splash blended ethanol fuels**  
 Engine speed:1000-4500 rpm; Spark timing: Fixed knock intensity  
 Same valve timing and injection timing maps for all fuels; CR=11.5:1



**Figure 5:** Full load results for splash blended ethanol fuels

**Effect of octane sensitivity**  
 Engine speed:1800 rpm; Spark timing: MBT/fixed knock intensity  
 Same valve timing and injection maps for all fuels; Lambda=1; CR=11.5:1



**Figure 6:** Results for RON and octane sensitivity effect

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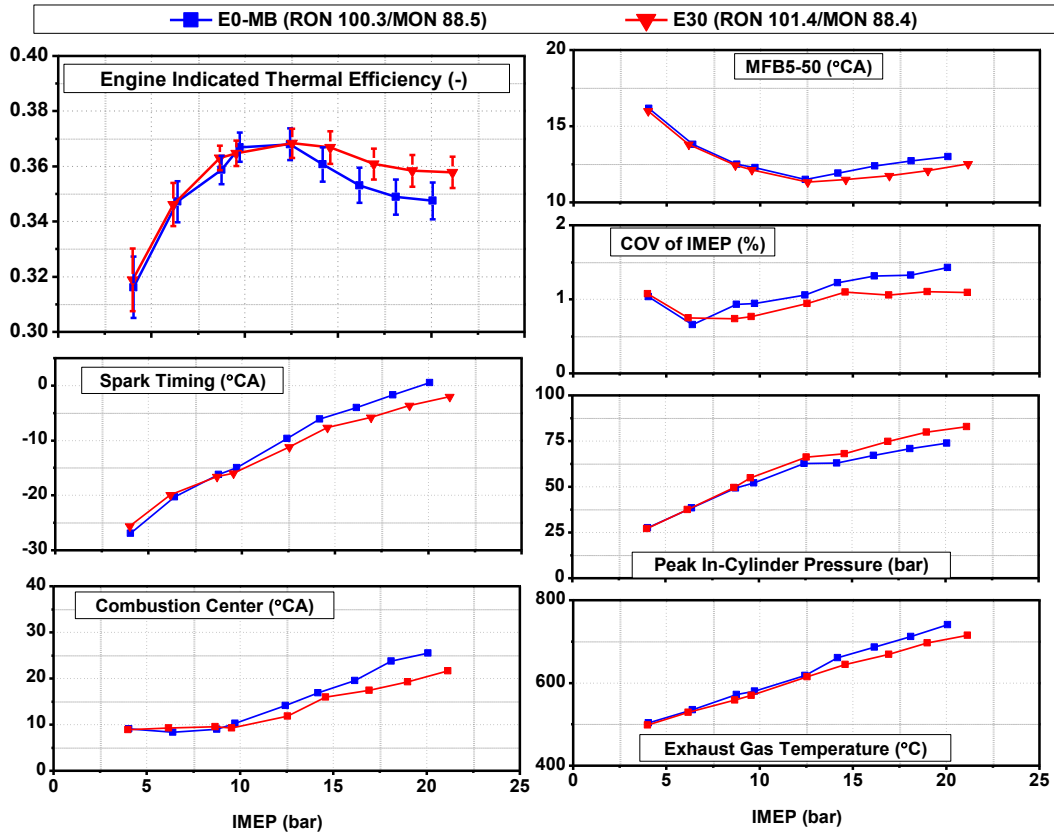
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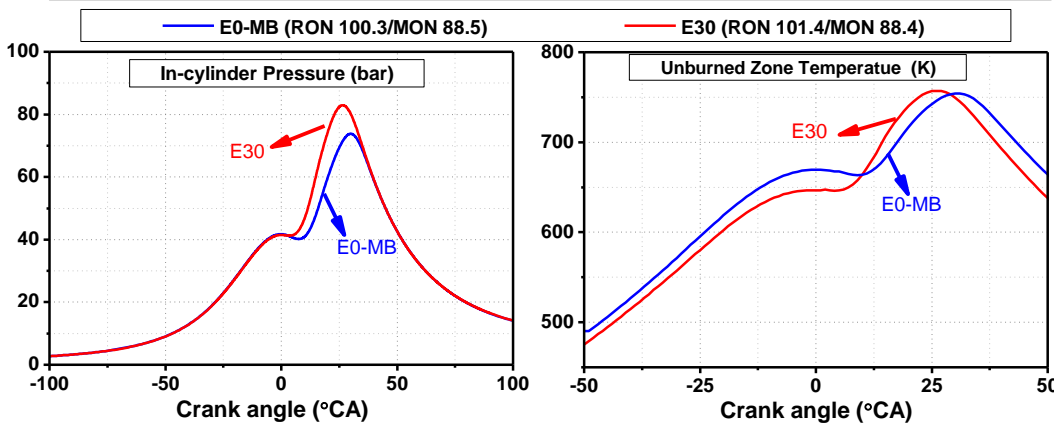
**Effect of charge cooling**  
 Engine speed:1800 rpm; Spark timing: MBT/fixed knock intensity  
 Same valve timing and injection maps for all fuels; Lambda=1; CR=11.5:1



**Figure 7:** Results for charge cooling effect

452  
453  
454

**Results of Charge Cooling**  
 Engine speed:1800 rpm; Intake manifold Pres.: 2 bar Spark timing: MBT/fixed knock intensity  
 Same valve timing and injection maps for all fuels; Lambda=1; CR=11.5:1



**Figure 8:** In-cylinder pressure and unburned zone temperature (calculated by AVL Concerto) of E0-MB and E30 at 1800 rpm engine speed and 2 bar intake manifold pressure

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458



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459

460

461 **Table 1:** Fuel properties\*

462 **Table 2:** Test matrix

463 **Table 3:** Key engine boundary conditions

464

465 **Table 1: Fuel properties\***

|                                       |                   | <b>E10</b> | <b>E20</b> | <b>E30</b> | <b>E85</b> | <b>E0-MB</b> | <b>F1</b> | <b>F2</b> | <b>F3</b> |
|---------------------------------------|-------------------|------------|------------|------------|------------|--------------|-----------|-----------|-----------|
| <b>Ethanol</b>                        | vol. %            | 10         | 20         | 30         | 85         | 0            | 10        | 10        | 10        |
| <b>RON</b>                            | -                 | 96.5       | 99.0       | 101.4      | 107.2      | 100.3        | 92.5      | 98.1      | 98.2      |
| <b>MON</b>                            | -                 | 85.2       | 87.2       | 88.4       | 89.0       | 88.5         | 87.7      | 92.2      | 87.8      |
| <b>Octane sensitivity</b>             | -                 | 11.3       | 11.8       | 13.0       | 18.2       | 11.8         | 4.8       | 5.9       | 10.4      |
| <b>HOV</b>                            | kJ/kg_fuel        | 427.7      | 490.3      | 551.3      | 864.3      | 365.5        | 402.9     | 394.5     | 423.5     |
| <b>HOV**</b>                          | kJ/kg_mixture     | 28.7       | 34.2       | 40.2       | 81.5       | 24.2         | 26.86     | 25.9      | 28.8      |
| <b>Oxygen Content</b>                 | wt. %             | 3.7        | 7.5        | 11.2       | 30.6       | 2.2          | 3.92      | 4.0       | 4.5       |
| <b>Lower Calorific Value</b>          | MJ/kg             | 41.6       | 40.1       | 38.4       | 29.6       | 42.1         | 42.6      | 43.0      | 41.6      |
|                                       | MJ/L              | 30.8       | 30.0       | 28.9       | 23.3       | 31.5         | 30.6      | 30.1      | 30.3      |
| <b>Stoichiometric AFR</b>             | -                 | 13.9       | 13.3       | 12.7       | 9.6        | 14.1         | 14.0      | 14.2      | 13.7      |
| <b>Density</b>                        | kg/m <sup>3</sup> | 741.7      | 747.2      | 752.9      | 785.8      | 748.4        | 718.0     | 698.6     | 730.0     |
| <b>Dry Vapour Pressure Equivalent</b> | kPa               | 77.2       | 75.4       | 73.1       | 36.0       | 64.7         | 54.5      | 59.8      | 59.3      |
| <b>Initial Boiling Point</b>          | °C                | 30.3       | 30.1       | 30.2       | 49.0       | 28.8         | 35.9      | 35.3      | 35.8      |
| <b>Final Boiling Point</b>            | °C                | 192.2      | 190.8      | 189.2      | 79.2       | 182.6        | 194.2     | 194.8     | 193.4     |

466 \*RON and MON were measured by CFR engines; HOV was estimated by using the detailed hydrocarbon analysis results from  
 467 GCMS, and a HOV library; Oxygen content was calculated from the GCMS results. Lower calorific value was calculated by  
 468 using the detailed hydrocarbon analysis results from GCMS, and a lower calorific value library.

469 \*\*At stoichiometric AFR

470  
 471 **Table 2: Test matrix**

|                       | <b>Fuels</b> | <b>Engine Speed</b> | <b>Intake manifold pressure</b> | <b>Spark timing*</b> |
|-----------------------|--------------|---------------------|---------------------------------|----------------------|
|                       |              | rpm                 | bar                             | -                    |
| Load sweep            | all fuels    | 1800                | 0.65-2                          | MBT/KLSA             |
| Spark timing sweep    | E10-E85      | 1800                | 1.6                             | KLSA-2 to KLSA+6     |
| Full load Performance | E10-E85      | 1000-4500           | varied                          | KLSA                 |

472 \* KLSA is defined by the knock limit listed in Table 3.

473  
 474 **Table 3: Key engine boundary conditions**

| <b>Parameter</b>                  | <b>unit</b> | <b>Boundary</b>   |
|-----------------------------------|-------------|---|
| Intake air temperature            | °C          | 34±2  |
| Peak in-cylinder pressure         | bar         | ≤ 130 (continuously)  |
| Exhaust temperature               | °C          | ≤ 840   |
| Mean peak knock intensity (MPKI): | bar         | For 1000-1800 rpm engine speed, MPKI ≤ 0.3 bar;<br>For 2500 rpm engine speed, MPKI ≤ 0.5 bar;<br>For 3500 rpm engine speed, MPKI ≤ 0.7 bar;<br>For 4500 rpm engine speed, MPKI ≤ 0.9 bar; |
| In-cylinder pressure rise rate    | bar/CAD     | ≤ 6   |
| Relative air fuel ratio           |             | 1 or > 0.75 if fuel enrichment is needed  |
| Exhaust back pressure             | bar         | 1 bar at throttled conditions, and the same as the intake manifold pressure at boosted conditions   |

475

476 **Appendix**

477

478 **Table A1:** Valve timing and injection strategy for load sweep test at the engine speed of 1800 rpm

| Speed | IMEP | Intake valve<br>open/close<br>timing @ 1mm<br>valve lift | Exhaust valve<br>open/close<br>timing @ 1mm<br>valve lift | Injection timing       | Injection<br>split ratio |
|-------|------|--|---|------------------------|--------------------------|
| rpm   | bar  | °aTDC  | °aTDC   | °aTDC                  | -                        |
| 1800  | 4    | -12.2/179.2  | -204.4/7.0  | -280                   | -                        |
| 1800  | 6.5  | -12.2/179.2  | -204.4/7.0  | -280; -240             | 1:1                      |
| 1800  | 8    | -12.2/179.2  | -204.4/7.0  | -280; -240             | 1:1                      |
| 1800  | 9.5  | -12.2/179.2  | -204.4/7.0  | -280; -240             | 1:1                      |
| 1800  | 12   | -2.2/189.2   | -214.3/-3.0   | -280; -240; -200       | 1:1:1                    |
| 1800  | 14   | -2.2/189.2   | -214.3/-3.0   | -280; -240; -200       | 1:1:1                    |
| 1800  | 16   | 12.8/204.1   | -214.3/-3.0   | -280; -240; -200       | 1:1:1                    |
| 1800  | 18   | 17.8/209.1   | -214.3/-3.0   | -280; -240; -200       | 1:1:1                    |
| 1800  | 20   | 17.8/209.1   | -214.3/-3.0   | -325; -285; -245; -205 | 1:1:1:1                  |

479

480 **Table A2:** Valve timing and injection strategy for full load test

| Speed | IMEP | Intake valve<br>open/close timing<br>@ 1mm valve lift | Exhaust valve<br>open/close timing<br>@ 1mm valve lift | Injection timing             | Injection<br>split ratio |
|-------|------|---|--|------------------------------|--------------------------|
| rpm   | bar  | °aTDC   | °aTDC  | °aTDC                        | -                        |
| 1000  | 15   | 12.8/204.1  | -219.3/-8.0  | -280; -240; -200             | 1:1:1                    |
| 1800  | 20   | 17.8/209.1  | -214.3/-3.0  | -280; -240; -200             | 1:1:1                    |
| 2500  | 22   | 22.8/214.1  | -214.3/-3.0  | -325; -285; -245; -205       | 1:1:1:1                  |
| 3500  | 21   | 12.8/204.1  | -214.3/-3.0  | -325; -285; -245; -205; -165 | 1:1:1:1:1                |
| 4500  | 20   | 2.8/194.2   | -214.3/-3.0  | -325; -285; -245; -205; -165 | 1:1:1:1:1                |

481

482 **Table A3:** Uncertainty assessment of key instrument

| Instrument                      | Manufacture        | Model number    | Measuring Range           | Uncertainty |
|---------------------------------|--------------------|-----------------|---------------------------|-------------|
| In-cylinder pressure transducer | AVL                | GU22C           | 0 – 250 bar               | ±1%*        |
| Dynamometer                     | HBM                | T40B            | -1000 – 1000 Nm           | ±0.01%      |
| Fuel flow meter                 | AVL                | 735             | Maximum 120 kg/h          | ±0.12%      |
| Air flow meter                  | Elster-Instromet   | Rabo G65        | 0 – 100 m <sup>3</sup> /h | ±0.1%**     |
| Thermocouple                    | Rössel Messtechnik | AL-KB-3,0-150-2 | -200 – 1300 °C            | ±0.1 °C     |

483 \*Thermal shock error to  $\Delta P_{\max}$

484 \*\*Dependent on calibration tool

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