

1 Investigation to charge cooling effect and combustion characteristics of ethanol direct injection in a gasoline port
2 injection engine

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13 Abstract

14 Ethanol direct injection has the potentials to increase the engine compression ratio and thermal efficiency by taking
15 advantages of ethanol fuel such as the high octane number and latent heat. In this study, CFD modelling and
16 experiments were carried out to investigate the charge cooling effect and combustion characteristics of ethanol direct
17 injection in a gasoline port injection (EDI+GPI) engine. Experiments were conducted on a single-cylinder spark
18 ignition engine equipped with EDI+GPI over a full range of ethanol ratio from 0% (GPI only) to 100% (EDI only).
19 Multidimensional CFD simulations to the partially premixed dual-fuel spray combustion were performed to
20 understand the experimental results. The simulations were verified by comparing with the experimental results.
21 Simulation results showed that the overall cooling effect of EDI was enhanced with the increase of ethanol ratio from
22 0% to 58%, but was not enhanced with further increase of ethanol ratio. When the ethanol ratio was greater than 58%,
23 a large number of liquid ethanol droplets were left in the combustion chamber during combustion and fuel
24 impingement on the cylinder wall became significant, leading to local overcooling in the near-wall region and over-
25 lean mixture at the spark plug gap. As a consequence, the CO and HC emissions increased due to incomplete
26 combustion. Compared with GPI only, the faster flame speed of ethanol fuel contributed to the greater peak cylinder
27 pressure of EDI+GPI condition, which resulted in higher power output and thermal efficiency. Meanwhile, the
28 mixture became leaner with the increase of ethanol ratio. As a result, the IMEP was increased, combustion initiation
29 duration and major combustion duration were decreased when ethanol ratio was in 0%-58%. The combustion
30 performance was deteriorated when ethanol ratio was greater than 58%. Experimental and numerical results showed

31 that the IMEP, thermal efficiency and emissions of this EDI+GPI engine can be optimized in the range of ethanol
32 ratio of 40-60%.

33 Keywords:

34 Ethanol direct injection; Gasoline port injection; CFD modelling; Cooling effect; Combustion characteristics

35 1. Introduction

36 Engine downsizing is a promising technology to achieve the future CO₂ reduction target of spark ignition (SI) engines
37 [1-4]. However one major issue associated with the downsized engines is the increased knock propensity [1, 4].
38 Recently ethanol direct injection (EDI) has emerged as a potential technology to fully implement the engine
39 downsizing. The engine knock propensity can be reduced by the higher octane number of ethanol fuel, and
40 supplemented by the cooling effect enhanced by direct injection and ethanol's greater latent heat.

41 Compared with port injection (PI), direct injection (DI) is more effective for charge cooling due to fuel evaporation
42 inside the combustion chamber. Moreover, cooling effect of DI can be further enhanced by the fuel with greater latent
43 heat of vaporization, such as ethanol fuel. Cooling effect of DI has been measured in different ways. The most
44 effective way may be to measure the in-cylinder temperature directly. Kar et al. [5] and Price et al. [6] used a cold
45 wire resistance thermometer to measure the in-cylinder temperature in PI and DI engines. However this method
46 requires fast response of the temperature sensor and protection for the fragile sensor. So the measurements were only
47 performed in non-firing conditions [5, 6]. The Planar Laser Induced Fluorescence (PLIF) thermometry technique was
48 used to measure the cylinder temperature of DI engines [7]. Up to date, the experimental methods to quantify the
49 charge cooling used the parameters linked to the charge cooling directly or indirectly, such as in-cylinder pressure,
50 volumetric efficiency, anti-knock ability, etc. Ahn et al. [8] used in-cylinder pressure to evaluate the cooling effect of
51 ethanol fuel. Wyszynski et al. [9] measured the volumetric efficiency of different fuels on a DI SI engine fitted with
52 both port and direct fuel injection systems. However, using intake air flow rate to quantify the amount of charge
53 cooling only captured part of the cooling effect that took place during the intake stroke. Fuel evaporation process may
54 continue after the intake valves are closed, and even in the combustion process [10]. To evaluate the cooling effect on
55 a special aim, knock onset was used to measure the charge cooling effect in a turbocharged SI engine equipped with
56 both PI and DI of blended ethanol/gasoline fuels [10, 11]. Similar investigation was carried out in an attempt to
57 identify the thermal and chemical benefits of DI and PI [12]. They reached the same conclusion that the ethanol's
58 cooling effect enhancement to the engine performance was comparable to that of its higher Octane number [11, 12].
59 To quantify the thermal and chemical benefits of ethanol fuel, it is reported that a 2-8 kJ/kg increase of "cooling

60 power" of the mixture had the same impact as one-point increase of research octane number (RON) [1]. Or 10% of
61 ethanol addition to gasoline results in five-point increase of RON [13].

62 Meanwhile, numerical simulations have also been applied to investigate the cooling effect. 0-D simulations (involving
63 no engine geometry) were performed to calculate the theoretical improvement in volumetric efficiency of DI over PI
64 [9]. 1-D gas dynamics and thermodynamics engine simulations were carried out to investigate the anti-knock effect of
65 direct injection with ethanol/gasoline blends [11]. As the 0-D and 1-D simulations were developed for special
66 purposes, the information obtained in the results was limited. Kasseris et al. [10] used 3-D numerical modelling to
67 investigate the effect of intake air temperature on the amount of realized charge cooling. The simulation results
68 showed that almost all the theoretical charge cooling was realized when the intake air temperature was increased to
69 120 °C. However the simulated evaporation rate of ethanol fuel in low temperature conditions (naturally aspirated
70 engines) was much lower than gasoline's [14, 15]. This limited the cooling effect of ethanol fuel.

71 Since ethanol has high latent heat and low evaporation rate, EDI is not appropriate to be used on SI engines alone in
72 cold conditions (e.g. cold start problem) [14]. One alternative way is to use it with gasoline port injection (GPI).
73 Studies have investigated the dual-injection concept. The dual-injection concept for knock mitigation with E85 DI
74 plus gasoline PI was tested [16]. The combustion characteristics of three different dual-injection strategies, including
75 gasoline PI plus gasoline DI, gasoline PI plus E85 DI, and E85 PI plus gasoline DI, were investigated [17]. The dual-
76 injection concept of gasoline PI and ethanol or DMF DI was studied as a flexible way to use bio-fuels [18]. The knock
77 mitigation ability [19] and combustion characteristics [20] of dual-injection strategy were examined. The leveraging
78 effect and knock mitigation of EDI in a GPI SI engine (EDI+GPI) were investigated recently [21, 22].

79 The above reviewed experimental studies have shown advantages of EDI+GPI over the conventional PI engines. The
80 thermal efficiency was improved [16-18, 21] and knock propensity was reduced [16, 19, 22], while some reported the
81 increase of HC, CO [21, 22] or NO emissions [19] when EDI was applied. Although experimental investigations are
82 reliable and essential in the development of EDI+GPI engine, they are costly and difficult to understand the in-
83 cylinder mixture formation and combustion mechanisms of this new combustion system. Nowadays, multi-
84 dimensional computational fluid dynamics (CFD) modelling has been proven a useful tool to exploit the detailed and
85 visualised information about the in-cylinder flows. The dual-fuel combustion of in-cylinder fuel blending by gasoline
86 port injection and early diesel direct injection was modelled with a 60 degree sector mesh of the combustion chamber
87 [23]. The combustion and emission characteristics of a dual-fuel injection system with gasoline port injection and
88 diesel direct injection were numerically investigated with a 45 degree sector mesh [24]. However, since the

89 computational meshes used in refs. [23, 24] did not include the intake manifold, the gasoline port injection spray was
 90 not modelled. The dual-fuel combustion with diesel direct injection and natural gas premixed with air in the intake
 91 manifold was simulated [25]. CFD modelling was conducted to investigate the spray, mixture preparation and
 92 combustion processes in a spray-guided DI SI engine [26]. CFD models coupled with detailed chemical reaction
 93 mechanisms were applied to simulate the multi-component fuel spray combustion [27, 28]. However, coupling the
 94 chemistry with the CFD solver is very time consuming and incompatible for complex industrial configurations [29,
 95 30]. Instead, Extended Coherent Flame Model (ECFM) was adopted to simulate the combustion process of SI engines
 96 [29, 31, 32]. To accommodate the increasingly complex chemical kinetics, realistic turbulence/chemistry interaction
 97 and multiple combustion regimes in three-dimensional time-dependent device-scale CFD modelling is a difficult task
 98 in turbulent combustion [33]. A hybrid approach of probability density function (PDF) method and laminar flamelet
 99 model was applied to address the issue [33]. To reduce the computational cost, the complex reaction mechanisms can
 100 be pre-computed and stored in look-up tables [30, 34]. The ECFM combined with PDF look-up tables were used to
 101 model the turbulent diesel spray flames [35, 36]. A presumed PDF model was applied to predict the turbulent flow
 102 behavior and temperature distribution of a diesel spray combustion flame [37]. A tabulated chemistry method was
 103 developed to investigate turbulence-chemistry interactions of premixed, non-premixed and partially premixed flames
 104 [30]. By reviewing the above numerical studies, few publication was found on studying the cooling effect and spray
 105 combustion of dual-injection engine. Moreover simultaneously tracking the evaporation and combustion processes of
 106 two fuels is challenging and computationally consuming.

107 In this study, the cooling effect and combustion characteristics of a novel fuel system, ethanol direct injection plus
 108 gasoline port injection (EDI+GPI), were numerically and experimentally investigated in a full range of ethanol ratio
 109 from 0% (GPI only) to 100% (EDI only).

Nomenclature		IMEP	Indicated mean effective pressure
ASOI	After the start of injection	MFB	Mass fraction burnt
BTDC	Before top dead centre	PDF	Probability density function
CAD	Crank angle degrees	PI	Port injection
CFD	Computational fluid dynamics	RON	Research octane number
DI	Direct injection	SI	Spark ignition
ECFM	Extended Coherent Flame Model	Φ	Equivalence ratio
EDI	Ethanol direct injection	CA0-10%	Combustion initiation duration
GPI	Gasoline port injection	CA10-90%	Major combustion duration
EDI+GPI	Ethanol direct injection plus gasoline port injection	E'X'	X% ethanol by volume. e.g. E46 is 46% ethanol via DI + 54% gasoline via PI

110 2. Experimental setup

111 2.1. EDI+GPI engine

112 Fig. 1 shows the schematic of the EDI+GPI research engine and Table 1 gives the engine specifications. The engine
113 was modified from a single cylinder, four-stroke, air-cooled SI engine which was used on the Yamaha YBR250
114 motorcycle. It was modified to EDI+GPI engine by adding an EDI fuel system to the engine. The EDI injector was a
115 six-hole injector with a spray angle of 34° and a bent angle of 17° . The EDI injector was mounted with spray plumes
116 bent towards the spark plug to create an ignitable mixture around the spark plug. Both the GPI injector and EDI
117 injector were controlled by an electronic control unit. The EDI+GPI fuel system offers the flexibility to operate the
118 engine over a full range of ethanol ratio from 0% (GPI only) to 100% (EDI only). The cylinder pressure, engine
119 torque, intake and exhaust temperatures, cylinder head temperature and emissions were measured during the
120 experiments, which provided experimental data for engine modelling. More information about the engine test system
121 and EDI injector can be found in [21, 38].

122 2.2. Engine operating conditions

123 Table 2 lists the tested engine conditions in the present study. The engine was run at 4000 rpm and 36% throttle open
124 which was the medium engine load in [21]. The lambda was monitored and kept around one by adjusting the mass
125 flow rates of the gasoline and ethanol fuels at a designated fuel ratio and a fixed throttle position. Horiba MEXA-
126 584L gas analyser can measure the lambda of multiple fuels with atomic ratios of hydrogen to carbon (H/C) and
127 oxygen to carbon (O/C) of the fuel input by the user. To ensure the accuracy and correction of the lambda value, the
128 lambda measured by the Horiba gas analyser was also compared with the one calculated using the mass flow rates of
129 the gasoline fuel, the ethanol fuel and the intake air. The intake air flow rate was measured by a ToCeil20N hot-wire
130 thermal air-mass flow meter. The gasoline and ethanol fuel flow rates were determined by the injection pulse width of
131 the injectors in the engine control unit. The fuel injectors were calibrated by the Hents Technologies Inc. at various
132 injection pressures and pulse widths. A linear function between the injector's pulse width and fuel mass was derived
133 from the calibration results. The calibration of fuel mass and pulse width has shown good and stable linearity at
134 different injection pressures. The EDI injection timing was 300 CAD BTDC and the GPI timing was 410 CAD BTDC.
135 EDI timing of 300 CAD BTDC was for providing sufficient time for ethanol fuel to evaporate and to mix with air
136 before the combustion took place. The spark timing was 15 CAD BTDC which was the spark timing in the original
137 engine control system. The EDI pressure was 6.0 MPa and the GPI pressure was 0.25 MPa. The ethanol ratio was

138 varied from 0% (GPI only) to 100% (EDI only), including E0, E25, E46, E58, E69, E76, E85 and E100 (E'X' means
139 X% ethanol by volume. e.g. E46 is 46% ethanol via DI + 54% gasoline via PI).

140 3. Computational models

141 3.1. Dual-fuel spray combustion modelling

142 The numerical simulations were performed with the CFD code ANSYS FLUENT. The in-cylinder flows were
143 modelled using the RANS based realizable k - ϵ turbulence model. The EDI and GPI sprays were simulated by the
144 Discrete Droplet Model (DDM) based on the Eulerian-Lagrangian approach. A set of sub-models were adopted to
145 take into account the effects of break-up, fuel evaporation, droplet-gas momentum exchange, and droplet-wall
146 interaction. The primary breakup process is modelled by the Rosin-Rammler Diameter Distribution Method based on
147 the blob injection concept which assumes the initial droplets or blobs to be similar to the injector hole diameter at the
148 nozzle exit [39-42]. The consequent droplet breakup process was modelled by the WAVE model [43]. Dynamic Drag
149 model was used to take into account the droplets distortion and drag [44]. Since the simulated cases were completely
150 warmed up engine conditions, the cylinder wall was hot and the Wall-jet model was adopted to model the droplet-wall
151 interactions [45]. Convection/Diffusion Controlled Model [46] was adopted to model the evaporation process of
152 ethanol and gasoline droplets. It uses the vapour pressure as the driving force for droplets evaporation and
153 incorporates the effect of the convective flow on the evaporating materials from the droplet surface to the bulk gas
154 phase. The evaporation model provided the combustion model with the amount of vapour fuel for each fuel.

155 Spray combustion in SI engines is a typical partially premixed combustion which shows features of both non-
156 premixed and premixed combustion. The fuel is injected into the combustion chamber in liquid form and evaporation
157 and diffusion processes occur prior to the combustion. By the time of combustion, part of the fuel has mixed with the
158 oxidizer in molecular level but inhomogeneously, and evaporating and mixing are still occurring. The dual-fuel spray
159 combustion process was modelled using the ECFM combustion model with the partially premixed combustion
160 concept in which both the mixture fraction Z and progress variable c were solved [29, 45, 47]. The combustion
161 process was initiated by releasing a specific amount of energy to the cells at the spark plug gap at the spark timing.
162 The presumed PDF look-up table was used to model the turbulence-chemistry interactions. The chemistry look-up
163 tables were generated using complex reaction mechanisms which incorporated the latest insights on combustion
164 chemical kinetics [34]. For single fuel combustion modelling (GPI only and EDI only conditions), a three-
165 dimensional PDF table was generated to determine the temperature, density, and species fraction in the turbulent
166 flame. For EDI+GPI dual-fuel combustion modelling, a five-dimensional PDF table was generated to take into

167 account the secondary fuel. The computational cost of implementing five-dimensional PDF table was much higher
168 than three-dimensional one. The thermal NO formation was modelled by the extended Zeldovich mechanism [29].

169 *3.2. Computational mesh*

170 The computational mesh was generated based on the scanned geometry of the cylinder head using the ANSYS
171 Meshing. Fig. 2 shows the computational mesh at the start of the calculation. It mainly consists of tetrahedral grids.
172 However the regions with moving boundaries were meshed to hexahedral grids for mesh deforming. A basic
173 requirement for the Lagrangian liquid phase description is that the void fraction within a cell is close to one [48]. To
174 meet this requirement, the grid sizes near the nozzles are at least five times larger than the nozzle diameters [28, 49].
175 An earlier study by the current authors [50] showed that the present mesh was sufficient to achieve the reasonable
176 accuracy and low computational cost. More details about the dynamic mesh and independence study can be found in
177 [50].

178 *3.3. Boundary and initial conditions*

179 The boundary and initial conditions were determined according to the experimental conditions described in Section
180 2.2. The wall temperatures were set up based on the typical temperature distributions for SI engines operating at
181 normal steady state conditions [51]. The wall temperatures were set to be 600 K for the cylinder head, 458 K for the
182 cylinder linear, 573 K for the piston, 523 K for the intake valve, and 923 K for the exhaust valve. The wall
183 temperatures of intake and exhaust ports are assumed to be 333 K and 723 K respectively. The inlet and outlet
184 pressure values were constant as the atmospheric pressure. The intake air temperature was set to be the room
185 temperature of the engine laboratory. Initial conditions for the cylinder, intake and exhaust manifolds were set up
186 according to the measured in-cylinder pressure and exhaust gas temperature.

187 *3.4. Comparison between measured and simulated results*

188 The comparison between the measured and simulated values of in-cylinder pressure and heat release rate at different
189 ethanol ratios are shown in Fig. 3. As shown in Fig. 3, the simulated cylinder pressure and heat release rate, including
190 their magnitudes and phases, agree well with the measured data from the engine experiments. As the ethanol ratio
191 increases to E76, the simulated in-cylinder pressure increases slightly more quickly than the measured one does after
192 the spark timing. However, the start phase and the magnitude of the heat release rate of the simulated curve still match
193 with the measured one at E76. Therefore, the accuracy of the simulation is considered within the acceptable limit
194 considering the current development of dual-fuel combustion modelling.

196 *4.1. Cooling effect and mixture preparation*

197 The cooling effect of EDI is evaluated by comparing the in-cylinder temperature of EDI+GPI (or EDI only) with that
198 of GPI only. Fig. 4 shows the spatial distributions of in-cylinder temperature at different ethanol ratios on a plane cut
199 below the spark plug at spark timing from simulation. The red dot and arrow indicate the position and direction of the
200 EDI injector. As shown in Fig. 4, the charge cooling in the area over the exhaust valve is more effective than that in
201 other areas. This cooling effect becomes stronger with the increase of the ethanol ratio. When the ethanol ratio is
202 greater than or equal to 58%, the near-wall area close to the exhaust valve is over cooled because the temperature is
203 reduced to be lower than 500 K while the mean cylinder temperature is around 690 K. The local overcooling is due to
204 the most concentration of ethanol droplets in this area. In the late compression stroke, the gas velocity becomes low
205 and the ethanol droplets move slowly, causing low heat transfer rate and thus local overcooling. As the ethanol
206 droplets evaporate and absorb the thermal heat from this area, this area has a lower temperature and richer mixture.
207 Such an over-cooled and rich mixture area causes incomplete combustion, and consequently increases the HC and CO
208 emissions.

209 Although overcooling occurs locally in some regions in cylinder, the overall cooling effect does not increase with
210 ethanol ratio when the ethanol ratio is greater than 58%. As shown in Fig. 5, the predicted mean in-cylinder
211 temperature at spark timing decreases quickly with the increase of ethanol content until the ethanol ratio reaches 58%.
212 However, when the ethanol ratio is greater than 58%, the overall cooling effect of EDI does not increase much. This is
213 because the EDI cooling effect is limited by the low evaporation rate of the ethanol fuel due to its low saturation
214 vapour pressure [15]. Fig. 6 shows the simulated results of the variation of the evaporated/unevaporated ethanol and
215 gasoline fuels with the ethanol ratio by spark timing. With the increase of ethanol ratio, the mean cylinder temperature
216 decreases, leading to reduced evaporation rates for both ethanol and gasoline fuels. The evaporation rate of gasoline
217 drops from 94.3% to 92.0% when the ethanol ratio increases from 0% to 85%. The evaporation rate of ethanol drops
218 from 64.0% to 56.8% when the ethanol ratio increases from 25% to 100%. As a result, the total mass of un-
219 evaporated gasoline and ethanol droplets increases rapidly from 0.873 mg to 9.367 mg when the ethanol ratio
220 increases from 0% to 100%. Higher ethanol ratio has greater cooling potential, but may leave a large number of liquid
221 droplets in the chamber by spark timing. These liquid droplets will keep evaporating during the combustion process
222 and the droplet combustion may occur. This is unfavourable for combustion and leads to high HC and CO emissions.

223 Since ethanol fuel evaporates slowly in the low temperature environment before the combustion takes place, high
224 ethanol ratio also leads to lean mixture in the combustion chamber. Fig. 7 shows the distributions of the equivalence
225 ratio (Φ) around the spark plug by spark timing. The equivalence ratio is defined as follows,

$$\Phi = \frac{Y_e \cdot (O/F)_e + Y_g \cdot (O/F)_g}{Y_{O_2}}$$

226

227 where Y_e , Y_g and Y_{O_2} are the local mass fractions of ethanol, gasoline and oxygen in each cell, $(O/F)_e$ and $(O/F)_g$ are
228 the stoichiometric oxygen/fuel ratios of ethanol and gasoline fuels. As clearly shown in Fig. 7, the equivalence ratio at
229 the plug position decreases with the increase of ethanol ratio. High ethanol ratio (> 58%) does not enhance the overall
230 cooling effect of EDI. On the contrary, it deteriorates the consequent combustion and emission processes. When the
231 ethanol ratio is higher than 58%, the equivalence ratio around the spark plug decreases to be less than 0.5 (0.44 in E76
232 and 0.37 in E100). Such a lean mixture is out of the ignitable equivalence ratio range of $0.5 < \Phi < 1.5$ [52]. The lean
233 mixture around the spark plug is difficult to be ignited and consequently leads to incomplete combustion and high HC
234 and CO emissions, whose results will be further discussed in Section 4.2.

235 Moreover, greater ethanol ratio requires longer injection duration of EDI. Longer injection duration enhances the
236 spray penetration and may lead to fuel impingement on the piston and cylinder walls, resulting in increased HC and
237 soot emissions during engine operation [53]. Fig. 8 shows the comparison of the measured and simulated EDI spray
238 patterns at 1.5 ms after the start of injection (ASOI) in a constant volume chamber. The injection pressure was 6 MPa,
239 the ambient temperature was 350 K and the ambient pressure was 1 bar. These conditions reproduced the in-cylinder
240 conditions for an early EDI injection of 300 CAD BTDC in the engine experiments. More information about the spray
241 experiments in the constant volume chamber can be found in [38]. As shown in Fig. 8, the ethanol spray tip
242 penetration reaches 70 mm at 1.5 ms ASOI. The penetration length 70 mm is about the bore diameter (74 mm) and the
243 duration 1.5 ms (equal to 36 CAD at engine speed of 4000 rpm) is close to the EDI injection duration (32 CAD) at
244 ethanol ratio of 46%. Fig. 8 implies that the ethanol fuel impingement may have occurred in engine conditions when
245 ethanol ratio is greater than 46%. Fig. 9 shows the distributions of the ethanol spray droplets at the end of EDI
246 injection at different ethanol ratios in the engine. As shown in Fig. 9, by the end of EDI injection, the ethanol spray tip
247 does not reach the cylinder wall when ethanol ratio is lower than 58%. With the increase of ethanol ratio, the spray
248 penetration length increases and more ethanol droplets reach the cylinder wall, resulting in more wall impingement.
249 This is another factor contributing to the increased HC and CO emissions in the engine experiments, which is shown
250 in Fig. 15.

251 Higher ethanol ratio requires greater latent heat for fuel evaporation. However, the amount of this cooling potential
252 realised is limited by ethanol's low evaporation rate. More ethanol content needs more energy and time to evaporate,
253 which may lead to incomplete evaporation in the same engine condition. The ethanol ratio and its evaporation are two
254 competing factors that determine the final level of cooling effect and combustion performance: lower ethanol ratio (<
255 58%) leads to a higher completeness of cooling effect, but limited by its cooling potential; higher ethanol ratio (> 58%)
256 contains more cooling potential, but only a small percentage of it may be realised. Moreover, when the ethanol ratio is
257 higher than 58%, the near-wall area next to the exhaust valve is over-cooled (shown in Fig. 4), the mixture at the
258 spark plug gap is over-lean (shown in Fig. 7) and the fuel impingement on the cylinder wall becomes more significant
259 (shown in Fig. 9). All these cause incomplete combustion and increased CO and HC emissions. When taking the
260 quality of the mixture into consideration, the competing of cooling potential and its evaporation suggests that 40-60%
261 of ethanol ratio can realise the maximum overall cooling effect while avoiding the local overcooling, the too lean
262 mixture at the spark gap and the fuel impingement on the cylinder wall. A similar ratio (30-50%) has been
263 recommended for ethanol/gasoline blends for the conventional single injection engines [5].

264 4.2. Combustion characteristics

265 To evaluate the combustion characteristics of the EDI+GPI, the in-cylinder pressure, indicated mean effective
266 pressure (IMEP), combustion initiation duration and major combustion duration are discussed. Fig. 10 shows the
267 measured variations of in-cylinder pressure with crank angle degrees at ethanol ratios from 0% to 100%. As shown in
268 Fig. 10, the peak cylinder pressure increases quickly with the increase of ethanol ratio from 0% to 58% and decreases
269 when the ethanol ratio is further increased from 58% to 100%. The in-cylinder pressure with EDI is lower than that of
270 GPI only during the compression stroke (<360 CAD) due to the cooling effect of EDI, leading to less negative work
271 on the piston. During the expansion stroke (>400 CAD), the pressure with EDI is larger than that of GPI only,
272 resulting in more positive work on the piston. Consistently shown in Fig. 11, the IMEP increases quickly when
273 ethanol ratio is in 0%-46% and slowly in 46%-76%, and decreases in 76%-100%.

274 Fig. 12 shows the combustion initiation duration and the major combustion duration at different ethanol ratios from
275 0% to 100% derived from the cylinder pressure shown in Fig. 10. The combustion initiation duration, indicated by
276 CA0-10%, is defined as the crank angle degrees from the spark timing to the timing of 10% of the fuel mass fraction
277 burnt (MFB). CA0-10% is directly relates to the combustion stability and only after CA0-10% does flame velocity
278 reach higher values with the corresponding fast rise in cylinder pressure and flame propagation [51]. The major
279 combustion duration, indicated by CA10-90%, is defined as the crank angle degrees from 10% to 90% MFB. The

280 shorter is the CA10-90%, the closer the combustion process is to the constant volume and consequently the higher the
281 thermal efficiency will be [51]. As shown in Fig. 12, the combustion initiation duration decreases with the increase of
282 ethanol ratio from 0% to 58%, indicating an improved combustion stability. However, the CA0-10% starts to increase
283 quickly when the ethanol ratio is higher than 58%. This can be explained by the results shown in Fig. 7. As shown in
284 Fig. 7, the equivalence ratio decreases with the increase of the ethanol ratio. Within 0%-58%, the equivalence ratio is
285 in the ignitable equivalence ratio range of $0.5 < \Phi < 1.5$. The faster flame speed of ethanol fuel contributes to the
286 shorter combustion initiation duration and thus higher combustion stability. However when the ethanol ratio is higher
287 than 58%, the mixture is too lean and out of the ignitable range (Fig. 7) which causes the increased CA0-10%. On the
288 other hand, the major combustion duration decreases quickly with the increase of ethanol ratio from 0% to 58% but
289 slowly from 58% to 76%, and increases when it changes from 76% to EDI only condition.

290 Fig. 13 shows the flame propagation and distributions of OH mass fraction at 375 CAD and 395 CAD varying with
291 the ethanol ratios. In premixed combustion modelling, the progress variable c is used to indicate the state of the
292 mixture, where $c=0$ indicates fresh mixture, $c=1$ is for burnt and $0 < c < 1$ indicates the flame-brush. As shown by the
293 images at 375 CAD in Fig. 13, the mixture burns more quickly in EDI+GPI condition than that in GPI only when
294 ethanol ratio is less than 76%. The flame speed decreases when the ethanol ratio reaches 100%. By the time of 395
295 CAD, the flame has reached most volume of the combustion chamber. The presence of OH radical is an indicator of
296 the main heat release rate event [54]. Fig. 13 shows that the generation of OH radical is weak at 375 CAD but
297 becomes intensive at 395 CAD. This is consistent with the experimental results shown in Fig. 10, where the cylinder
298 pressure of E100 is smaller in 360-390 CAD but becomes higher after 400 CAD than the pressure of low ethanol ratio
299 conditions. Although EDI+GPI conditions have higher combustion speeds, there are still some unburnt mixture in the
300 near wall region. This is because the ethanol droplets concentrate and evaporate in the near wall region. Fig. 14 shows
301 the distributions of ethanol liquid droplets, equivalence ratio and cylinder temperature at 395 CAD. The ethanol
302 droplets evaporate and absorb thermal heat from the mixture in the near wall region. As a result, this region has a very
303 rich mixture ($\Phi > 2.0$) and is over-cooled (< 500 K). The overcooling and over-rich mixture in the near wall region
304 make it hard for the flame to propagate to this region. Consequently, this region has extensive CO and HC emissions
305 as a result of incomplete combustion. On the other hand, the cylinder temperature is much lower in EDI+GPI
306 condition than that in GPI only condition due to the enhanced cooling effect and lean mixture in EDI+GPI.
307 Particularly, the extremely high temperature region (~ 2500 K) observed in GPI only in Fig. 14 is disappeared when
308 EDI is applied. Following the thermal NO_x mechanism of Zeldovich, the NO formation is less significant in EDI+GPI
309 condition. These explain the measured emission values from the EDI+GPI engine tests. As shown in Fig. 15, the

310 measured CO and HC emissions increase, and NO emission decreases with the increase of ethanol ratio from 0% to
311 100%. Moreover the CO and HC emissions become significantly higher when the ethanol ratio is greater than 58%.
312 The combustion performance of EDI+GPI engine is improved when implementing EDI within ethanol ratio of 0%-
313 58%. The cylinder pressure and IMEP are increased and the combustion initiation and major combustion durations are
314 decreased when ethanol ratio is increased from 0% to 58%. When further increasing the ethanol ratio from 58% to
315 100%, the combustion initiation duration and major combustion duration start to increase, while the cylinder pressure
316 decreases, and IMEP increases slightly from 58% to 76% and decreases from 76% to 100%. Regarding the engine
317 emissions, the NO emission decreases when EDI is applied due to the lower combustion temperature and cooling
318 effect. Meanwhile, the HC and CO emissions are increased, and are extremely high at high ethanol ratios (>58%) due
319 to local overcooling and incomplete combustion. Although the engine shows the maximum IMEP at 76%, the
320 exhaust-out CO and HC emissions are very high when ethanol ratio is higher than 58%. The overall cooling effect
321 does not increase with ethanol ratio greater than 58% but leaves a large number of ethanol droplets unevaporated
322 during combustion. Furthermore, over-lean and local overcooling occur, fuel impingement becomes more significant
323 on cylinder wall, and combustion initiation and major combustion durations increase when ethanol ratio is high.
324 Based on comparison of results in all the aspects, the optimal engine performance may exist in the range of ethanol
325 ratio of 40-60% in terms of IMEP, combustion efficiency, cooling effect and emissions.

326 5. Conclusions

327 The cooling effect and combustion characteristics of a novel fuel system, ethanol direct injection plus gasoline port
328 injection (EDI+GPI), were numerically and experimentally investigated in a full range of ethanol ratio from 0% (GPI
329 only) to 100% (EDI only). The engine was run at medium load and stoichiometric condition with engine speed of
330 4000 rpm, spark timing of 15 CAD BTDC and throttle open of 36%. The EDI pressure was 6.0 MPa and the EDI
331 timing was 300 CAD BTDC. The GPI pressure was 0.25 MPa and the GPI timing was 410 CAD BTDC. The main
332 conclusions can be drawn as follows.

- 333 1. Compared with GPI only, EDI+GPI demonstrates stronger effect on charge cooling with lower in-cylinder
334 temperature and pressure during the compression stroke. The overall cooling effect increases with the
335 increase of ethanol ratio within 0%-58%. Further increase of ethanol ratio does not increase the overall
336 cooling effect, but leaves a large number of liquid ethanol droplets in the combustion chamber during
337 combustion. Moreover, the local overcooling in the near-wall region and the fuel impingement on the cylinder
338 wall become more significant and the mixture becomes too lean when the ethanol ratio is higher than 58%.

- 339 2. The IMEP is increased, and combustion initiation and major combustion durations are decreased when
340 ethanol ratio is in the range of 0%-58%. The combustion performance is deteriorated when the ethanol ratio is
341 greater than 58%, indicated by decreased IMEP and increased combustion initiation and major combustion
342 durations. This is caused by the over-lean mixture around the spark plug, local overcooling and fuel
343 impingement at high ethanol ratio conditions (>58%).
- 344 3. The NO emission is decreased with the increase of ethanol ratio due to the enhanced cooling effect and
345 decreased combustion temperature. Meanwhile, the CO and HC emissions are increased with the increase of
346 ethanol ratio due to the incomplete combustion and increased fuel impingement on cylinder wall. The
347 incomplete combustion is caused by the fact that ethanol fuel evaporates slowly in the low temperature
348 environment before combustion, which consequently leaves a large number of liquid ethanol droplets
349 concentrating in the near-wall region, resulting in locally over-cooled and over-rich mixture.
- 350 4. The experimental and numerical results showed that the IMEP, thermal efficiency and emission performance
351 of this EDI+GPI engine can be optimized in the range of ethanol ratio of 40-60%, resulted from the effective
352 charge cooling and improved combustion efficiency while avoiding the wall wetting, over-lean and local
353 overcooling issues.

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Table 1 EDI+GPI engine specifications.

Engine type	Single cylinder, air cooled, four-stroke
Displacement	249.0 cc
Stroke	58.0 mm
Bore	74.0 mm
Connecting rod	102.0 mm
Compression ratio	9.8:1
Intake valve open	22.20 CAD BTDC
Intake valve close	53.80 CAD ABDC
Exhaust valve open	54.60 CAD BBDC
Exhaust valve close	19.30 CAD ATDC
Ethanol delivery system	Direct injection
Gasoline delivery system	Port injection

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Table 2 Engine operating conditions.

Ethanol ratio by volume	E0	E25	E46	E58	E69	E76	E85	E100
EDI fuel mass (mg)	-	4.0	8.0	10.7	13.4	15.0	17.3	21.5
GPI fuel mass (mg)	13.4	11.0	8.5	7.0	5.5	4.2	2.7	-

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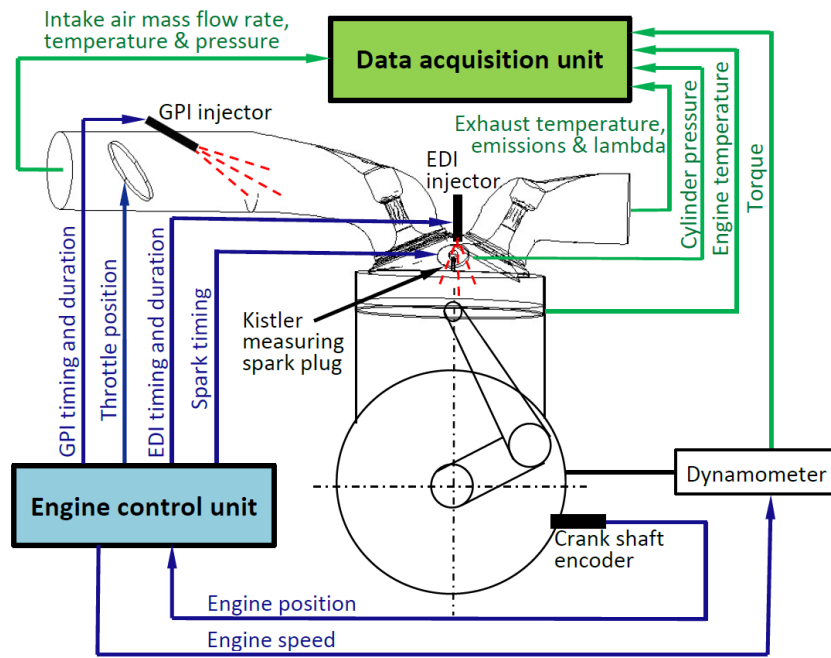


Fig. 1. Schematic of the EDI+GPI engine system. (single column fitting image)

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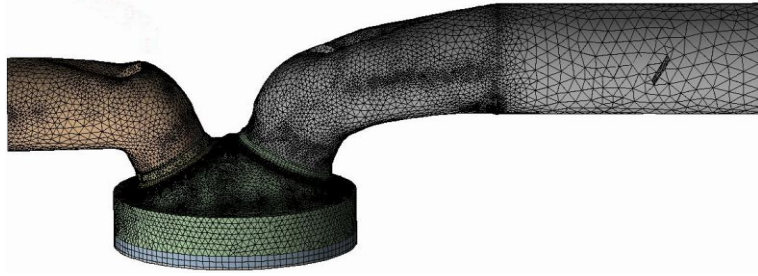


Fig. 2. Computational mesh. (single column fitting image)

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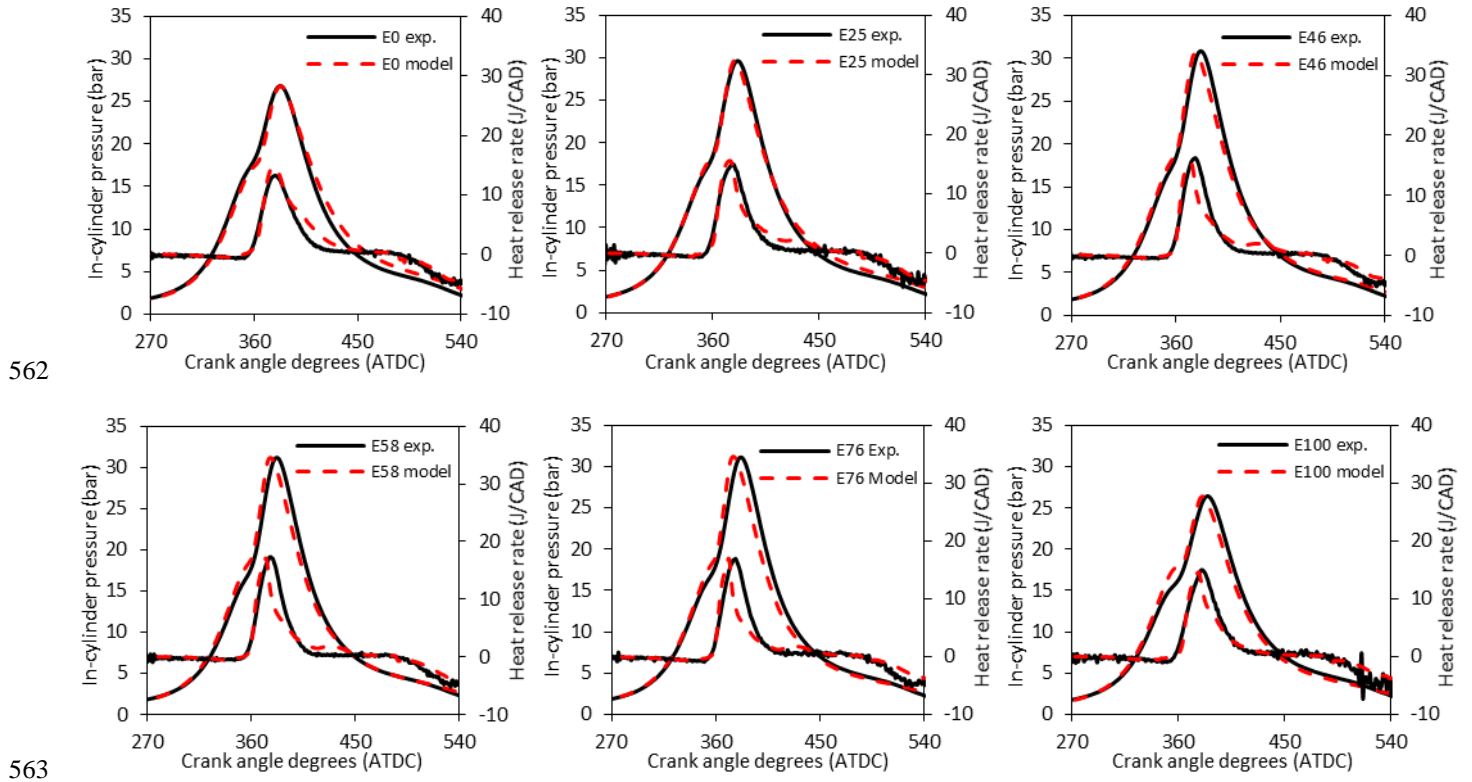
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564 Fig. 3. Comparison between the measured and simulated values of in-cylinder pressure and heat release rate at
 565 different ethanol ratios. (2-column fitting image)

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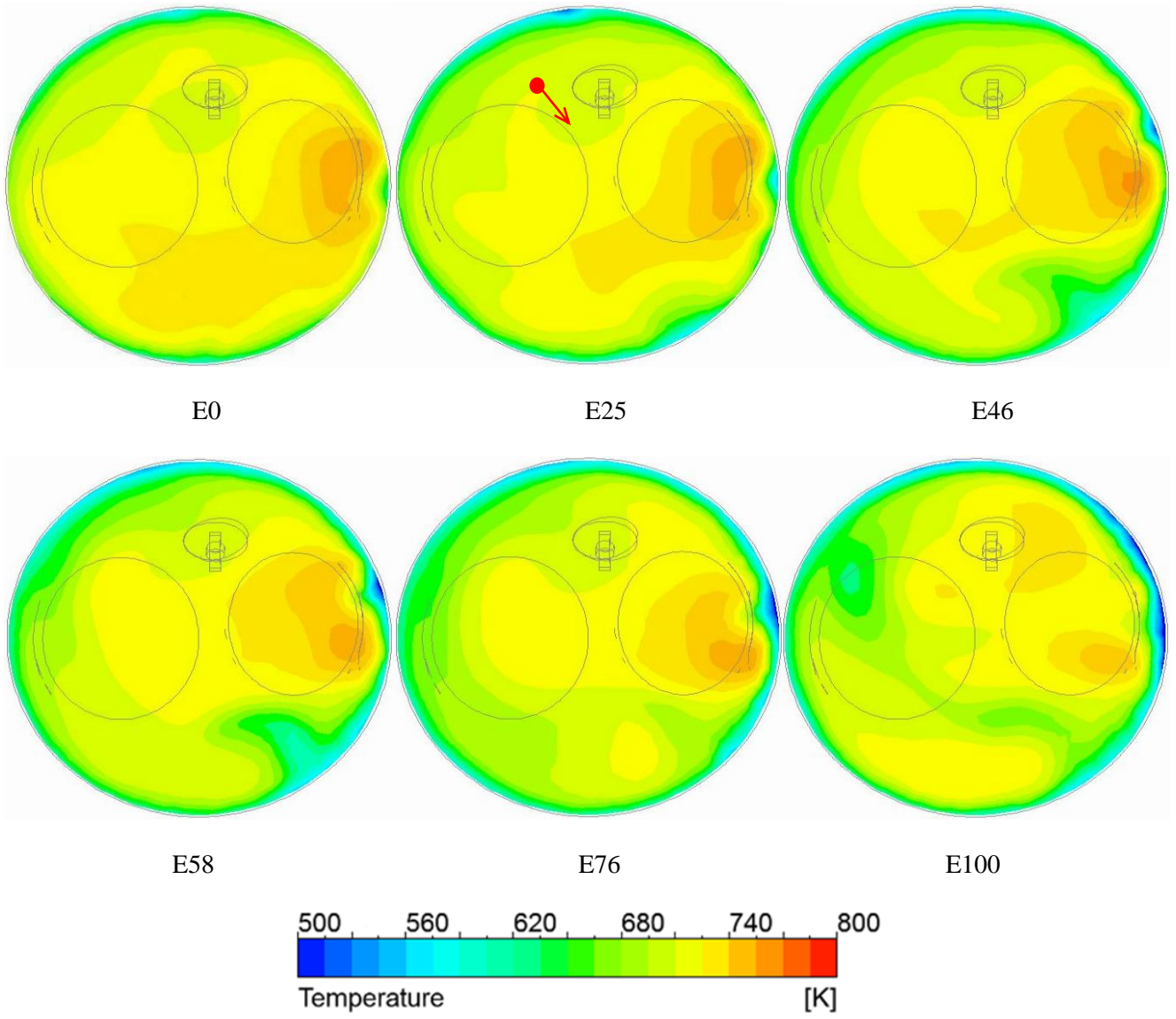
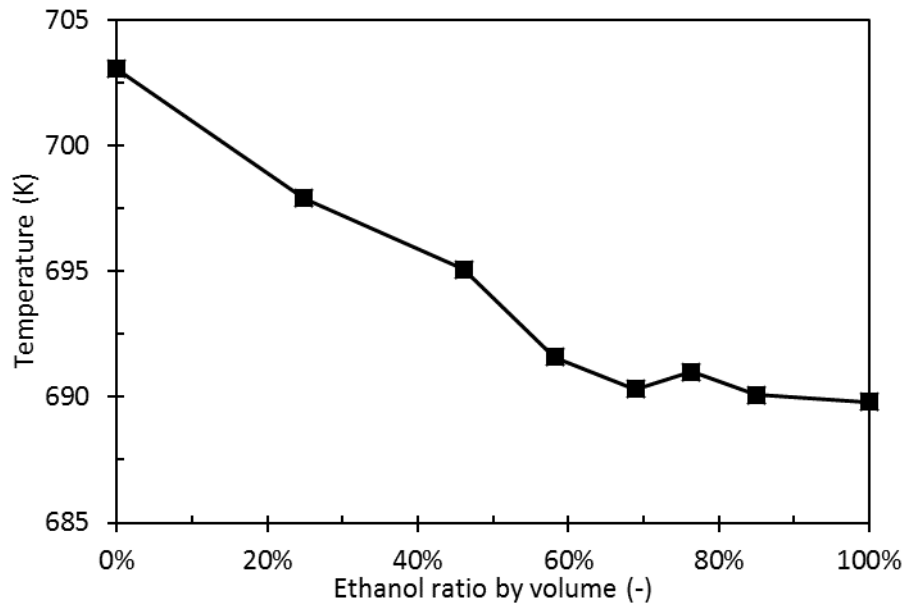


Fig. 4. In-cylinder temperature distributions by spark timing. (2-column fitting image)



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Fig. 5. Variation of mean in-cylinder temperature by spark timing with the ethanol ratios. (single column fitting

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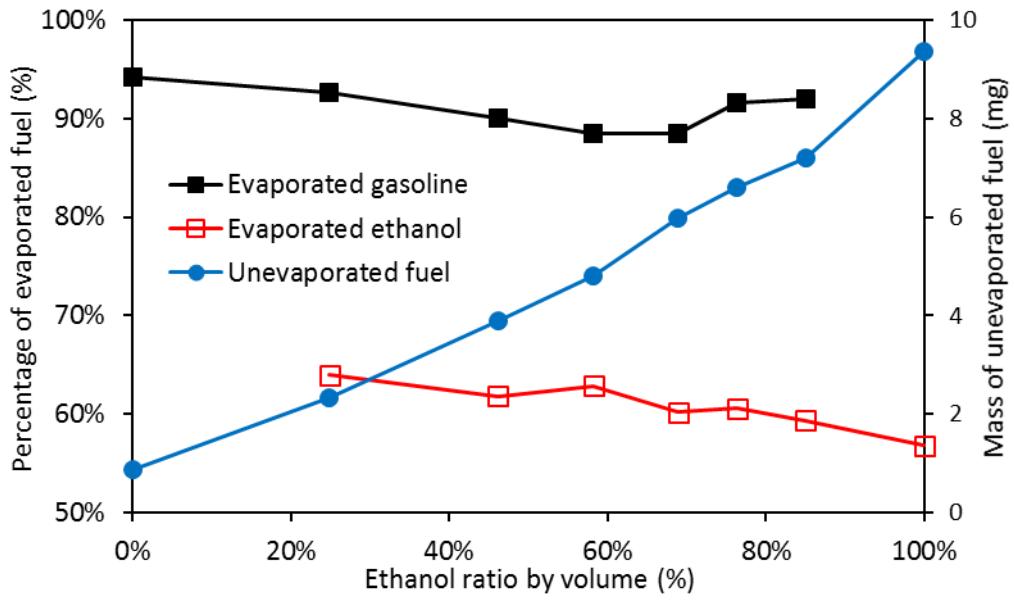
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Fig. 6. Completeness of the ethanol and gasoline evaporation by spark timing. (single column fitting image)

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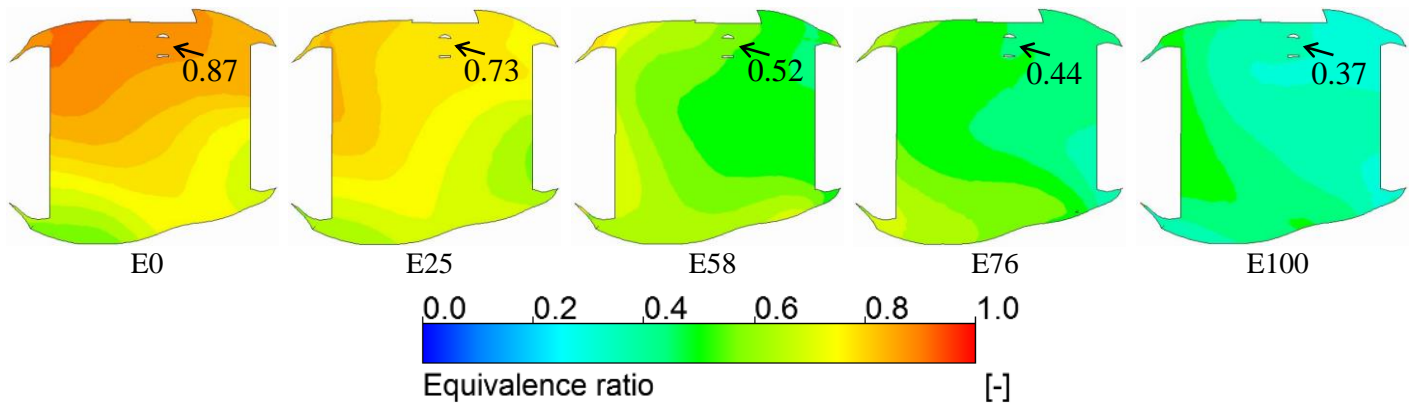
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628 Fig. 7. Distributions of the equivalence ratio around the spark plug by spark timing. (2-column fitting image)

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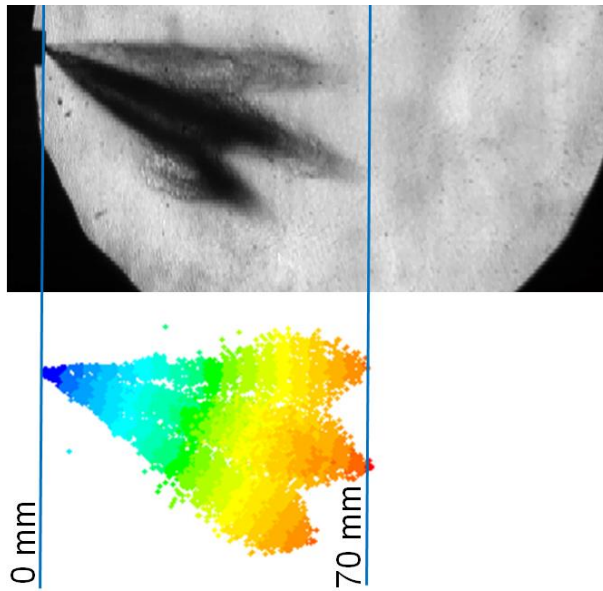
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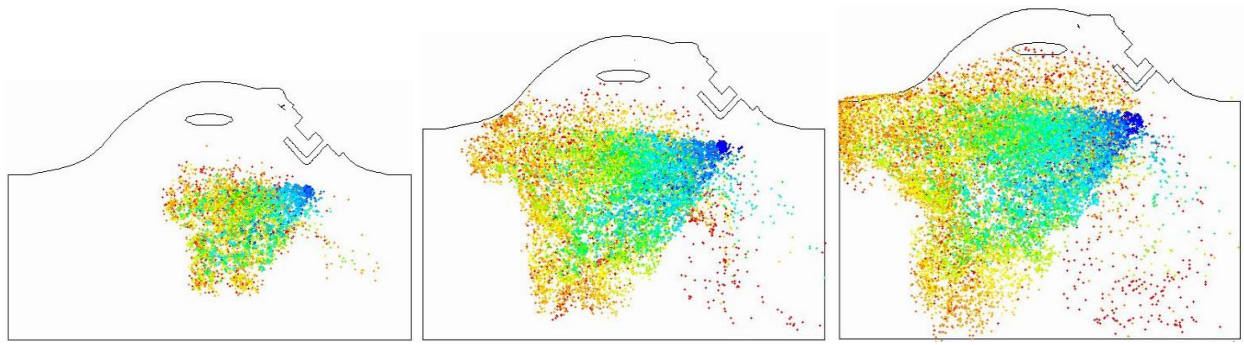
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Fig. 8. Comparison of the experimental and numerical results of EDI spray pattern at 1.5 ms ASOI in a constant volume chamber @ 6.0 MPa injection pressure, 1 bar ambient pressure and 350 K ambient temperature [38, 50].

(single column fitting image)



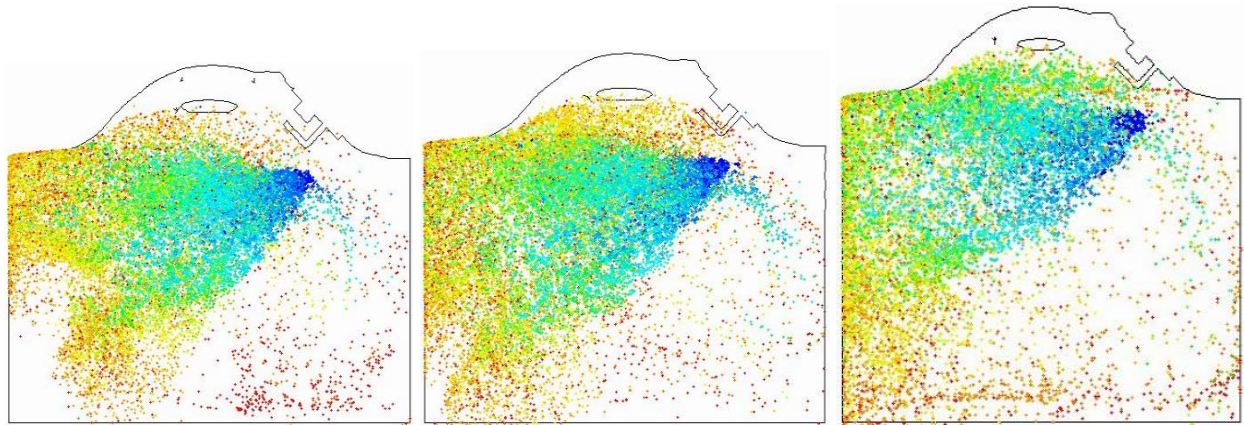
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E25 (16 CAD ASOI)

E46 (32 CAD ASOI)

E58 (43 CAD ASOI)



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E69 (54 CAD ASOI)

E76 (60 CAD ASOI)

E100 (86 CAD ASOI)

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Fig. 9. Distributions of the ethanol spray droplets at the end of EDI injection. (2-column fitting image)

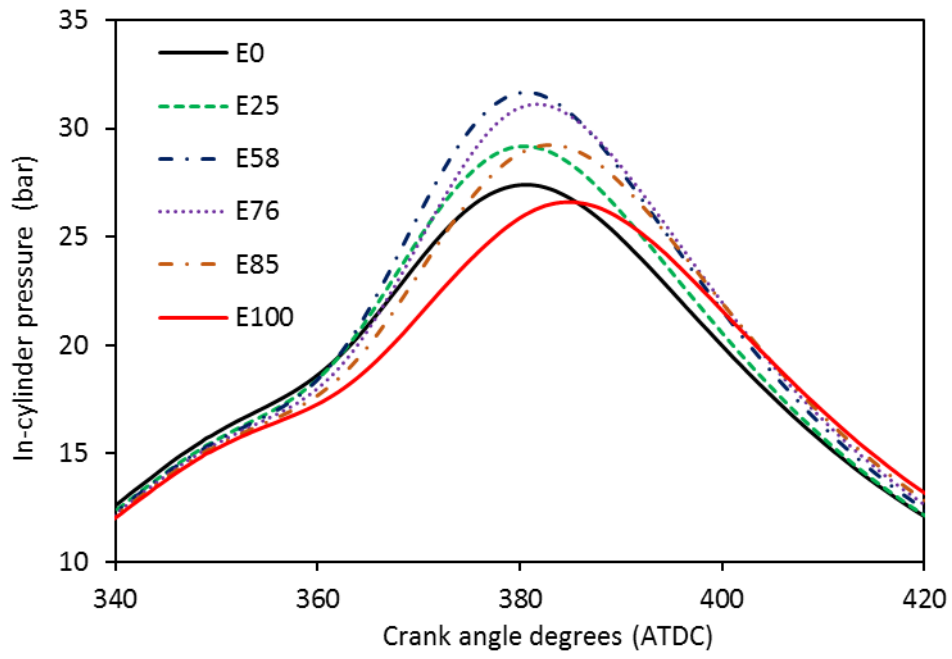
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Fig. 10. In-cylinder pressure varying with the ethanol ratios. (single column fitting image)

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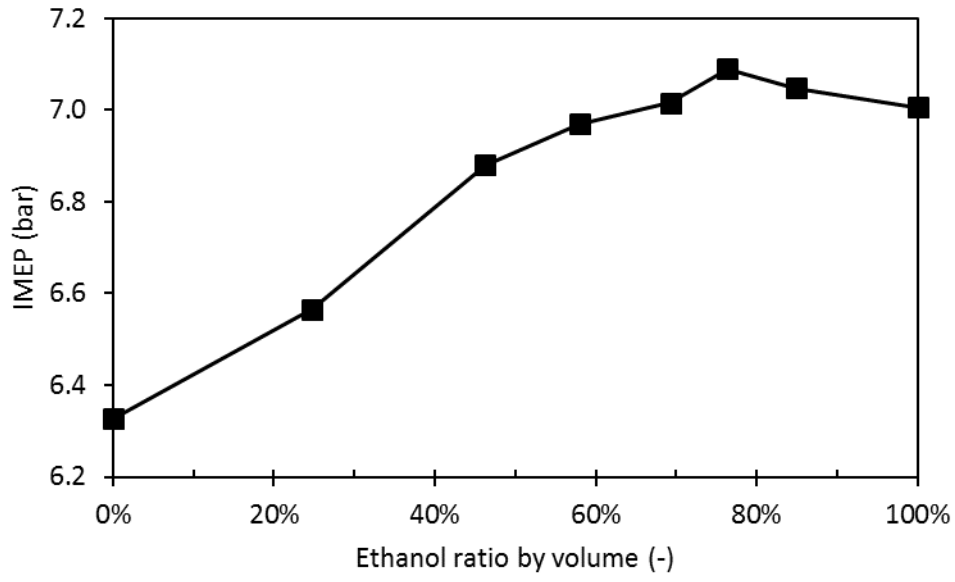
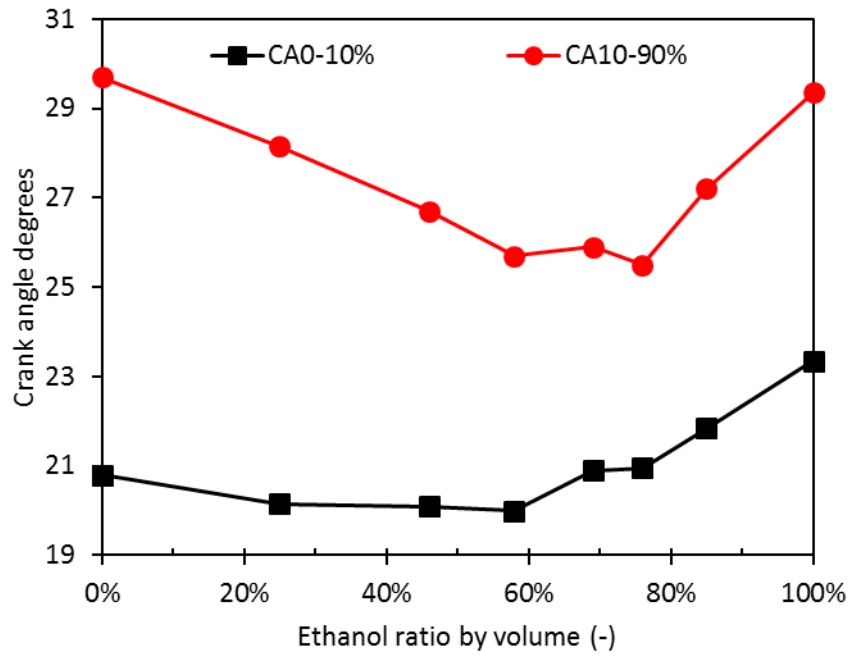


Fig. 11. IMEP varying with the ethanol ratios. (single column fitting image)

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Fig. 12. CA0-10% and CA10-90% varying with the ethanol ratios. (single column fitting image)

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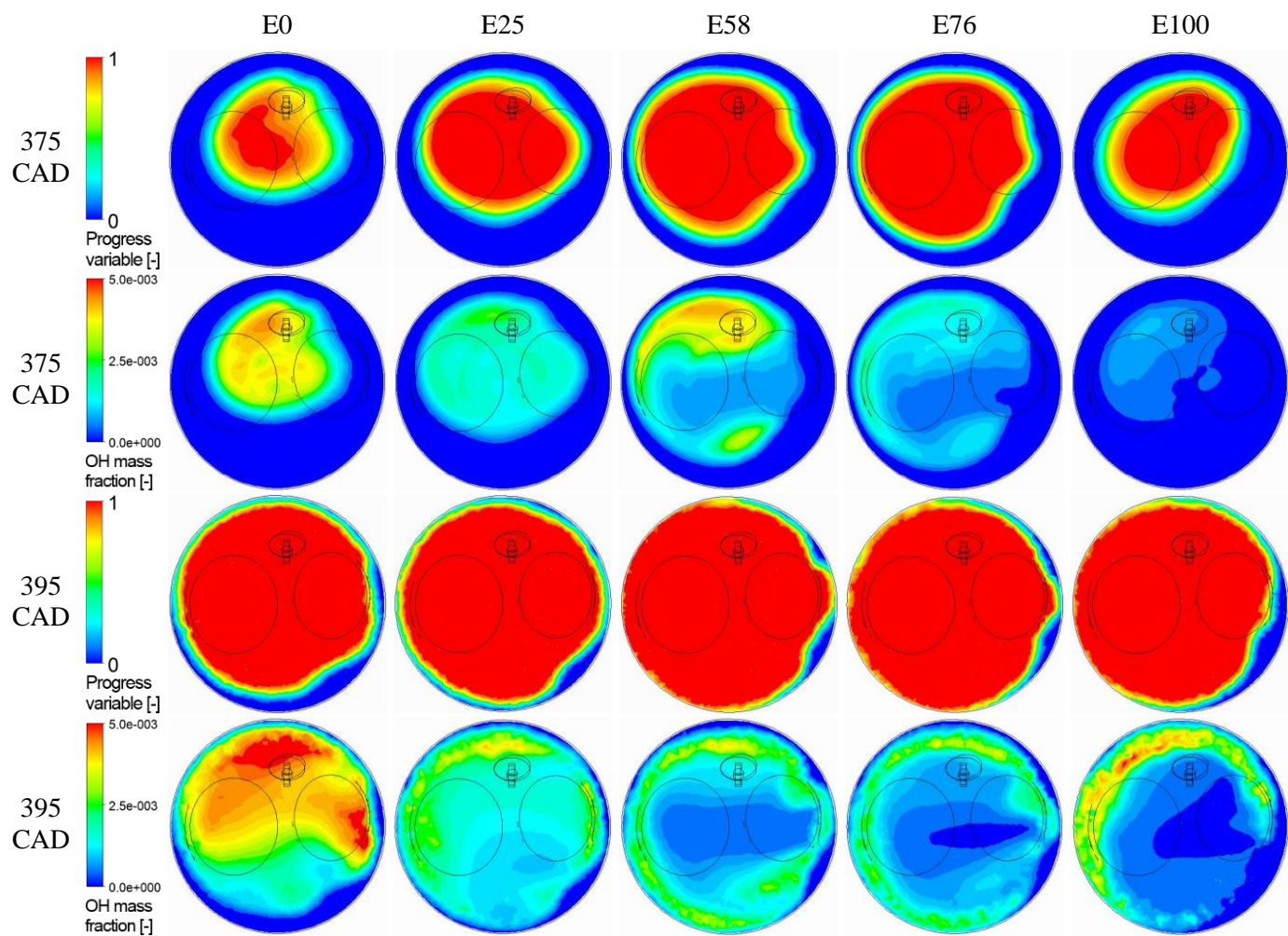
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715 Fig. 13. Flame propagation and distributions of OH mass fraction at 375 CAD and 395 CAD varying with the ethanol
 716 ratios. (2-column fitting image)

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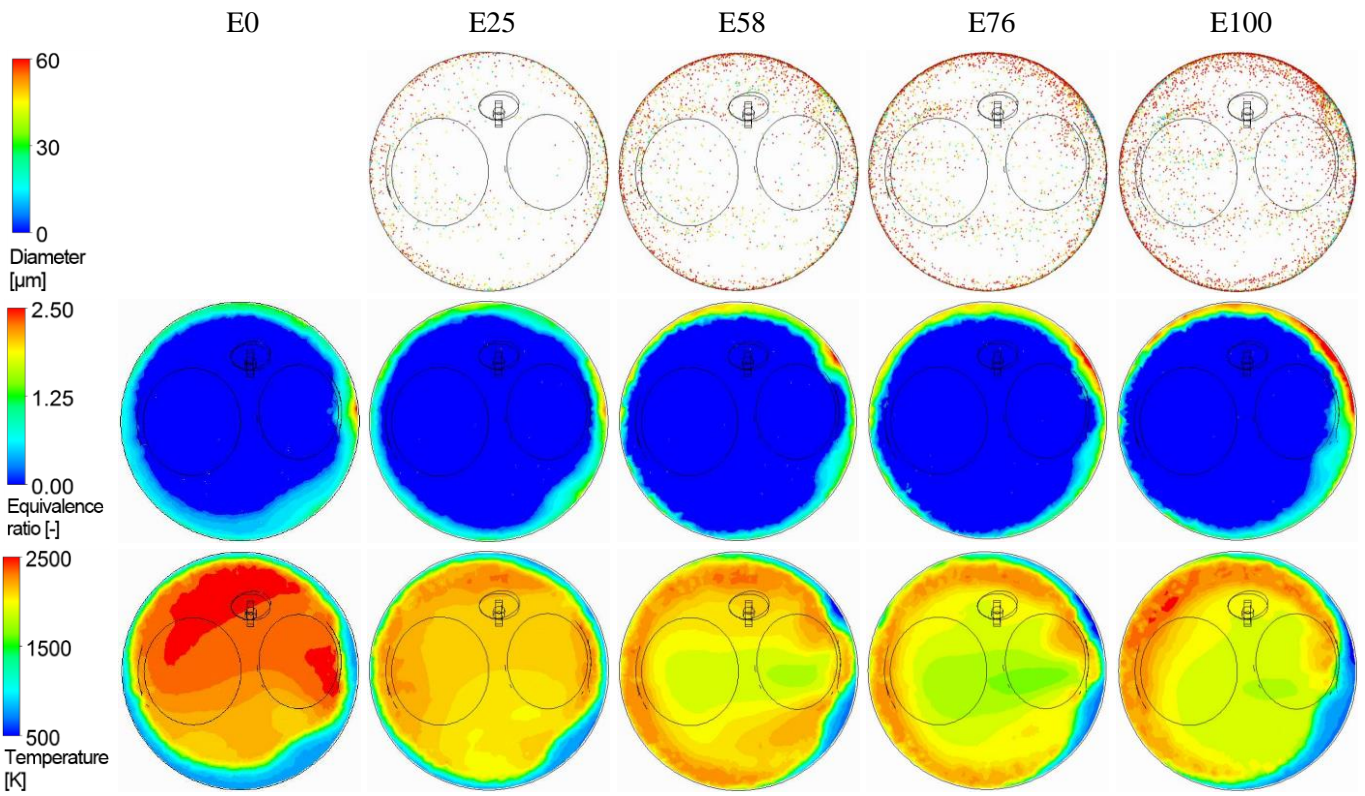
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727 Fig. 14. The distributions of ethanol liquid droplets, equivalence ratio and cylinder temperature at 395 CAD varying
 728 with the ethanol ratios. (2-column fitting image)

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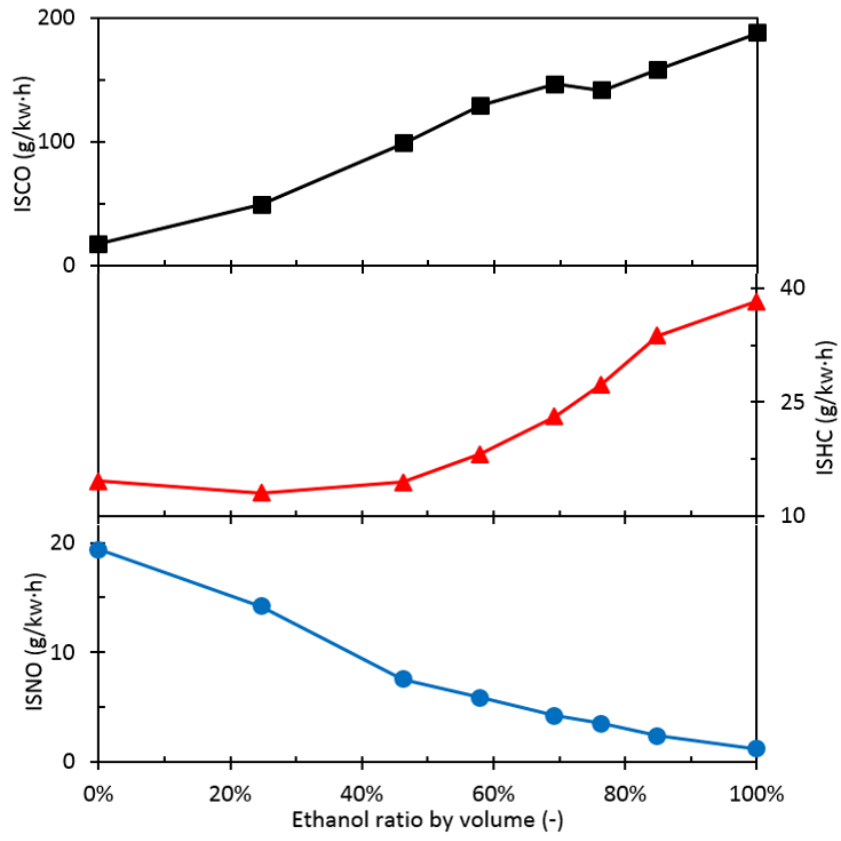
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Fig. 15. Measured engine emissions varying with the ethanol ratios. (single column fitting image)