Impact of gasoline and Diesel blends on combustion noise and pollutant emissions in Premixed Charge Compression Ignition engines

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Abstract

Research efforts in the automotive sector focus on developing new combustion concepts for mitigating the emissions of nitrous oxides and soot of conventional Diesel combustion. One of the most promising concept is the Premixed Charge Compression Ignition. In this, the fuel burns in premixed conditions, avoiding the formation of soot whereas nitrous oxides are controlled using large amounts of exhaust gas recirculation. Because of the premixed combustion, high fuel-burning velocities are produced, whence combustion noise is deteriorated. In order to mitigate this drawback, different blends of gasoline and Diesel fuels are being considered due to their suitability for this combustion characteristics. The effect of these fuel blends on emissions, performance and engine noise is analysed in this paper with the aim to provide additional knowledge of the fundamental issues of this particular combustion mode. The study also includes sweeps of both the start of injection and the amount of exhaust gas recirculation, in order to evaluate further degrees of freedom in the optimisation of the engine settings. Results show that the consideration of the engine noise together with both performance and emissions, reduces dramatically the margin of variation of the combustion settings, limiting therefore the operation range of the engine.

Keywords: PCCI combustion, Diesel engines, Combustion noise, Pollutant emissions, gasoline/Diesel blends

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1. Introduction

Improvements in performance, driveability and comfort, together with their low fuel consumption, have placed Diesel engines as the most widely used power-plant in both heavy and light duty vehicles [1]. Along the years, Diesel engines have been subjected to quite restrictive emissions regulations which have forced manufacturers to incorporate new solutions, thus achieving a high standing engine in comparison with the gasoline engine [2]. Despite being regarded as an environmentally friendly engine due to its low consumption and the consequent low carbon dioxide (CO_2) emission levels, the Diesel engine is not free from drawbacks. In particular, it exhibits a high level of pollutant emissions, most notably nitrous oxides (NO_x) and soot, mainly because of the poor control achievable on its combustion process. Currently, this is the greatest challenge of Diesel engines if it must comply with the emission standards in the near future. In order to face this difficulty, active and passive solutions have been explored in the last years. Among the active solutions, those advanced concepts promoting a low-temperature combustion have provided the most encouraging results for pollutant formation control [3].

Such a reduction in the combustion temperature can be achieved by means of either homogeneous or premixed charge compression ignition concepts (HCCI and PCCI) [4]. In these concepts, soot formation is controlled by promoting a premixed combustion while the NO_x can be simultaneously controlled by using a large amount of exhaust gas recirculation (EGR) [5]. In spite of such benefits, these new combustion concepts are limited to low load operation due to the complex combustion control required [6]. Additionally, as a consequence of the high rates of heat release produced, excessively high rates of pressure rise are expected and consequently unacceptable noise levels might be emitted by the engine [7].

The premixed combustion can be promoted by using simultaneously higher EGR rates and fuels with high volatility. Large amounts of EGR in the induced charge reduce the oxygen content inside the cylinder, so that the combustion efficiency can be deteriorated and hence both unburned hydrocarbon (UHC) and carbon monoxide (CO) emissions and fuel consumption can be increased [8]. In order to avoid the negative effect of using high EGR rates, fuels with a low cetane number (CN) and high volatility are a good alternative [9]. Since these fuels are more resistant to auto-ignition, a better fuel-air mixing rate can be achieved before the combustion starts [10]. This feature provides additional degrees of freedom when deciding the EGR rate that allows improving the trade-off between engine emissions and fuel economy [11].

In order to operate with premixed combustion, Diesel engines can be fueled with methanol, ethanol, natural gas, biogas, hydrogen or gasoline/Diesel (G/D) blends [12]. Some investigations have revealed that the use of gasoline in Diesel engines permits a better control of the combustion process that allows for the simultaneous reduction of NO_x and soot emissions [13] while torque values higher than those achievable with standard Diesel fuel are obtained. The high volatility of gasoline facilitates the fuel evaporation and improves the fuel-air mixing, so that a homogeneous charge formation is feasible with such a fuel [14]. Additionally, a longer ignition delay (ID) is expected with gasoline due to its low CN, allowing more time to increase the mixing quality before autoignition. However, since an excessive ID could result in an inefficient heat-to-work conversion process, the determination of the suitable proportion of gasoline in the blend is of crucial importance to avoid any penalties in engine performance [15].

Due to the low flammability of the gasoline when higher EGR rates are used, it is very difficult to promote the combustion at low loads, this drawback being magnified when the engine operates at cold conditions [16]. In those conditions, an increase of UHC and CO emissions and of the rate of heat release are expected [17]. The lower viscosity of G/D blends also affects the fuel injection system (FIS). In comparison with a standard FIS, a larger injection pump is required and higher leakages are also expected. Therefore, the fuel delivery must be reduced and consequently the engine power decreases [18].

In particular, PCCI combustion is intrinsically noisy precisely because of the premixed combustion phase, in which intense rates of pressure change are produced [19]. Engine noise is originated by the contribution of both pressure and mechanical forces produced during combustion which cause the vibration of the block. Pressure forces are caused by the abrupt rise of pressure occurring in the combustion chamber at the start of combustion, while mechanical forces are induced by combustion itself [20]. For this reason, at certain operating conditions both contributions are coupled [21]. In addition, the bowl geometry also plays an important role in engine noise control, since it has a noticeable influence on the development of resonant pressure fluctuations, which are induced by the ignition characteristics [22]. Regarding this, Diesel engine noise is highly dependent on any parameter that directly or indirectly affects the characteristics of the combustion process [23]. In particular, previous investigations have shown the sensitivity of

Diesel combustion noise to injection settings [24] and to the fuel used [25].

In this paper, the suitability of G/D blends for improving the trade-off between pollutant emissions, combustion noise and fuel economy of automotive Diesel engines operating under PCCI combustion concept is analyzed. With this aim, an experimental study on a light-duty DI Diesel engine operating at conditions for which such a concept is most suitable was performed. The effect on the engine parameters mentioned above of two fuel blends with different gasoline concentrations was evaluated by comparison with standard Diesel fuel. The metrics obtained from the decomposition of the in-cylinder pressure signal were used to evaluate both the objective [26] and subjective [27] aspects of combustion noise.

The methods and materials used in this study are presented in Section 2, where the suitability of the variables considered for achieving the objectives of the study is justified. In Section 3, the experimental configuration and the diagnostic technique used in order to predict the combustion noise are described. The results obtained are discussed in Section 4 in three separate subsections focusing on pollutant emissions, combustion noise and engine performance, respectively. In addition, the trade-off between these engine parameters is analyzed in Section 5. Finally, Section 6 summarizes the most relevant conclusions extracted from the work.

2. Materials and methods

A multi-cylinder compression-ignited (CI) automotive engine running at 1500 rpm and low load (38 Nm / 0.298 MPa of brake mean effective pressure –bmep) was used. This operating condition was chosen due to the following reasons:

1. The engine noise is unacceptable when it operates under conventional Diesel combustion.
2. It is a frequent condition (about 13% of total time) during a standard MVEG (Motor Vehicle Emissions Group) test cycle.
3. At this condition the PCCI combustion concept offers most of its potential.

Taking into consideration the results obtained by the authors in previous studies on PCCI combustion [19], the intake temperature was set to 45°C while the temperatures of the rest of the engine fluids were kept constant at the nominal values. All the tests were performed keeping constant the fuel quantity, considering a single injection of 10 mg/stroke at 80 MPa of injection pressure.

In order to analyse the impact of G/D blended fuels on PCCI combustion, three engine settings were chosen to define the test plan:

- Six values of injection timing advanced enough so as to run the engine under PCCI combustion but controlling the wall-impingement of the fuel in order to ensure that any lack of torque was not caused by this phenomenon.
- Three levels (10%, 12% and 14.5%) of oxygen concentration at the intake ($[O_2]_{IN}$) which could be used for PCCI combustion.
- Three different fuels: standard Diesel and two G/D blends. The percentages of volumetric concentrations of gasoline in the G/D blended fuels were 25% and 50%. According to other reported studies, these concentrations ensure suitable combustion stability when operating with the PCCI combustion concept [15]. A fuel lubricity additive (500 ppm) was included also in the G/D blended fuel in order to avoid any damages to the FIS.

The ranges of change of these variables allow to dispose a wide sample of PCCI combustion issues. The test matrix resulting from the above considerations is summarized in Table 1, while the main physical properties of the fuels used in the tests are given in Table 2.

After each test, the fuel line was emptied in order to avoid any effect from the fuel used in previous tests. The fuel tank was refilled with the other fuel, and then the engine was run under conventional Diesel combustion at high load as long as necessary before any new measurements were performed.

3. Experimental setup

The same experimental configuration used by the authors in previous studies was used also in this investigation [19, 24]. The tests were performed on a test bench composed of an 1.6 l four-cylinder Euro IV turbocharged direct injection (DI) Diesel engine directly coupled to an asynchronous dynamometer. The production EGR system of the engine was modified in order to achieve the temperature and oxygen concentrations in the intake required to run the engine under PCCI combustion. With this aim an almost 40% larger EGR cooler was used together with an external cooling system, so that an accurate control of intake temperature was ensured. The main specifications of the engine and the injector are summarized in Table 3.

Even though the engine noise was predicted by means of the procedure presented in Section 4, engine noise measurements in free field conditions were also performed in order to check, at any time of the work, the reliability of the procedure used for predicting the noise level.

With the purpose of controlling and characterizing the operation of the engine the following instrumentation were used:

- K type thermocouples for measuring the temperature of all the engine fluids.
- A HORIBA MEXA-720 portable analyzer was used to measure the NO_x emissions, the O_2 concentration in the exhaust, the equivalence ratio and the excess air ratio (λ).
- The O_2 concentration in the intake air was measured with a lambda probe placed in the intake manifold.
- An AVL 451S filter-type smoke meter was used for measuring the filter smoke number (FSN). The correlation proposed by Christian et al. [28] was used to estimate the soot emission from FSN measurements.
- Kistler 6055Bsp glow-plug piezoelectric sensors were used to measure the pressure trace in each cylinder with a sampling frequency of 50 kHz, so that a bandwidth similar to the human domain of hearing (20 Hz - 20 kHz) was available.

The accuracy of the instrumentation used is summarized in Table 4. In each test, in-cylinder pressure traces of 50 consecutive cycles were recorded. In order to modify the engine settings accordingly to the method described above, an open ECU (Engine Control Unit) was used. Additional details of the experimental configuration can be found in [19].

4. Combustion noise assessment

As in previous studies, the objective and subjective aspects of combustion noise were characterized making use of the predictive procedures developed by Torregrosa et al. [26] and Payri et al. [27], respectively. Both procedures are based on the identification of cause-effect relations between the source and the radiated noise. With this approach, both the overall noise (ON) and its sound quality, which is quantified by a mark, are predicted from the in-cylinder pressure decomposition [21].

In these procedures, the in-cylinder pressure signal is decomposed into three sub-signals that represent the main phenomena which characterize the combustion process. These sub-signals are:

- The compression-expansion sub-signal, that represents the increment and the reduction of in-cylinder pressure due to the volume variation experienced by the charge as a result of the piston motion.
- The combustion sub-signal, that characterizes the combustion process. The temporal evolution of the in-cylinder pressure during combustion is strongly influenced by the rate of heat release which, in turn, is affected by the injection settings.
- The resonance sub-signal, that characterizes the oscillations of the burned gas inside the combustion chamber caused by the abrupt pressure rise rates [29].

The overall level and the sound quality of combustion noise were predicted by empirical correlations containing operation and combustion indicators. Torregrosa et al. [26] found that the overall noise is highly correlated with an operation indicator, I_n, that quantifies the contribution of engine speed to noise, and two combustion indicators, I_1 and I_2, which are the contributions of the fuel burning velocity and of the resonance inside the combustion chamber [22], respectively. The correlation for the overall level is represented by the following expression:

$$\mathrm{ON} = C_0 + C_n I_n + C_1 I_1 + C_2 I_2 + \epsilon_{\mathrm{ON}} \tag{1}$$

In this equation, C_i are coefficients dependent on the engine family and size, ϵ is the error and the indicators are defined by the following equations:

$$I_n = \log\left[\frac{n}{n_{\mathrm{idle}}}\right] \tag{2}$$

$$I_1 = \frac{n}{n_{\mathrm{idle}}}\left[\frac{(\mathrm{d}p/\mathrm{d}t)^{\mathrm{max1}}_{\mathrm{comb}} + (\mathrm{d}p/\mathrm{d}t)^{\mathrm{max2}}_{\mathrm{comb}}}{(\mathrm{d}p/\mathrm{d}t)^{\mathrm{max}}_{\mathrm{comp}}}\right] \tag{3}$$

$$I_2 = \log\left[E_0\frac{E_{\mathrm{res}}}{E_{\mathrm{comp}}}\right] \tag{4}$$

where n and n_{idle} are the actual and idle speed, respectively; $(\mathrm{d}p/\mathrm{d}t)^{\mathrm{max1}}_{\mathrm{comb}}$ and $(\mathrm{d}p/\mathrm{d}t)^{\mathrm{max2}}_{\mathrm{comb}}$ represent the two highest peaks of the pressure derivative during combustion, $(\mathrm{d}p/\mathrm{d}t)^{\mathrm{max}}_{\mathrm{comp}}$ is the peak value of the pressure derivative corresponding to the compression-expansion component, E_0 is a scaling factor, E_{res} is the signal energy of the resonance and E_{comp} is the signal energy of the compression-expansion signal.

In addition, Payri et al. [27] verified that the combustion indicators defined above were highly correlated also with the sound quality of combustion noise. Sound quality is quantified by a mark ranging from 0 to 10 which represents the satisfaction degree of an average customer. The correlation proposed by Payri et al. is the following:

$$\mathrm{Mark} = 10 - c_1 I_1 - c_2 I_2 + \epsilon_{\mathrm{Mark}} \tag{5}$$

Here c_i coefficients are also characteristic of the engine family and size, but their values are different from those of Equation (1).

5. Results and discussion

In this section the results in terms of pollutant emissions, combustion noise and engine performance will be discussed in detail. With this purpose, the results obtained with the Diesel standard fuel and the G/D blends under PCCI combustion will be compared with those obtained under conventional Diesel combustion. Then, the reference NO_x and soot emission levels are 80 ppm and 20 mg particles/m^3 respectively, while the bmep is 0.298 MPa. Regarding combustion noise, a mark equal to 7 was considered as the acceptable comfort threshold of an average customer [27].

5.1. Pollutant emissions and engine performance

The analysis of pollutant emissions was focused on NO_x and soot levels, since the main advantage provided by the PCCI combustion concept is precisely the reduction of these pollutants.

Figure 1 shows the trends followed by NO_x and soot emissions. In general, the best NO_x-Soot trade-offs are consistently obtained at advanced SOE (start of energizing), where the combustion switches from the conventional Diesel combustion concept to the PCCI combustion concept, thus confirming its potential to control both pollutants simultaneously in production engines. However, moderate-to-high levels of EGR are mandatory to control mixture reactivity and NO_x emissions due to the local mixture heterogeneities with zones in stoichiometric conditions. In these zones the maximum combustion temperatures are still high despite fuel premixing. This is implicitly corroborated by the reduction in NO_x attainable by advancing the SOE, which extends the ignition delay (ID) as shown in Fig. 2, so there is more time available for decreasing the local air/fuel ratios before the onset of the combustion process [30]. The impact of G/D blends on NO_x emissions is moderate, since the adiabatic flame temperatures of gasoline and Diesel are quite similar, and thus the only benefit is a consequence of the longer ID observed in Fig. 2 as the gasoline fuel fraction increases [31]. This retards the onset of combustion thus producing lower maximum pressures and temperatures during the combustion process.

The soot emission trends observed in Fig. 1 corroborate the transition from conventional Diesel combustion to PCCI combustion as SOE is advanced. Particularly for medium-to-low $[O_2]_{IN}$ soot emissions initially increase but if SOE is further advanced they sharply decrease down to very low levels. This sharp soot emissions reduction at early SOE is caused by the longer ID and mixing time availability, resulting in lower local equivalence ratios below the soot formation limit, which is known to be around 2.5 to 2 [32]. Figure 1 also confirms the positive effect of increasing the gasoline proportion on soot emissions, which is directly related to its impact on ID and mixing time shown in Fig. 2. The physical and chemical properties of G/D blends, such as low initial boiling point and high octane number [33], contribute to the extension of the premixed combustion phase, due to a higher ID [34]. A parameter that allows to justify the observed reduction in soot emissions is the ratio between ID and the injection time (IT), which describes the mixing time availability in relation to the injection duration, so that the higher this parameter is, the lower are the maximum local equivalence ratio and consequently the lower the soot formation. As observed in Fig. 2, this parameter increases by decreasing $[O_2]_{IN}$ (lowest effect), advancing the SOE and also increasing the gasoline fuel fraction. Despite that the trigger level of ID/IT required to operate in PCCI combustion conditions depends on different factors, including the engine hardware configuration, according to the results obtained its value should be well above 3, which gives a first idea about the suitable SOE range for implementing this advanced combustion concept.

As a final remark, it has been proven that blending gasoline fuel with the conventional Diesel fuel in compression ignition engines is an attractive alternative to make the implementation of the PCCI combustion concept easier, since these blends require less EGR and SOE advance to achieve extremely low NO_x and soot emissions.

5.2. *Combustion noise*

Figure 3 shows the results corresponding to the noise characterization of the engine at all the conditions considered. In a first view, the results evidence that the overall noise is inversely proportional to the mark that represent its sound quality. In addition, it is observed that the noise is increased as the injection timing is delayed and, as expected, there is a great impact of the oxygen concentration in the intake on the noise issues. For all the injection timings and fuels tested, combustion noise is improved when the oxygen concentration is reduced.

Regarding the subjective perception of engine noise, Fig. 3b, 3d and 3f show that the level of acceptance can be surpassed when the oxygen concentration is lower than 12% and for a narrow range of advanced injections, which depends on the fuel. Since advanced injections could lead to a penalty in the bmep, these conditions will be the object of a thorough discussion in the following sections.

Concerning the effect of the fuel on the combustion noise, the results are not conclusive altogether, except for the mark estimated for the lowest $[O_2]_{IN}$ considered, for which it is clear that the sound quality of engine noise is enhanced when the G50 blend of fuel is used.

The estimated values of the combustion indicators which characterize the combustion noise source are shown in Fig. 4. The plots show that the indicator I_1, that is related to the fuel burning velocity, increases if either the oxygen concentration is increased or the injection is delayed. In the same way, the energy of the resonance (characterized by indicator I_2) increases when $[O_2]_{IN}$ also increases. Fig. 4b shows that, with the highest $[O_2]_{IN}$, the energy of the resonance is scarcely sensitive to the injection timing and therefore, taking into account the results of Fig. 3a and 4a, one can conclude that the indicator I_1 strongly contributes to the engine noise. Moreover, Fig. 4d and 4f show that I_2 increases as the injection timing is delayed when EGR is used and consequently the oxygen concentration is reduced.

In the next sub-sections a detailed analysis of the sensitivity of combustion noise to the combustion parameters and fuels considered in the investigation is presented.

5.2.1. *Sensitivity to start of injection*

Figure 5 shows the in-cylinder pressure trace and the evolution of the pressure derivative of the combustion signal –which is related to the rate of fuel burning– obtained for different injection timings when the engine operates with an $[O_2]_{IN}$ of 10% and is fueled with standard Diesel. In-cylinder pressure traces of Fig. 4f and 5a show that the amplitude of the high-frequency pressure oscillations is increased as the injection timing is delayed. The increase of the amplitude of this oscillations is mainly caused by the increase of the pressure rise rate, which is associated with the fuel burning velocity. Indeed, Fig. 5b shows that the pressure derivative peaks increase as the injection is delayed. Due to the contribution of both effects, the increase of the energy of resonance and the rate of fuel burned, the combustion noise is deteriorated as shown in Fig. 3f.

5.2.2. Sensitivity to oxygen concentration in the intake

Previous results have shown that both pollutant emissions and combustion noise are strongly affected by the oxygen concentration in the intake. Reducing $[O_2]_{IN}$, a longer ignition delay can be achieved and consequently the combustion is shifted towards the top dead center. In this situation, the fuel burning velocity is reduced, and therefore both the rate of pressure change and the intensity of the resonance in the combustion chamber are reduced. This fact is evidenced by the results shown in Fig. 6, where the in-cylinder pressure traces and the pressure derivative of the combustion signals for the engine operating with an injection timing of -30 crank angle degree after Top Dead Center (cad aTDC), standard Diesel fuel and three oxygen concentrations are depicted. A less abrupt combustion is promoted as the concentration is reduced and the sound quality of the combustion noise is thus enhanced.

5.2.3. Sensitivity to the fuel

The results shown in Fig. 3 evidence that the effect on combustion noise of the fuel burned starts to be apparent for low values of oxygen concentration in the intake. In particular, the most relevant differences were observed when this concentration is as low as 10%. This effect is made clear in the plots in Fig. 7, which show the pressure and pressure derivative evolutions inside the cylinder for an SOE of -34 cad aTDC and the three fuels considered. These plots clearly show that the peak value of the rate of burned fuel decreases as the volumetric concentration of gasoline in the G/D blend of fuel increases, whence less knock can be noticed in the in-cylinder pressure traces. As in the previous analysis, these conditions lead to a smoother combustion process with which a better subjective perception of the noise emitted by the engine can be achieved.

Up to this point, the results lead to conclude that the best strategies in order to improve the PCCI combustion in terms of pollutant emissions reduction and noise improvement is obtained by employing 10% of $[O_2]_{IN}$ and G50 blend. In the following section, the performance of the engine using such strategies will be analyzed in detail.

5.3. Engine performance

With respect to engine performance, Fig. 8 confirms that the bmep progressively decreases when advancing SOE for all fuel blends; however, the combustion phasing traced by the crank angle degree at which 50% of the total heat release occurs (CA50) does not follow a clear pattern. Thus, the negative impact of advancing SOE on bmep is expected to be basically caused by the higher CO and UHC emissions resulting from increasing the over-lean regions and liquid fuel impingement onto the combustion chamber walls.

An interesting trend is observed by comparing the results using gasoline/Diesel blends with those obtained with standard Diesel fuel. The increment seen in bmep is caused mainly by the better combustion phasing, as it is significantly retarded as a result of the decreased reactivity provided by the blended fuel. In addition, the longer ignition delay and mixing time enhances the homogeneity of the mixture, which has also a positive impact on bmep [35]. Another aspect helping to increase bmep is the less liquid fuel wall impingement expected for the blended fuel as a result of the higher volatility of gasoline compared to Diesel fuel, which would result in lower UHC emissions.

6. Assessment of trade-offs between parameters

A global analysis of the results previously reported is performed in this section in order to identify all the relevant tendencies, by using the contour plots shown in Fig. 9 with injection timing and $[O_2]_{IN}$ as axes.

The left column shows the contour corresponding to NO_x and soot levels, where the white zone represents the range in which any of the two pollutants is above the target limits. According to these results, the optimum zone (green) from the point of view of pollutant emissions is extended using gasoline/Diesel blends, especially at low $[O_2]_{IN}$ conditions.

Subsequently, if pollutant emissions results are combined with engine bmep results, as show in the center column, the best zones for all fuels are those where $[O_2]_{IN}$ is lower than 12% and SOE ranges between -30 and -18 cad aTDC. In addition, the range with suitable bmep levels is extended when using gasoline/Diesel blends.

The right column confirms that including the combustion noise target reduces strongly the optimum range of operation. Contrarily to the bmep, for which the optimum range was located towards the right of the contour map, the combustion noise pattern is completely opposite. The sound quality improves for very advanced SOE and $[O_2]_{IN}$ below to 11%, that is, the optimal range is found towards the bottom-left corner of the map. For Diesel and G25 gasoline/Diesel blend fuels, no intersection is observed between the acceptable sound quality target –represented by a mark of 7– and the bmep target. In fact, a very limited region fulfilling all targets is observed only for the G50 gasoline/Diesel blend when the SOE is approximately -30 cad aTDC and $[O_2]_{IN}$ is 10%.

These results confirm that the trade-off between combustion noise and engine bmep is a key restrictive factor for the PCCI combustion concept, in which low NO_x and soot levels are produced. Additionally, it has been proven that the use of gasoline/Diesel blends leads to a wider operation range where the PCCI combustion concept provides promising results not only concerning pollutant emissions, but also considering thermal efficiency and combustion noise.

7. Conclusions

There is no doubt about the great potential of the PCCI combustion concept in order to reduce NO_x and particulate emissions of Diesel engines. However, these benefits are counteracted by the deterioration of engine noise and even of performance. In this work, the suitability of gasoline/Diesel blends for addressing these issues was analyzed. The use of such blends reduces liquid impingement onto the combustion chamber walls, enhances the air-fuel mixing process and provides a better combustion phasing, so that engine performance improves when compared to the results obtained using standard Diesel fuel. Additionally, for a given oxygen concentration at the intake, the lower cetane number of these blends permits extending the ID, so that a less abrupt combustion is promoted and consequently the engine noise level decreases. Therefore, these properties allow for considering engine settings keeping the EGR rate at levels attainable with current production EGR systems.

Regarding pollutant emissions, the results show that keeping constant both the injected fuel mass and the injection pressure, NO_x formation is mainly affected by the oxygen concentration of the charge, while soot formation is clearly dominated by the volumetric concentration of gasoline in the blend. Moreover, independently of the SOE, the best NO_x-soot trade-off was obtained with $[O_2]_{IN}$ of 10% and a gasoline concentration of 50% in the fuel blend.

The results also show that combustion noise is more sensitive to $[O_2]_{IN}$ variations than to the fuel used. In addition, it appears that there exists a threshold level of $[O_2]_{IN}$, between 12% and 10%, above which the contribution of the gasoline content in the blend to the improvement of the sound quality of combustion noise is more apparent. Again, the noise comfort limit (Mark=7) is surpassed when the most advanced injection settings (between -38 and -30 cad aTDC) are used together with the lowest oxygen concentration of the charge at the intake ($[O_2]_{IN}$=10%). However, a loss of bmep has been observed at these advanced injection conditions, even though this effect can be partly tackled by using gasoline/Diesel blends as fuel.

In summary, this investigation confirms that gasoline/Diesel blends are suitable fuels in order to preserve the potential of PCCI combustion for reducing pollutant emissions by controlling its negative impact on engine efficiency. Nevertheless, the consideration of engine noise together with such emissions-performance trade-off, reduces dramatically the degrees of freedom for the optimization of engine settings, since the optimal solution appears to be a single combination of $[O_2]_{IN}$, SOE and fuel composition.

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Figure captions

Fig. 1 Effect of injection timing, oxygen concentration and fuel on pollutant emissions: NO_x (left column) and soot (right column).

Fig. 2 Effect of injection timing, oxygen concentration and fuel on: ID (left column) and ID/IT (right column).

Fig. 3 Effect of injection timing, oxygen concentration and fuel on engine noise: ON (left column) and Mark (right column).

Fig. 4 Effect of injection timing, oxygen concentration and fuel on combustion indicators: I_1 (left column) and I_2 (right column).

Fig. 5 Effect of injection timing on in-cylinder pressure (a) and pressure derivative of the combustion signal (b) for Diesel standard and $[O_2]_{IN}$ of 10%.

Fig. 6 Effect of the $[O_2]_{IN}$ on in-cylinder pressure (a) and pressure derivative of the combustion signal (b) for Diesel standard and SOE of -30 cad aTDC.

Fig. 7 Effect of fuel on in-cylinder pressure (a) and pressure derivative of the combustion signal (b) for $[O_2]_{IN}$ of 10% and SOE of -34 cad aTDC.

Fig. 8 Effect of injection timing and fuel to bmep (a) and CA50 (b) for $[O_2]_{IN}$ of 10%.

Fig. 9 Trade-off maps of pollutant emissions (left column), bmep (center column) and mark (right column) achieved at PCCI combustion with the three types of fuel used: D (top row), G25 (center row) and G50 (bottom row).

Table 1: Test matrix considered in the study.

Injection pressure (MPa)	$[O_2]_{IN}$ (%)	SOE (Deg)	Fuel (in volumetric concentrations)
80	10	-38 to -18 cad aTDC each 4	D:100% Diesel
	12		G25: 25% gasoline / 75% D
	14.5		G50: 50% gasoline / 50% D

Table 2: Properties of fuels used.

Fuel (in volumetric concentrations)	Density 15°C (kg/m^3)	Cetane index	Viscosity 40°C (cSt)	Heating value ($MJ\ kg^{-1}$)
D	839.3	51.2	2.676	42.900
G25	814.3	36.6	1.525	43.060
G50	777.5	28.3	1.031	43.250

Table 3: Engine and injector specifications.

Engine Type		Direct-injection Diesel engine
Cylinders		4 in line
Bore	(mm)	75
Stoke	(mm)	88.3
Compression ratio		18:1
Injector nozzle holes		6
Nozzle holes diameter	(mm)	0.124
Spray angle	(deg)	150

Table 4: Accuracy of the instrumentation used.

Sensor	Variable	Accuracy [%]
Piezoelectric	In-cylinder pressure	0.4
Thermocouples	Temperature of all fluids	0.35
Encoder	Engine speed	0.006
Exhaust gas analyzer	NO_x emissions and O_2 concentration in the exhaust	2
Lambda probe	O_2 concentration in the intake	1.81
Smoke meter	FSN	3
Piezoresistive	Intake and exhaust pressure	0.65
Torque meter	Torque	0.1
Fuel mass flow meter	Fuel mass	0.2
Air mass flow meter	Air mass	0.12

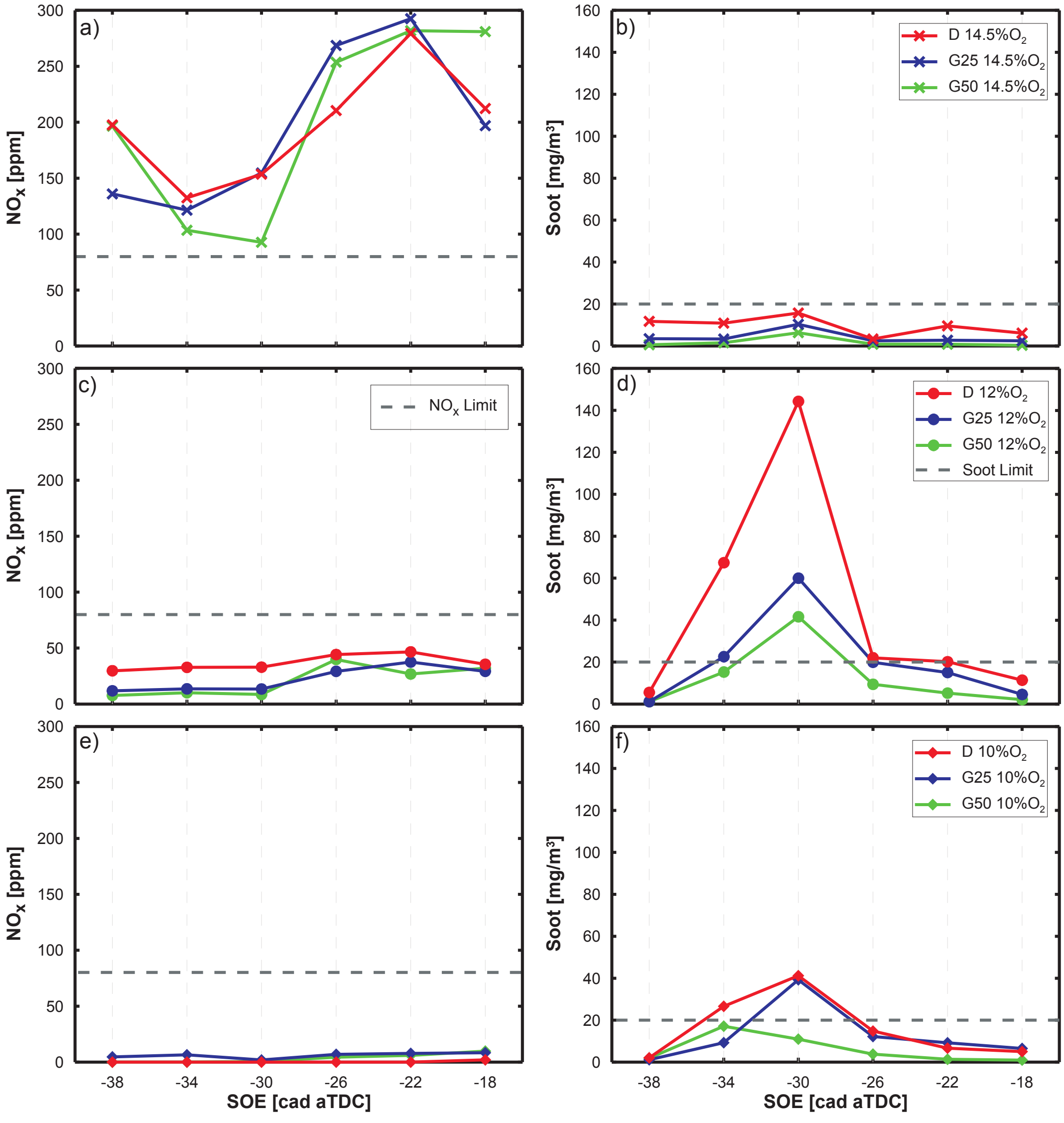

Figure 1: Effect of injection timing, oxygen concentration and fuel on pollutant emissions: NO_x (left column) and soot (right column).

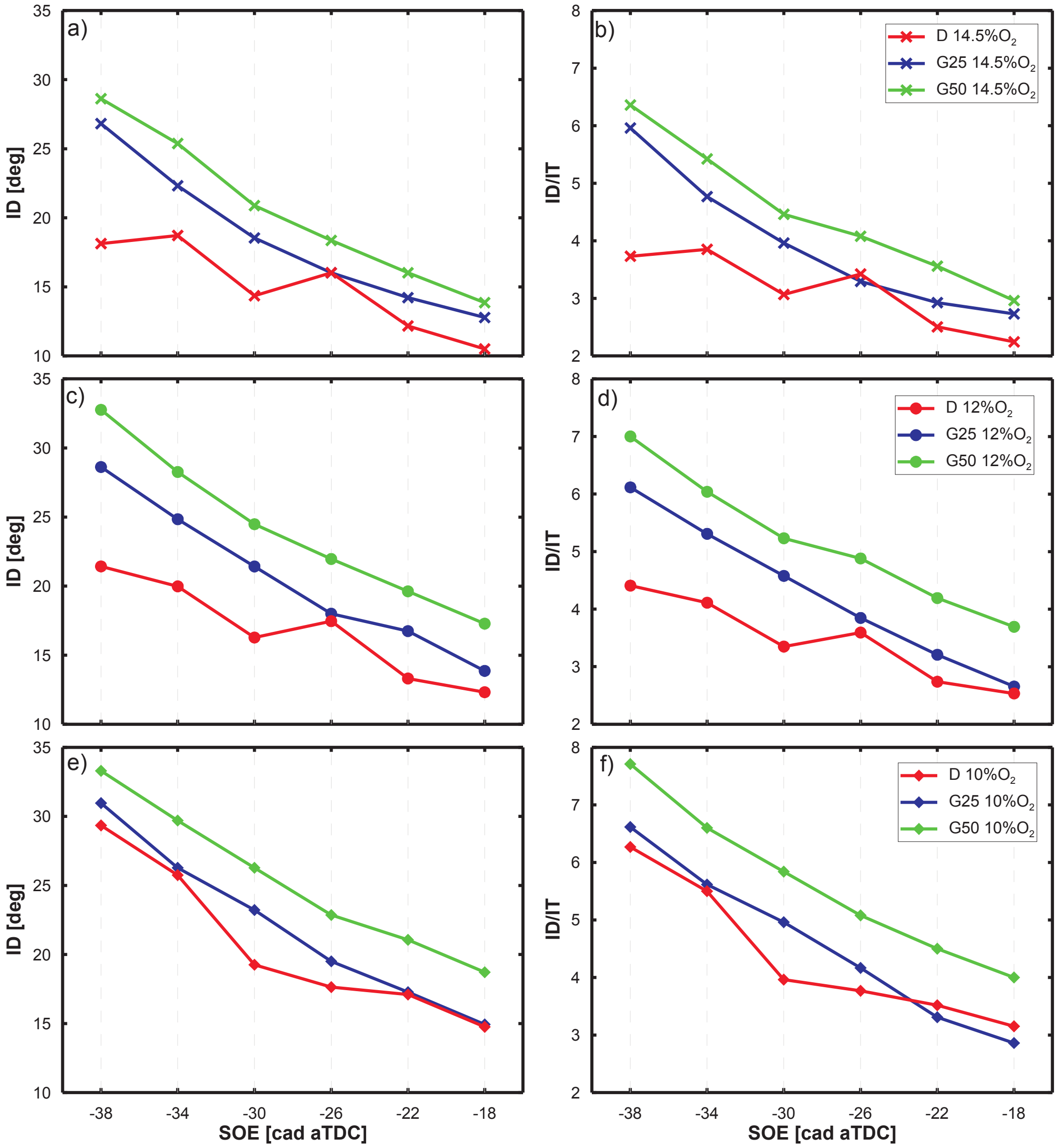

Figure 2: Effect of injection timing, oxygen concentration and fuel on: ID (left column) and ID/IT (right column).

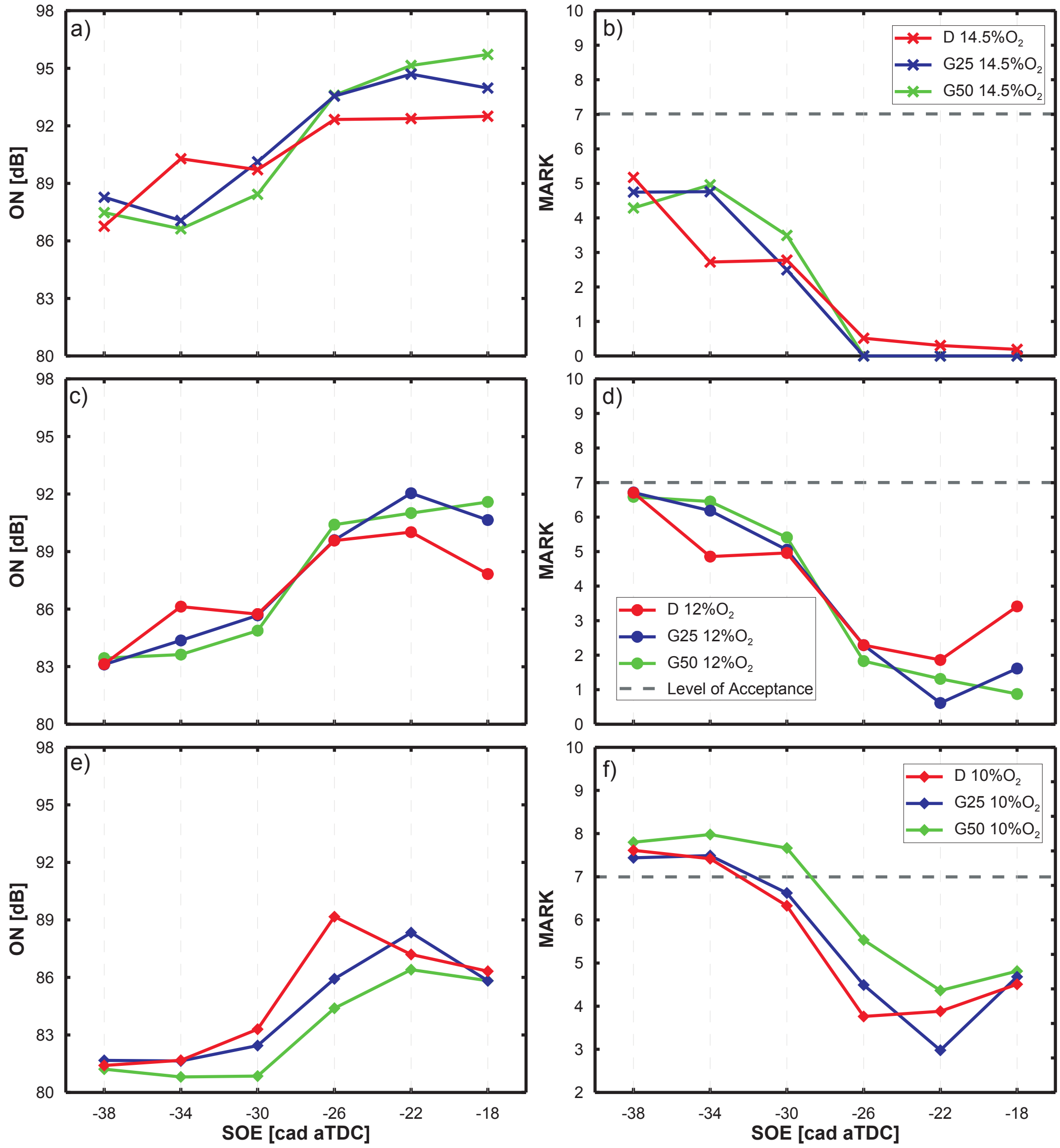

Figure 3: Effect of injection timing, oxygen concentration and fuel on engine noise: ON (left column) and Mark (right column).

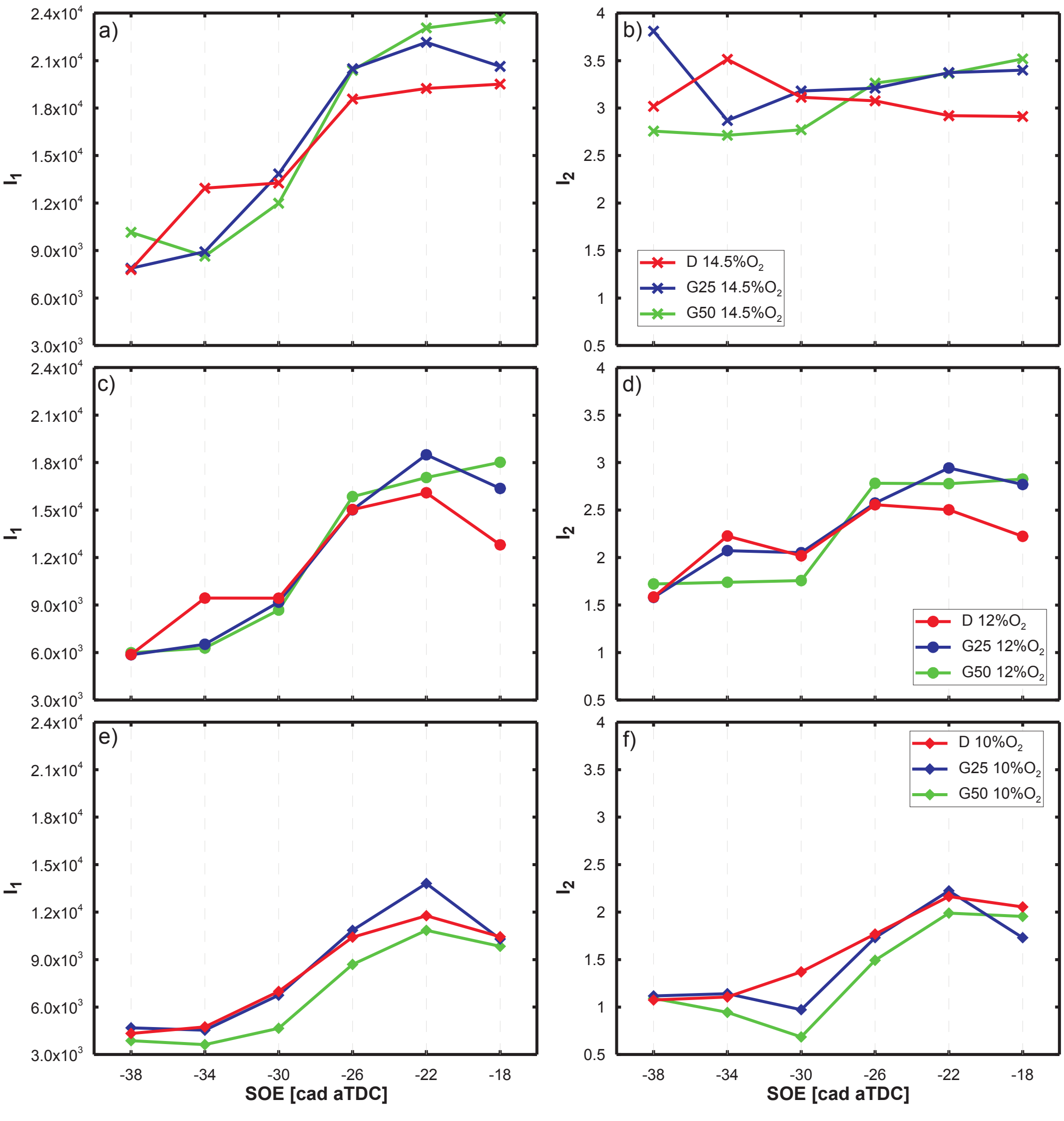

Figure 4: Effect of injection timing, oxygen concentration and fuel on combustion indicators: I_1 (left column) and I_2 (right column).

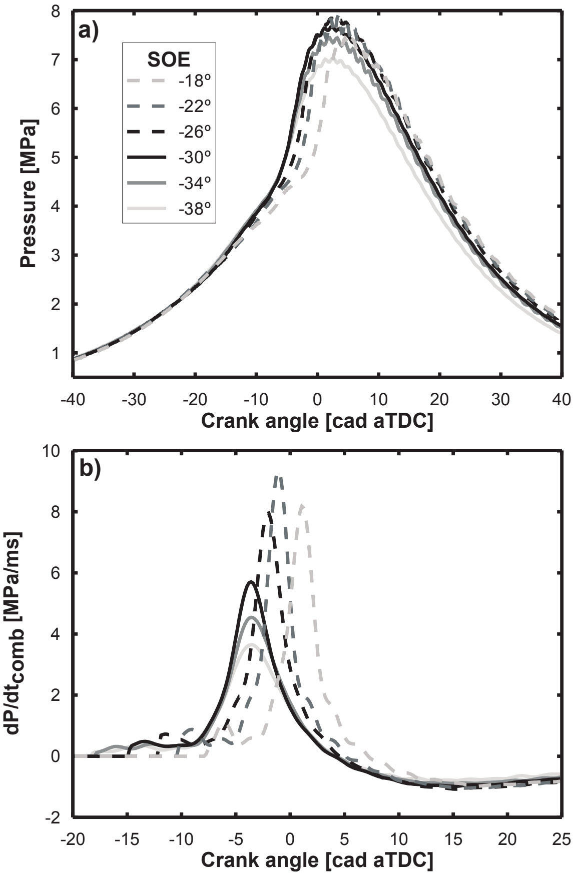

Figure 5: Effect of injection timing on in-cylinder pressure (a) and pressure derivative of the combustion signal (b) for Diesel standard and $[O_2]_{IN}$ of 10%.

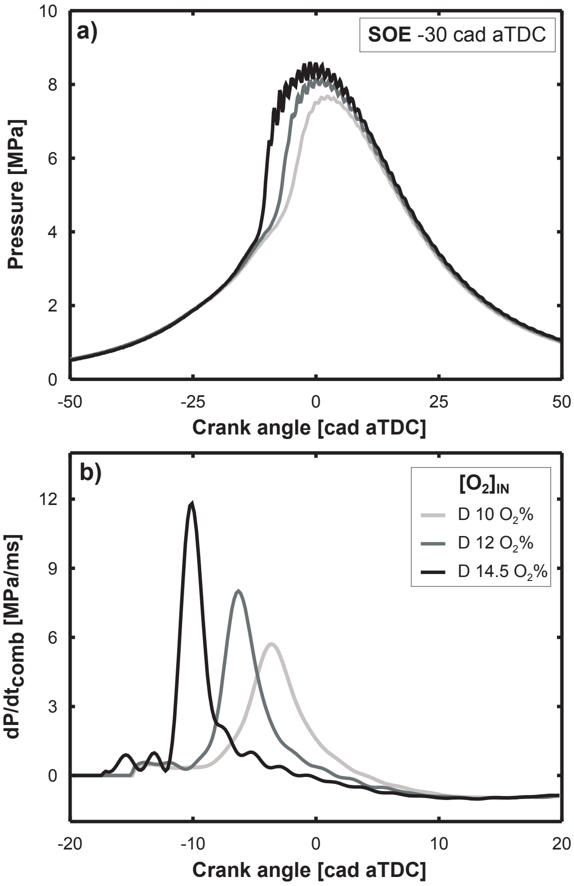

Figure 6: Effect of the $[O_2]_{IN}$ on in-cylinder pressure (a) and pressure derivative of the combustion signal (b) for Diesel standard and SOE of -30 cad aTDC.

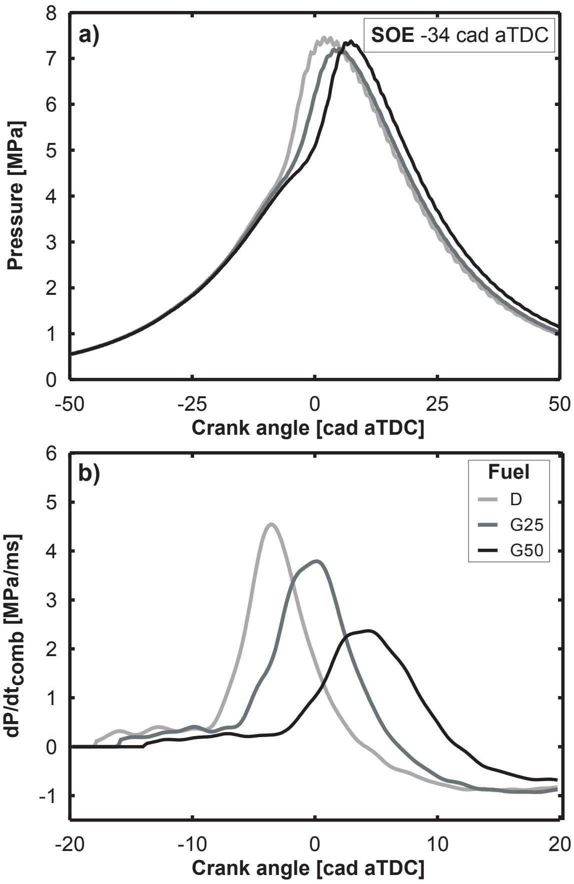

Figure 7: Effect of different fuels on: in-cylinder pressure (a) and pressure derivative of the combustion signal (b) for $[O_2]_{IN}$ of 10% and SOE of -34° cad aTDC.

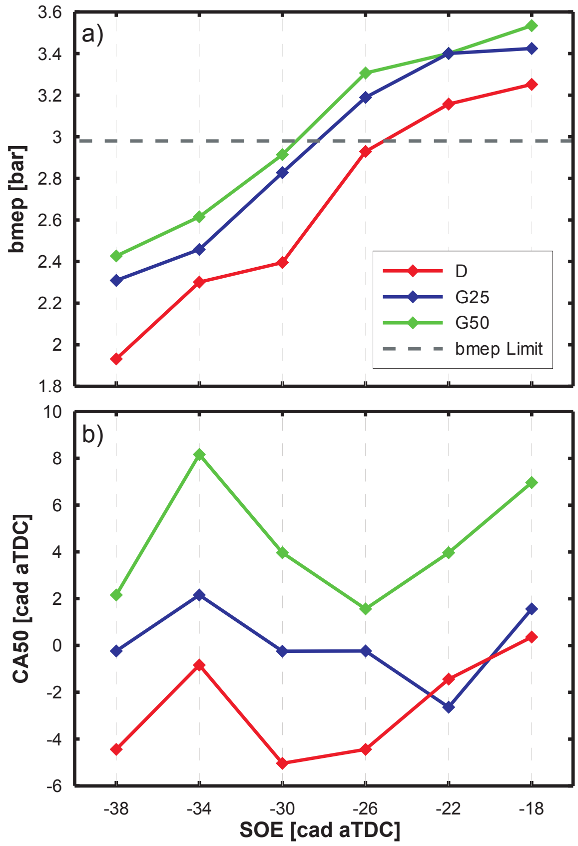

Figure 8: Effect of injection timing and fuel on bmep (a) and CA50 (b) for $[O_2]_{IN}$ of 10%.

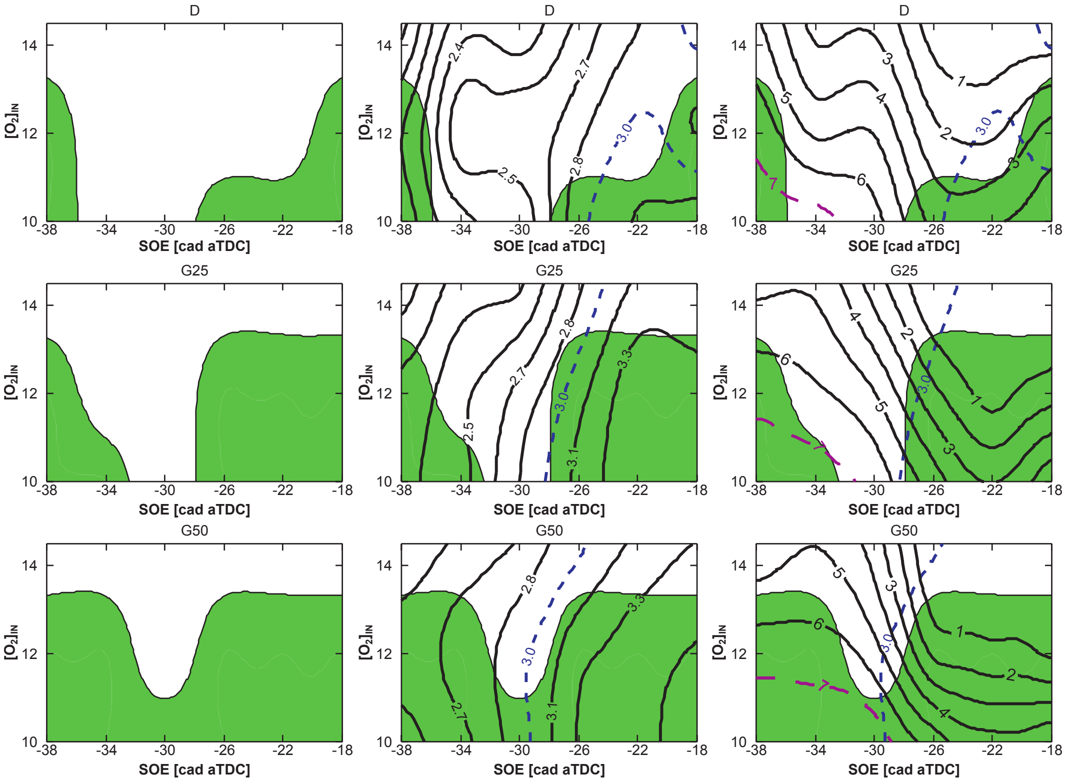

Figure 9: Trade-off maps of pollutant emissions (left column), bmep (center column) and mark (right column) achieved at PCCI combustion with the three types of fuel used: D (top row), G25 (center row) and G50 (bottom row).