Clean and efficient dual-fuel combustion using OMEx as high reactivity fuel: comparison to diesel-gasoline calibration

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Abstract

From previous results in a single-cylinder engine platform, it can be concluded that the dual-mode dual-fuel (DMDF) concept can be a potential solution to overcome the major constraints found with other single-fuel low temperature combustion modes. To extend these findings to a real application, this work evaluates the potential of the diesel-gasoline DMDF concept on a multi-cylinder 8L engine in terms of performance and emissions. To do this, a full engine calibration map was obtained following a specific methodology. The emissions results show that diesel-gasoline DMDF allows to achieve EURO VI NOx and soot emissions in a great portion of the engine map. Nonetheless, the levels of these pollutant at high load conditions exceed the EURO VI limits by far due to the need of implementing a diffusive combustion strategy with high EGR levels to avoid excessive in-cylinder pressure gradients. To mitigate this issue, the use of Oxymethylene ether (OMEx) instead of diesel fuel is proposed. A dedicated engine calibration was developed for the OMEx-gasoline DMDF concept following the same methodology. The results show that the oxygen content in the OMEx molecule allows to achieve a fully EUVI compliant engine calibration in terms of NOx with engine-out soot levels lower than 0.01 g/kWh. Moreover, due to the lower stoichiometric air-fuel ratio with this fuel, the air management system requirements are lower, reducing the pumping losses and increasing the brake thermal efficiency in most of the calibration map.

Keywords

Reactivity controlled compression ignition; Dual-fuel combustion; Oximethylene ether; EURO VI emissions; Synthetic fuels

1. **Introduction**

Recent reports estimated that the transportation sector is responsible for more than 20% of the total CO_2 emitted to the atmosphere [1][2]. Inside this sector, the medium- and heavy-duty applications are the second most important CO_2 source after the light-duty vehicles. In particular, they represent around 28% of the total CO_2 share in the current transportation scenario [3]. Moreover, these applications are also responsible for other emissions released to the environment as soot and nitrogen oxides (NOx) [4][5]. To limit the pollutant levels emitted by the internal combustion engines to the atmosphere, strict regulations were created [6]. To achieve the emissions standards imposed by the normative, different devices are used [7]. In this sense, a conventional medium- and heavy-duty powertrain consists of the internal combustion engine (power unit) and a complex after treatment system (ATS) to reduce the impact of the exhaust gases on the environment [8][9]. This system is composed of a diesel oxidation catalyst (DOC) that converts the unburned hydrocarbons to CO_2 [10][11], a diesel particle filter (DPF) that traps the soot particles, oxidizing them in a regeneration step [12][13], and finally a selective catalytic reduction system (SCR) that converts the NOx formed during the combustion to inert nitrogen [14]. These devices represent an expensive solution for both manufacturers and consumers. It is estimated that the conventional ATS used in an EU VI compliant heavy-duty 10 L engine can increase the final price of the truck more than $8000, which is transferred to the consumer as a vehicle price increase [15]. In addition, the ATS have other inherent drawbacks as the maintenance costs, operation costs (urea consumption in the SCR [16] and diesel fuel in the DPF [17]) and the engine efficiency reduction due to the increase of the pumping losses. These drawbacks enhance the CO_2 production, being contrary to what is searched in a short-term future. As stated, the future regulations aim to reduce the main hazardous pollutants (NOx, soot, particulates, HC, CO) while achieving significant reductions in the CO_2 emitted by the vehicle. The targets for CO_2 reduction are already set as 15% in 2025 (H2025) and 30% in 2030 (H2030) as compared to the current levels [18]. In this sense, alternatives able to simultaneously reduce the engine-out emissions as well as the CO_2 are needed to be integrated together in the vehicle. The improvement of the powertrain to achieve this goal can be tackled by different means: improving the combustion process (conversion efficiency), reducing the exhaust emission (decrease the ATS dimensions and associated fluid consumptions) and using alternative fuels with a positive carbon balance.

The increase of the conversion efficiency and simultaneous reduction of NOx and soot emissions has been extensively investigated in the last years [19][20]. Some of the major advancements were based on the application of low temperature combustion (LTC) techniques [21]. This type of combustion relies on achieving a high premixing degree and diluted environments, allowing to promote fast combustion processes and low in-cylinder temperatures [21]. As a consequence of the fast combustion process the heat transfer losses are reduced [22] and the fuel-to-work conversion efficiency is optimized [23]. Moreover, the use of premixed mixtures and high exhaust gas recirculation (EGR) levels allows to avoid the conventional NOx-soot trade-off. Different LTC concepts were developed in the last years, as the homogeneous charge compression ignition (HCCI) [24], partially premixed combustion (PPC) [25][26] and reactivity controlled compression ignition (RCCI) [27]. Among them, it has been found that the RCCI combustion mode offers important advantages compared to HCCI and PPC [28]. In this sense, the use of two fuels with different reactivity (a low reactivity fuel (LRF) and a high reactivity fuel (HRF)) allows to mitigate the combustion control issues found with the single-fuel LTC concepts, as HCCI [29] and PPC [30], and extend the operating limits by modifying the mixture reactivity on demand [31][32]. Despite of the improvements, it was found that it is not possible to apply RCCI combustion along the whole engine map due to either excessive pressure gradients at high load or excessive HC and CO emissions at low load [33], thus requiring the use of conventional combustion strategies to cover the critical parts of the map [34]. To overcome this issue, Benajes et al. proposed the dual-mode dual-fuel combustion (DMDF) concept [35]. This concept relies on modifying the injection and air management strategies compared to RCCI to mitigate the mechanical restrictions found with a fully premixed combustion [36]. Thus, while the injection strategy at low and medium load is set to promote a premixed RCCI combustion, as the engine load is increased, the high reactivity fuel injection is shifted towards the top dead center (fire) to promote a dual-fuel diffusive combustion with low pressure gradients. It should be remarked that even at high load conditions, the low reactivity fuel is still present in significant fractions (more than 30% of the total energy), having a fundamental role on the mixture preparation and combustion progress. Moreover, the DMDF concept has found to be flexi-fuel capable, allowing to use different fuels either as low reactivity [37][38] or high reactivity fuel [39]. Recent studies demonstrated that this concept still has benefits in terms of emissions and performance compared to conventional diesel combustion [40]. Nonetheless, further improvements are required to achieve the CO_2 targets for H2025 and H2030. One feasible path is the use of alternative fuels that promote a reduction of the CO_2 footprint in their lifecycle. This can be accomplished by different means as using bio-based fuels as ethanol or introducing advanced fuels, generally called e-fuels that rely on using the CO_2 as raw material during their production process [41]. This second path also enables the use of alternative energy sources as wind power and solar to produce the fuel. Among the different e-fuels reported, Oxymethylene ether (OMEx) appears as a good direct substitute of the diesel fuel to be used in compression ignition engines, or to be used blended with diesel, to provide benefits in soot and CO_2 emissions [42]. Moreover, the studies performed by Deutz et al. reported that the use of OMEx also allows to minimize the NOx emissions, since the EGR levels can be modified without exceeding the soot limits imposed by the authors [43]. The use of diesel-OMEx blends can enhance the market penetration of this fuel since it is expected that no modifications are needed in the distribution system as well as in the hardware of the internal combustion engine, i.e., it can be considered a drop-in fuel [44]. Nonetheless, the benefits on the CO_2 reduction will be decreased since OMEx will not replace totally the fossil fuel (diesel) [45].

The use of OMEx in its net form is not deeply addressed in the literature. Previous results from the authors demonstrated that the DMDF combustion concept operating with OMEx can realize EURO VI NOx with ultra-low soot levels at four different operating conditions, even in the case of full load operation [39]. Only small fuel consumption penalties were reported due to the larger combustion durations as a result of the low lower heating value (LHV) of OMEx. To expand these findings, the current work aims to evaluate the real potential of using OMEx as high reactivity fuel in all the engine map. To do so, a dedicated calibration operating in DMDF combustion with OMEx and gasoline as high and low reactivity fuels, respectively, is carried out in multi-cylinder engine (MCE) platform. The performance, combustion and emission results are compared to the DMDF diesel-gasoline calibration, also developed in this work, which is considered as the reference condition for comparison as it uses market fuels. Each one of the calibrations were obtained following a specific calibration methodology optimizing the brake thermal efficiency while maintaining the emissions values under pre-established limits. The studies about OMEx are emerging at this moment in the literature, and there are very few investigations of its usage in internal combustion engines, no one dealing with a complete calibration in a production engine operating under an advanced combustion mode. These kind of studies are necessary since the use of OMEx in dual-fuel combustion can contribute to solve the most significant challenges of the heavy-duty transportation sector (NOx, soot and CO_2).

2. Materials and methods

This section describes the experimental facilities, fuels and the calibration methodology used during the experimental tests.

2.1. Engine and test facility description

Table 1 presents the main characteristics of the medium-duty engine platform used during the investigation. The characteristics and external devices were maintained as similar as possible to those of the original commercial engine (high pressure injection system, turbocharger, high pressure (HP) EGR line). An additional low pressure (LP) EGR system was added to deal with the air management requirements. In this sense, the EGR flow can be controlled by both HP and LP paths. This allows to regulate the mass flow through the turbine, guaranteeing enough energy to meet the desired intake pressure. Moreover, the nominal compression ratio was reduced from 17.5:1 to 12.75:1 to allow extending the dual-fuel operation towards the full load. This was achieved by means of the piston geometry modification, optimizing the piston shape for the proposed combustion concept. More details can be found at [46]. A multi-point port fuel injection system was installed to provide the low reactivity fuel in the original intake manifold.

Table 1. Engine characteristics.

Engine Type	4 stroke, 4 valves, direct injection
Number of cylinders [-]	6
Displaced volume [cm^3]	7700
Stroke [mm]	135
Bore [mm]	110
Piston bowl geometry [-]	Bathtub
Compression ratio [-]	12.75:1
Rated power [kW]	260 @ 2200 rpm
Rated torque [Nm]	1450 @ 1200-1600 rpm

The engine was installed in a fully instrumented test cell, allowing to monitor the relevant pressure, temperature and flows as well as providing detailed information about the combustion process and emissions. The test cell scheme is presented in Figure 1, where both EGR routes can be differentiated. The HP route depart from the turbine inlet, passing through an intercooler, before being mixed with the compressed air. By contrast, the low-pressure line is placed after the turbine, being filtrated in a DPF and dried at a water filter to avoid particles as well as liquid water to reach the EGR valves and the compressor blades. A backpressure valve is placed after the LP EGR split to regulate the differential pressure between the exhaust and intake manifolds. Nonetheless, this pressure increase is limited, since it impacts the pumping work, decreasing the brake efficiency.

The injection parameters (except for the HRF injection pressure) of both fuels were controlled by means of a NI PXIe 1071 board. This board performed additional tasks as controlling other external devices as the back-pressure valve and the LP EGR concentration. In addition, the same board was used to record the high frequency data and perform an online heat release analysis, providing instantaneous results of the main combustion metrics (CA50, heat release rate, pressure gradients, etc.). This analysis was performed for each individual cylinder to assess possible dispersions on the air, EGR and LRF quantities (cylinder-to-cylinder dispersions). For this, AVL Kistler 6125C pressure transducers were mounted together with an AVL 364 encoder with a resolution of 0.2 crank angle degree (CAD).

Dedicated gas analyzers were installed to quantify the engine-out emissions. The gaseous emissions NOx, HC, CO and CO_2 were measured by means of a five-gas Horiba MEXA-7100 DEGR analyzer while the soot emitted was measured by means of an AVL 415S smoke meter in filter smoke number (FSN) units [47]. Two AVL 733 S fuel balances provided instantaneous measurement of both LRF and HRF while the air mass flow was measured by an Elster RVG G100 sensor. Each operating condition was measured three times during 40 s. The average values were recorded by means of an AVL PUMA interface. This software was also responsible to control the engine speed while the engine torque was regulated by the fuel injected. Table 2 presents the accuracy of the measurement devices and sensors used in this investigation.

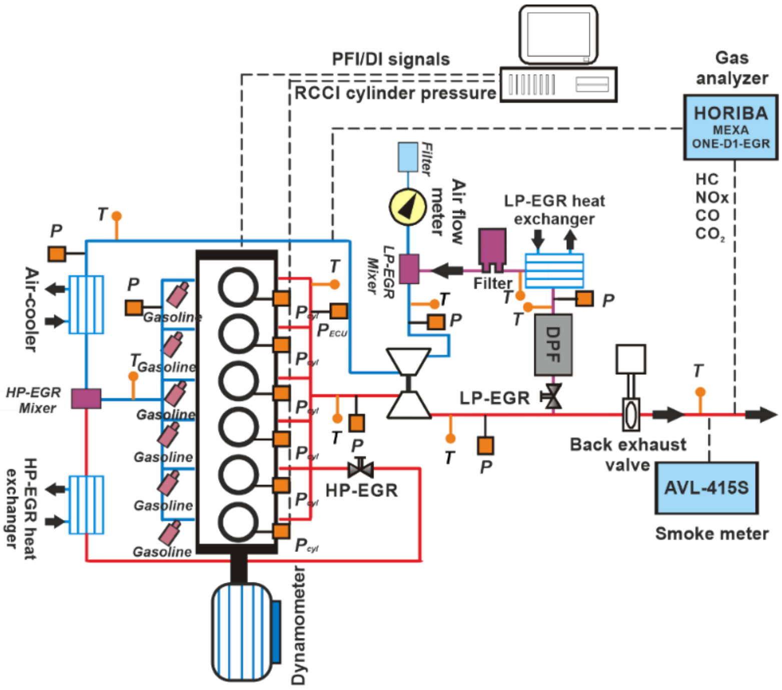


Figure 1. Test cell scheme.

Table 2. Accuracy of the instrumentation used in this work.

Variable measured	Device	Manufacturer / model	Accuracy
In-cylinder pressure	Piezoelectric transducer	Kistler / 6125C	±1.2 %
Intake/exhaust pressure	Piezoresistive transducers	Kistler / 4045A	±25 mbar
Temperature in settling chambers and manifolds	Thermocouple	TC direct / type K	±2.5 °C
Crank angle, engine speed	Encoder	AVL / 364	±0.02 CAD
NOx, CO, HC, O_2, CO_2	Gas analyzer	HORIBA / MEXA 7100 DEGR	4%
FSN	Smoke meter	AVL / 415	±0.025 FSN
Gasoline/diesel fuel mass flow	Fuel balances	AVL / 733S	±0.2%
Air mass flow	Air flow meter	Elster / RVG G100	±0.1%

2.2. Fuels characteristics

The characteristics of the fuels used in this work are presented in Table 3. For both calibration maps, commercial gasoline was used as LRF. From Table 3 the differences between diesel and OMEx regarding its physical and chemical properties can be highlighted. Both density and viscosity present differences higher than 30%. It is interesting to note that OMEx presents a high cetane number, i.e., a higher reactivity compared to the diesel fuel [48]. Significant changes can also be verified regarding the molecule composition. As previously stated, OMEx is known by having high oxygen content. In this investigation, OMEx fuel containing oxygen a mass fraction of 47.1% was used. This high oxygen concentration implies a reduction in the amount of energy stored by carbon in the molecule, decreasing its LHV.

Table 3. Properties of the different fuels used.

	EN 228 gasoline	EN 590 diesel	OMEx
Density [kg/m^3] (T= 15 °C)	720	842	1067
Viscosity [mm^2/s] (T= 40 °C)	0.545	2.929	1.18
Cetane number [-]	-	55.7	72.9
Carbon content [% m/m]	-	86.2	43.6
Hydrogen content [% m/m]	-	13.8	8.82
Oxygen content [% m/m]	-	0	47.1
RON [-]	95.6	-	-
MON [-]	85.7	-	-
Lower heating value [MJ/kg]	42.4	42.44	19.04

2.3. Engine calibration procedure

A dedicated methodology was designed to assure comparable results between both engine calibration maps (diesel-gasoline and OMEx-gasoline). The use of a step-by-step routine allows to optimize each operating condition in a similar manner. For the sake of clarity, the methodology was divided into three different steps: load achievement (Figure 2), fulfilment of NOx and soot constraints (Figure 3 and Figure 4), and fuel consumption optimization (Figure 5).

The first step is aimed to obtain the desired engine load using a dual-fuel low temperature combustion. To do this, the engine is started in conventional diesel combustion (CDC) as presented in Figure 2. In sequence, the gasoline injection is increased towards the desired load. Depending on the operating condition, the engine mechanical limits provided by the original equipment manufacturer (OEM) can be reached. It should be remarked that the limits are assessed for each one of the cylinders by means of six in-cylinder pressure transducers. In these cases, the SOI is delayed in steps of 1 CAD, shifting the combustion process towards the expansion stroke, reducing the maximum temperature and pressure and, therefore, reducing the mechanical demands. In specific high load conditions, the high pressure and temperature enables the autoignition of the LRF resulting in excessive pressure gradients. In these cases, delayed SOI with lower premixed energy ratio (PERs) must be used to achieve the desired engine load, denominated as premixing reduction. The subsequent actions aim to guarantee a stable combustion process. This is assessed by means of the coefficient of variation of the indicated mean effective pressure (COV_{IMEP}), stablishing a value of 4% as limit. The load achievement fulfilling the described constraints is the conditional to move to the next step.

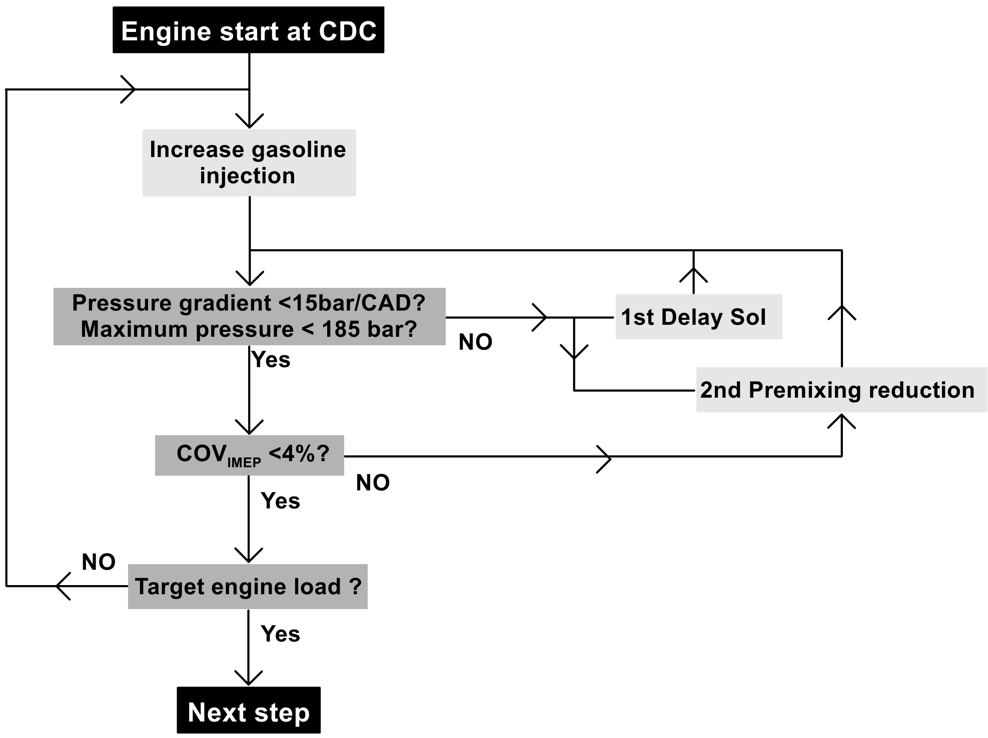


Figure 2. Representation of the different steps of the first stage of the calibration methodology.

The following step of the methodology addresses the settings modifications aiming to achieve pre-established emission limits. At first, EUVI NOx and soot <0.01 g/kWh are set as target for each operating condition. Nonetheless, different restrictions prevent the achievement of the normative in specific cases that will be presented and discussed in detail in the results section. Figure 3 depicts the different scenarios that could be obtained during the step 1 regarding soot and NOx emissions: 1) soot <0.01 g/kWh and NOx below the EUVI, 2) only soot <0.01 g/kWh, 3) only NOx below the EUVI and 4) NOx out of EUVI and soot >0.01 g/kWh. For the first case, no modification is required, and the methodology can proceed to the 3rd step. For the cases that do not fulfill the NOx constraints, specific strategies aiming to inhibit the thermal NOx formation mechanism should be employed. Some of the most effective paths to achieve this are the increase of the mixture dilution and the shifting of the combustion process to higher volumes, generally, at the cost of efficiency losses. The third scenario addresses conditions that are not able to provide soot emissions lower than 0.01 g/kWh. In this case, the settings must be modified to enhance the soot oxidation and to reduce its formation. The first part can be tackled by increasing the temperature and pressure inside the cylinder as well as the oxygen content. This can be achieved by means of an early combustion process and higher inlet pressures. The soot formation can be reduced by avoiding rich zones inside the combustion chamber. For this, one can increase the injection pressure to improve the spray penetration and vaporization (better mixing). Last, the first stage can deliver operation conditions that are not able to fulfill neither soot <0.01g/kWh nor NOx limits. This challenging scenario demands a balance from the last two cases, i.e., strategies to decrease NOx will increase the soot formation and vice-versa. In this sense, the settings modifications are explored towards the fulfillment of both emission constraints. However, some operating conditions are not able to achieve simultaneous reduction of both contaminants. Generally, these conditions are found from medium to full load points where the engine hardware works near to the design limits (e.g. pressure and temperatures at the turbocharger, in-cylinder pressure gradients). Therefore, the emission limits must be relaxed to allow achieving similar power output as that from the stock engine running under conventional diesel combustion.

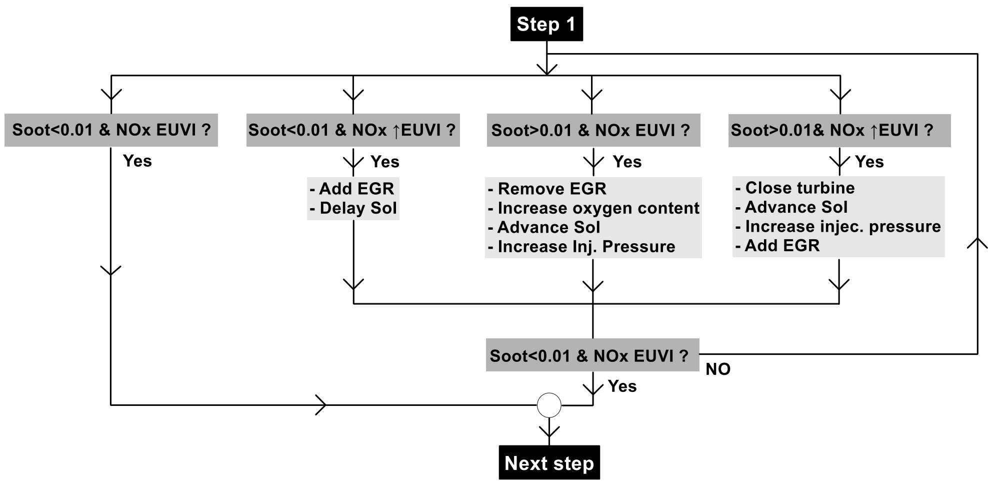


Figure 3. Representation of the different steps of the second stage of the calibration methodology.

It is interesting to remark that during the OMEx calibration, it was found that none of the operating conditions achieved soot emissions higher than 0.01 g/kWh. This allowed to simplify the previous calibration stage as depicted in Figure 4. At the end, one should be only take care of achieving the EUVI NOx limits, solving most of the challenges that were previously discussed.

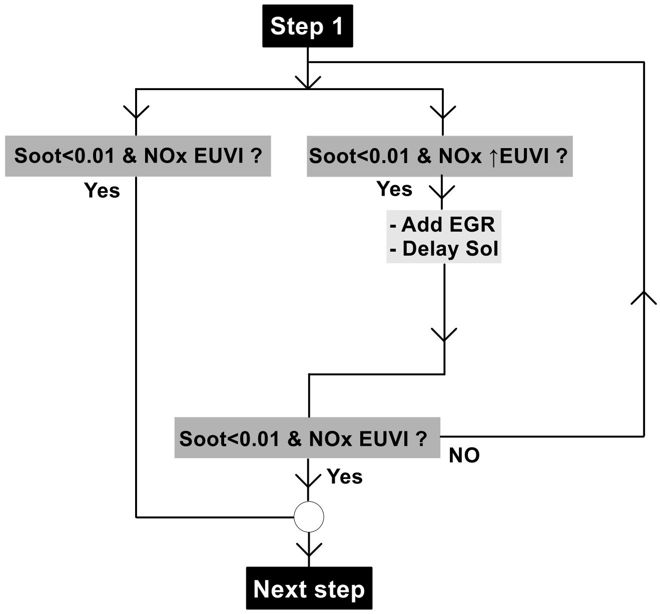


Figure 4. Representation of the different steps of the second stage of the calibration methodology using OMEx as high reactivity fuel.

The third and last stage of the calibration methodology aims to improve the brake specific fuel consumption for each operating condition. This is achieved by splitting the routine in loops according to the dominant parameters of the combustion process, as it can be seen in Figure 5. The first loop addresses the split of the two different EGR sources: high pressure EGR and low pressure EGR. The use of the different routes impacts the whole air management system dynamics. Using only HP EGR means a reduction of the mass flow through the turbine, i.e., a reduction of the total energy available to produce work and move the compressor. Since the engine stock turbocharger has a variable geometry nozzle, it can be closed to compensate the mass flow reduction at the cost of higher pumping losses. By contrast, the use of only LP EGR increases the amount of energy in the turbine. Nonetheless, the pressure difference from the exhaust line after the turbine and the intake pressure before the compressor requires the use of a backpressure valve. This helps to increase the pressure difference, driving the exhaust flow back to the intake manifold. Unfortunately, this pressure increase after the turbine is multiplied by the turbine expansion ratio, meaning significant increases in the pumping losses. The right split between both lines means a path to reduce the losses by pumping while providing the required amount of EGR flow to achieve the desired LTC operation. In this sense, this split was varied for each operating condition having the engine torque as reference. At the end, the split with higher torque was chosen as the optimized air management system setup. In sequence, similar strategies were used to determine the best injection timings, EGR levels (maintaining the optimized split) and PER values. It should be remarked that these parameters are moved in a small range, since they must preserve the emissions constraints from the first step of the optimization procedure.

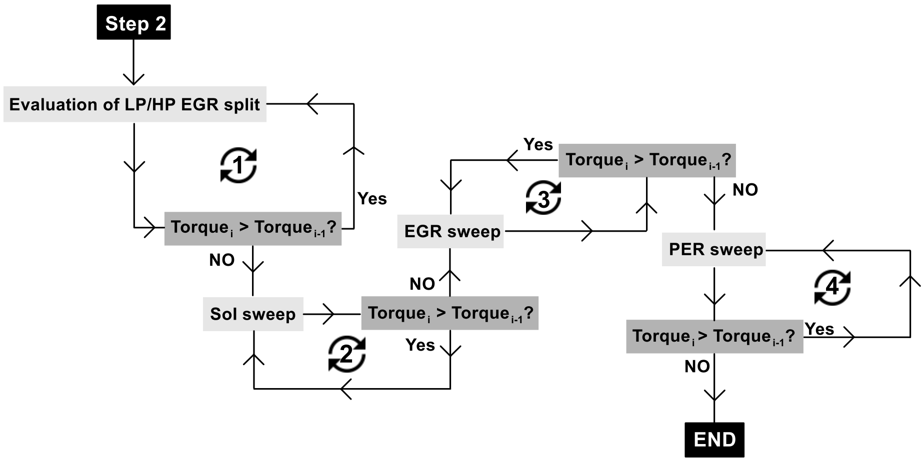


Figure 5. Representation of the different steps of the third stage of the calibration methodology.

Figure 6 summarizes the calibration routine with some of the most important remarks of each one of the three steps. It should be noted that an additional PER sweep step is included in this last Figure. This has as aim to validate the assumption made at step 1 which considers the highest PER as the best initial solution for each operating condition.

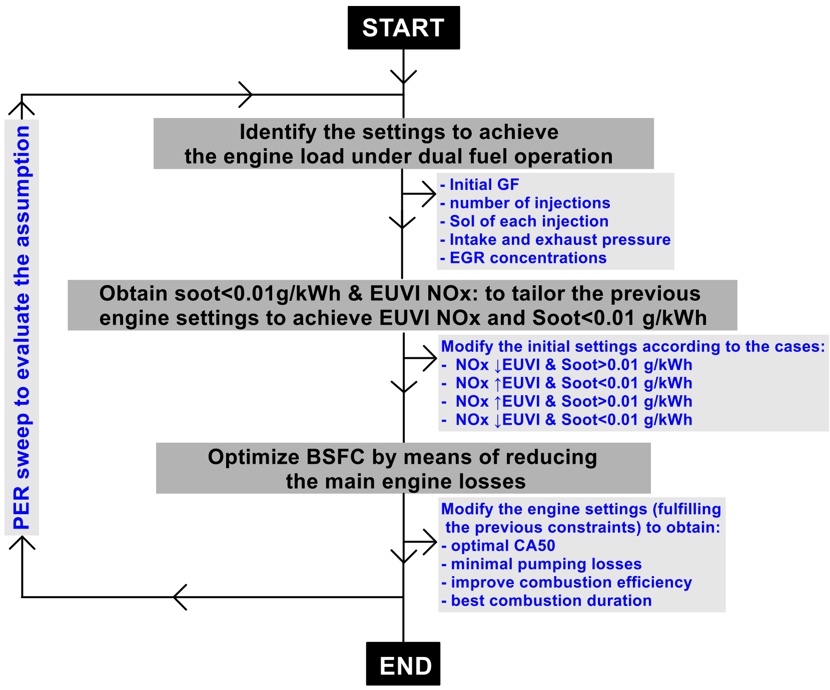


Figure 6. Summary of the methodology used to determine the reference condition for each fuel blend.

3. Results and discussion

This section describes the combustion strategies used to calibrate the DMDF concept with each pair of fuels (diesel-gasoline and OMEx-gasoline) given the challenges that were found during the engine mapping. In sequence, the performance and emissions results for both concepts are presented. For each parameter, the results of the diesel-gasoline and the OMEx-gasoline calibration are shown. Finally, the maps showing the difference between both calibrations are presented to highlight the advantages and drawbacks of using the different reactivity fuels.

3.1. Combustion strategies description

Figure 7 shows a conceptual model of the injection strategy developed to cover the whole engine map. The conceptual engine map is divided into different areas considering the NOx and soot constraints found in each region. At low-to-medium engine load, two HRF injections were used following the principles of the fully premixed RCCI combustion: a first injection to increase the reactivity of the mixture near to the cylinder wall and a second injection to provide ignitable regions; both early injections in the compression stroke. This allowed to achieve EURO VI NOx and ultra-low soot levels while maintaining a high efficiency in a great portion of the map, as presented in Figure 7 (a). Nonetheless, as the engine load was increased towards the full load operation, the high pressure gradients required a modification of the injection strategy. The HRF pilot injection was removed as it provided an instantaneous ignition of the mixture. In this sense, the HRF injection was concentrated in one injection to increase the amount of fuel and provide more energy release by the diffusive combustion process. To achieve this diffusive combustion, the start of injection was shifted towards the TDC fire. However, the use of this strategy enhances the soot production and the NOx emissions. The first is a result of the premixing decrease, producing rich mixture zones. This diffusive combustion process also extends the combustion duration, which means that the nitrogen molecules have higher residence times at high temperatures. Therefore, these conditions require to relax the emissions constraints to allow achieving higher engine loads. First, the soot emissions limits were increased to 2 FSN. Nonetheless, as the engine load approaches to full load operation, it was also necessary to relax the NOx emissions constraints up to 2 g/kWh.

Figure 7 (b) depicts the calibration strategy using OMEx as HRF. It can be seen that the injection strategy follows the same pattern than for diesel-gasoline since the excessive pressure gradients was a common factor for them, and it was the main reason for the injection modification. However, it is interesting to note that the final OMEx-gasoline calibration is EUROVI compliant for NOx with soot <0.01 g/kWh in the whole engine map. As the burning of OMEx does not produces soot emissions, the air management system can be modified to provide the required dilution levels in which the NOx emissions are inhibited. Nonetheless, it can be observed that there is a slight decrease in the maximum load achieved from 1200 to 1800 compared to the original diesel-gasoline calibration. The low LHV of the OMEx leads to almost twice the flow rate in the high-pressure pump. At high loads, the limits of the high-pressure pump are achieved, which prevents to achieve the full load at these specific engine speeds.

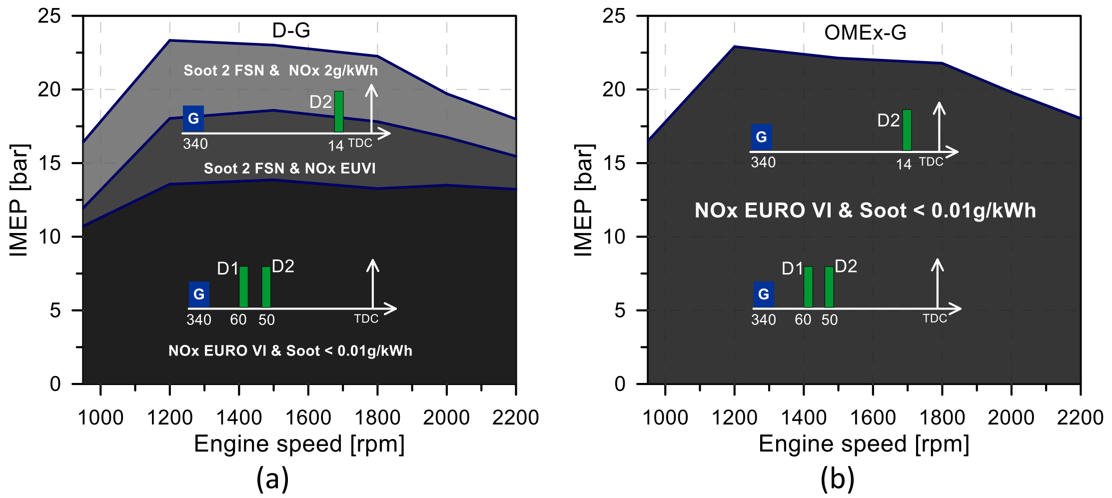


Figure 7. Conceptual description of the injection strategy as well as the different limits in terms of NOx and soot for (a) diesel-gasoline calibration and (b) OMEx-gasoline calibration.

Figure 8 shows the different HRR shapes that are identified in the calibration map, depending on the injection strategy used. For fully premixed cases, the combustion is characterized by a fast and uniform HRR since there is no injection during the combustion development. By contrast, a dual-fuel diffusive combustion is found at high engine loads. In this case, the LRF quantities are reduced and the HRF injection is located closer to the TDC. In this sense, once the HRF injection starts, there is a first premixed peak originated from the LRF and the small amount of HRF. Once the premixed phase finishes, the combustion process is sustained along the expansion stroke by a diffusive combustion of the HRF. An additional phase can be also considered, denominated partially premixed combustion. It addresses the transition from the fully premixed combustion to the dual-fuel diffusive one. As it can be seen in Figure 8, in this combustion mode most of the energy is released during the premixed phase. Nonetheless, a small portion of the energy is still released in a diffusive manner, illustrated by the inflection at the end of the HRR profile, helping to avoid excessive pressure gradients.

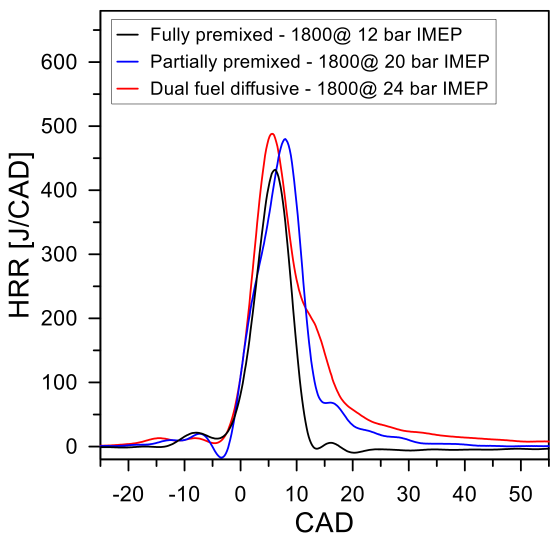
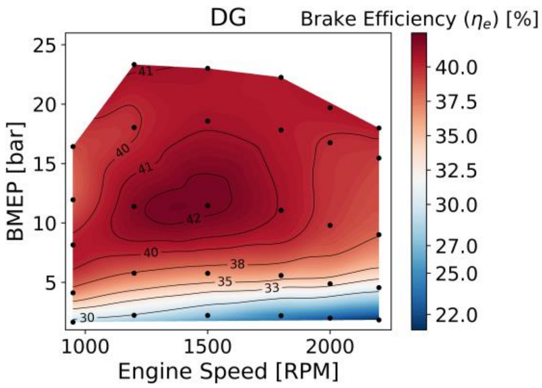


Figure 8. Example of the characteristic heat release rates for fully premixed, partially premixed and dual fuel diffusive combustion process.

3.2. Dual-fuel mapping comparison

This subsection presents the performance, combustion and emissions results in form of calibration maps for both pair of fuels tested diesel-gasoline and OMEx-gasoline and the maps showing the difference between both calibrations. Figure 9 presents the brake efficiency values obtained with the DMDF concept using diesel (Figure 9 (a)) and OMEx (Figure9 (b)) as high reactivity fuels. Some interesting differences can be remarked for each one of them. First, they differ regarding the location of the maximum efficiency zone in the calibration map. The diesel-gasoline combustion presents a maximum efficiency near to medium load conditions, where the concept can operate in fully premixed combustion without emissions and mechanical restrictions. Once the engine load is increased, the engine settings must be modified to tackle the increase of the pressure gradients and achieve a simultaneous reduction of the pollutants. As discussed in the calibration methodology description (subsection 2.3), these strategies generally result in efficiency losses, since they affect the combustion phasing and the air management system. By contrast, the OMEx-gasoline calibration provides the highest efficiency condition at a higher engine load (almost 75% of engine load). Moreover, the OMEx peak efficiency is 2% higher than that from the diesel-gasoline calibration. The combination of both results is interesting from an application point of view for both conventional and hybrid applications [49]. Since this engine platform is generally used for product distribution and long-haul applications, the engine tends to operate always in higher engine loads, increasing the importance of the medium to high engine load zones on the final truck performance. Moreover, this high load high efficiency condition can be explored in hybrid applications as series architectures, developing the control strategy to operate inside this zone and optimize the benefits of the efficiency improvement.



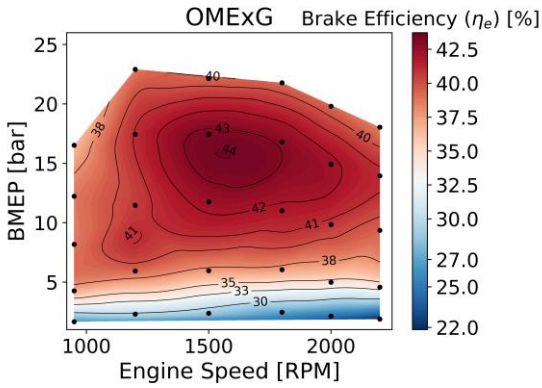


Figure 9. Brake efficiency maps for (a) diesel-gasoline and (b) OMEx- gasoline.

Figure 10 shows the absolute difference between the calibration maps to highlight the benefits and drawbacks of replacing diesel by OMEx as HRF. As it can be seen, the use of OMEx fuel provides higher efficiency values than those found with diesel for most of the calibration map. The most critical conditions are found at low engine speed and at full load conditions. It is believed that the higher injection durations and consequent combustion durations plays a dominant role on the conversion efficiency at high load conditions. Nonetheless, additional losses mechanisms as pumping losses and combustion phasing can also influence the results. Each one of these parameters are evaluated next to highlight the effect of the HRF replacement.

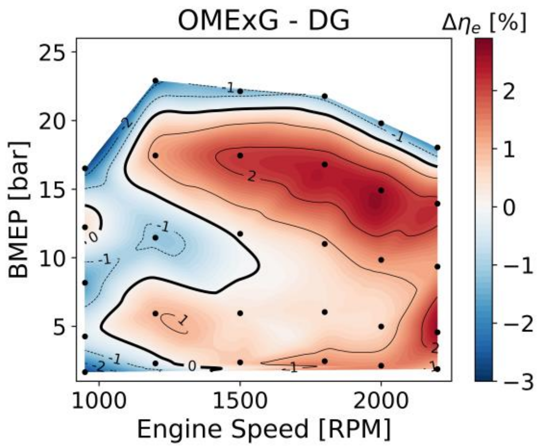


Figure 10. Brake efficiency difference map between OMEx-gasoline and diesel-gasoline.

Figure 11 depicts the combustion duration (CA90-CA10) results for both diesel-gasoline and OMEx-gasoline calibrations. It can be verified that the combustion regime has a dominant role on the combustion duration values. At fully and partial premixed combustion, the CA90-CA10 results are generally lower than 15 CAD. However, at high to full load, the use of the diffusive combustion extends these numbers to values higher than 20 CAD.

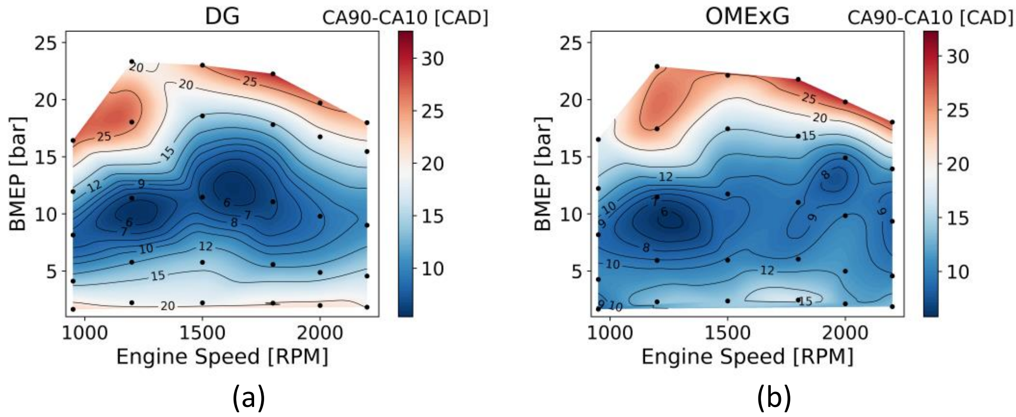


Figure 11. Combustion duration maps for (a) diesel-gasoline and (b) OMEx- gasoline.

The difference maps depicted in Figure 12 allow to state that the use of OMEx as HRF usually provides lower to similar combustion duration values, mainly in the premixed region. This can be attributed to the high reactivity of this fuel (i.e., high cetane number), which enhances the combustion velocity. Nonetheless, this trend is inverted at high load conditions as consequence of the lower LHV of OMEx. In this sense, the injection durations must be increased, extending the combustion process towards the expansion stroke and providing higher time for heat transfer phenomena, thus decreasing the engine efficiency.

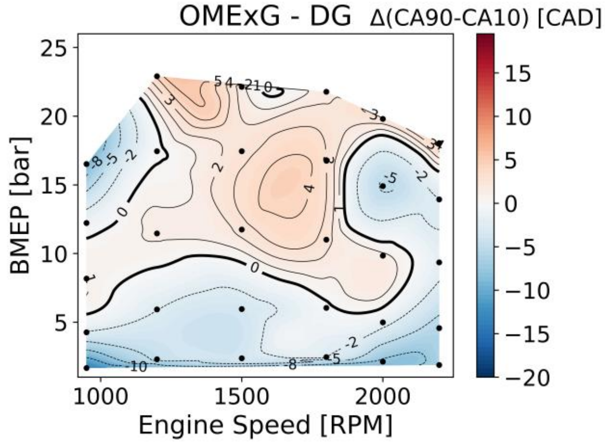
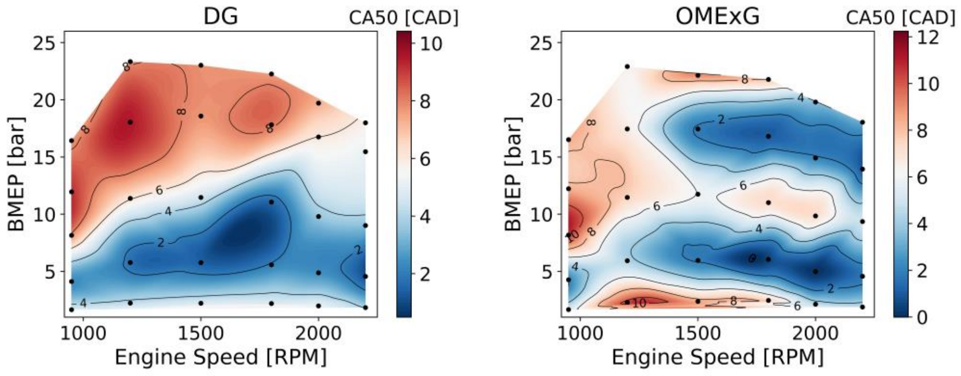


Figure 12. Combustion duration difference map between OMEx-gasoline and diesel-gasoline.

The combustion phasing (CA50) also plays a fundamental role on the combustion efficiency. This parameter is defined as the crank angle at which the 50% of the energy contained in the fuel is released, and generally presents an optimum value. In case of having an earlier CA50 than the optimum, the compression work is increased, meaning that more energy is provided to the gases instead to generate work through the piston. By contrast, delayed CA50 shifts the combustion process to the expansion stroke, resulting in lower in-cylinder pressures and work transfer to the piston. The use of diesel-gasoline in premixed conditions required the use of early combustion phasing as presented in Figure 13. This is consequence of the early injection timings needed to achieve enough time to ensure a proper mixing of the fuel before the start of combustion to reduce the soot emissions. The replacement of diesel by OMEx solved the issues related to the soot formation. In this sense, the injection strategy can be moved to realize better combustion phasing without impacting the soot formation.



(a) (b)

Figure 13. 50% of mass fraction burned maps for (a) diesel-gasoline and (b) OMEx-gasoline.

Figure 14 depicts the difference map for the CA50 values considering both pair of fuels. It is possible to see that the there is a trend inversion of the CA50 map at around 50% engine load. This correlates to the different strategies to achieve the NOx limits at high load conditions for the HRF evaluated. Since the diesel-gasoline combustion demonstrated to provide higher soot emissions at high load, the NOx emissions cannot be decreased by means of the EGR addition, as this strategy would remove oxygen from the combustion chamber. Alternatively, the combustion process was shifted towards to expansion stroke to decrease the in-cylinder pressure and temperature to inhibit the

NOx formation mechanism. At these conditions, the soot emission was decreased by closing the VGT rack position to increase the oxygen content, with the consequent increase of the pumping losses. Contrary to this, the OMEx-gasoline combustion provided zero soot independently on the engine load. Therefore, the NOx emissions can be controlled by increasing the EGR amount to reduce the in-cylinder temperature. This allowed to achieve a balance between NOx, EGR and combustion phasing, optimizing them to reduce the fuel consumption.

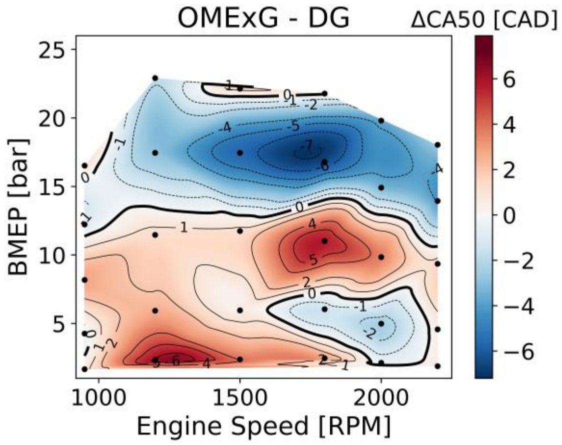
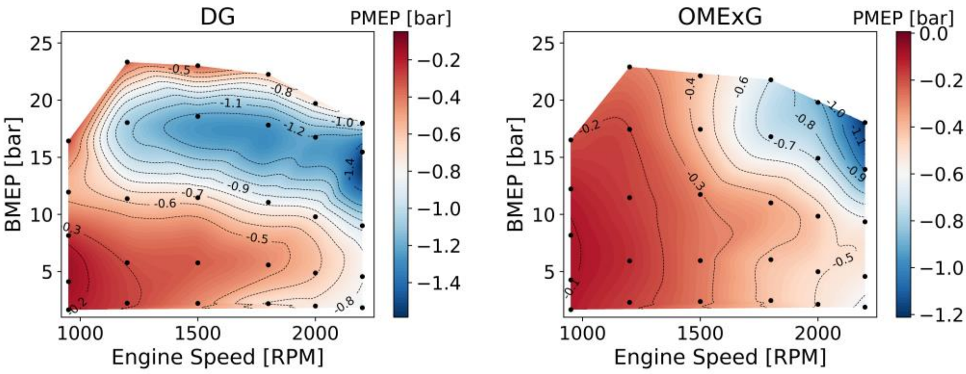


Figure 14. 50% of mass fraction burned difference map between OMEx-gasoline and diesel-gasoline.

Another significant source of efficiency loss is the pumping work that should be provided by the engine during the charge renovation process. This process addresses the exhaust and intake strokes. Generally, the pressure difference between the exhaust and intake pressure is negative, meaning a loss of work. It should be remarked that there are specific cases where the average intake pressure can surpass the exhaust, resulting in a positive work. The average pressure at intake and exhaust process are affected by the air management system. i.e., the rack position of the VGT, the operating point inside the turbine and compressor maps and the mass flows through them. Figure 15 (a) and Figure 15 (b) depict the pumping mean effective pressure (PMEP) for both calibrations. As it can be seen, the diesel-gasoline calibration present a significant increase in the pumping losses at medium to high engine loads. These conditions are prone to produce soot. Therefore, to maintain the soot levels under the proposed constraints, the VGT should be closed to provide higher fresh air mass. This flow throttling results in an increase of the turbine inlet pressure, increasing the pumping losses. By contrast, the use of OMEx provides zero soot emissions and also contributes to the oxygen mass inside the combustion chamber. In this sense, the air management requirements are decreased allowing to operate with more opened turbine rack positions compared to diesel-gasoline, which enables a significant reduction of the total pumping losses.



(a) (b)

Figure 15. Pumping losses maps for (a) diesel-gasoline and (b) OMEx- gasoline.

Figure 16 shows the pumping losses difference map between OMEx-gasoline and diesel-gasoline. As it can be seen, the use of OMEx allows to achieve PMEP reductions up to 70%. Most of the OMEx calibration presents PMEP benefits, mainly on the zone where the diesel-gasoline calibration requires closing the turbine to maintain the soot and NOx emissions under the desired constraints (medium to full load). It is also interesting to remark that the OMEx calibration presents a more uniform behavior, being mainly affected by the engine speed as a consequence of the higher flow rates at the exhaust manifold, increasing the engine back pressure. Therefore, it can be concluded that this set of parameter allows to improve the OMEx operation compared to the diesel in terms of brake efficiency values.

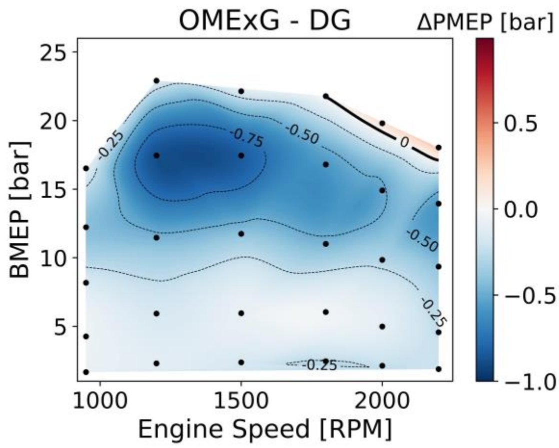
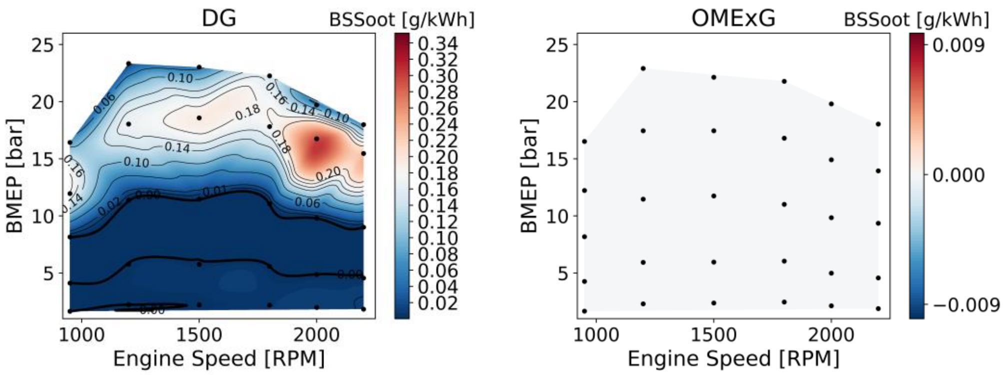


Figure 16. Pumping losses difference map between OMEx-gasoline and diesel-gasoline.

Figure 17 presents the soot emissions results obtained with the proposed calibration methodology for both HRF. As it can be seen in Figure 17 (a), contrasting results are found for diesel-gasoline depending on the combustion strategy used. Fully premixed zones were able to provide zero soot emissions as the fuel is premixed with the air, thus avoiding rich zones inside the combustion chamber. By contrast, the diffusive zone results in significant engine-out soot levels as consequence of the low oxygen content and the high diesel mass injected during the combustion development. Therefore, this fuel burns in low oxygen environments producing soot in comparable levels to those found with modern diesel engines. A global evaluation of the impact of this contrasting emission map should be performed by means of a driving cycle analysis, since the operation time differs from point to point, which results in different weights for these zones.

Previous studies reported that the use of OMEx allows to reduce the soot formation due to its high oxygen content as well as its non-direct carbon-to-carbon bonds. In this sense, the replacement of diesel by OMEx should benefit the engine-out soot. During the experimental evaluations, the AVL 415S was not able to provide measurements different than zero, independently on the operating condition evaluated. This means that large soot molecules are not formed during the OMEx combustion. Nonetheless, oxygenated fuels are prone to produce nanoparticle and specific measurements should be performed to assess them.



(a) (b)

Figure 17. Brake specific soot maps for (a) diesel-gasoline and (b) OMEx- gasoline.

From the difference maps depicted in Figure 18, it can be seen that both HRF are able to provide similar soot emissions up to medium load. In this region, the fuel characteristics do not affect the soot formation since the fuel can be fully premixed in an environment with excess of air. Nonetheless, the significant increase of the soot emissions for the diesel-gasoline calibration at the high load zone is a direct consequence of the hardware limitations of the engine, which cannot provide more oxygen due to the excessive temperature and pressure values at the turbine inlet. In this sense, the use of OMEx seems to be a path to overcome this issue, providing enough air to oxidize the carbon molecules.

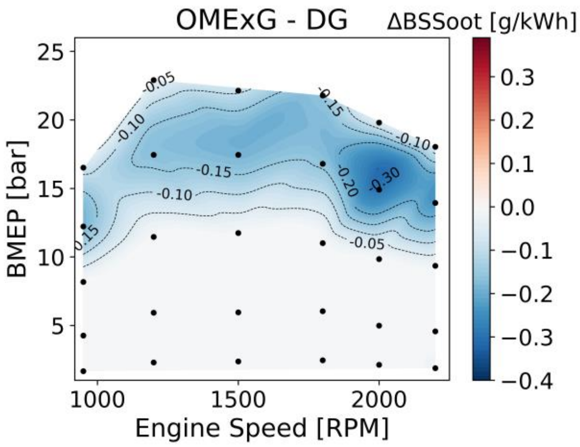
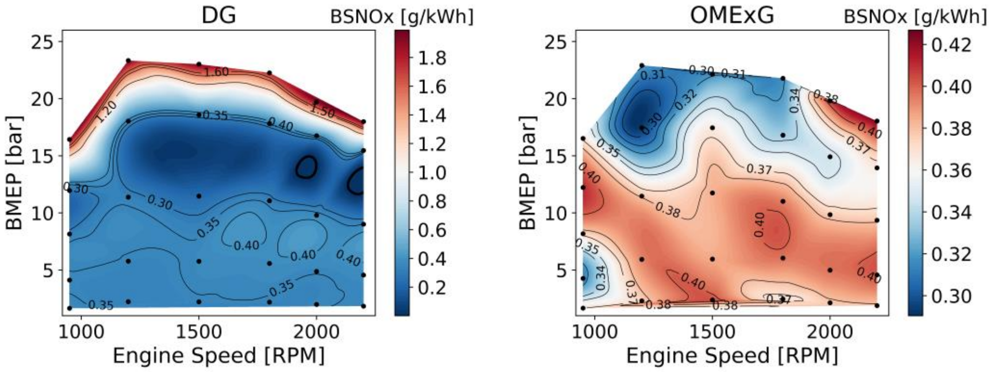


Figure 18. Brake specific soot difference map between OMEx-gasoline and diesel-gasoline.

One of the advantages of the premixed combustion is the capability of avoiding the conventional NOx-soot trade-off. Nonetheless, in the zone of the maps where this combustion mode cannot be implemented (high loads), both emissions start to correlate. This means that, to accomplish with the soot limit at the zones of the maps prone to produce soot, the NOx emissions must be increased by reducing the EGR amount. This can be inferred from Figure 19, where the calibration maps for both fuels are depicted. For diesel, the NOx emissions can be maintained at EUVI levels up to 80% of the engine load. This is a consequence of extending the soot emissions limit to 2 FSN. Higher loads are limited by the turbocharger, which cannot provide enough air due to the mechanical limitations at the compressor. In this sense, the EGR concentration must be reduced to allow a higher amount of fresh air flowing through the compressor and maintaining the soot levels under the proposed constraints. Consequently, the NOx emissions limit is increased up to 2 g/kWh, since higher temperatures are achieved with this action. By contrast, the use of OMEx as HRF was found to be an effective way to achieve zero soot emissions for the whole calibration map. Therefore, the EGR levels can be managed according to the requirements imposed by the NOx emissions without impacting the soot emissions. This allows to also fulfill the EURO VI limits independently on the operating condition evaluated by means of reducing the in-cylinder temperature with EGR dilution, as depicted in Figure 19 (b).



(a) (b)

Figure 19. Brake specific NOx maps for (a) diesel-gasoline and (b) OMEx- gasoline.

The difference NOx map shown in Figure 20 remarks the conditions where the NOx emissions starts to be out of the EURO VI limits for the diesel-gasoline calibration while the reference OMEx-gasoline map is close to 0.4 g/kWh. As it can be seen, the diesel-gasoline calibration exceeds the NOx emissions only in a narrow zone close to the full load conditions. It is expected that this small zone should have a low impact on the final results of a driving cycle. This assumption considers that the remaining time at full load conditions are low, having a low weight on the final results.

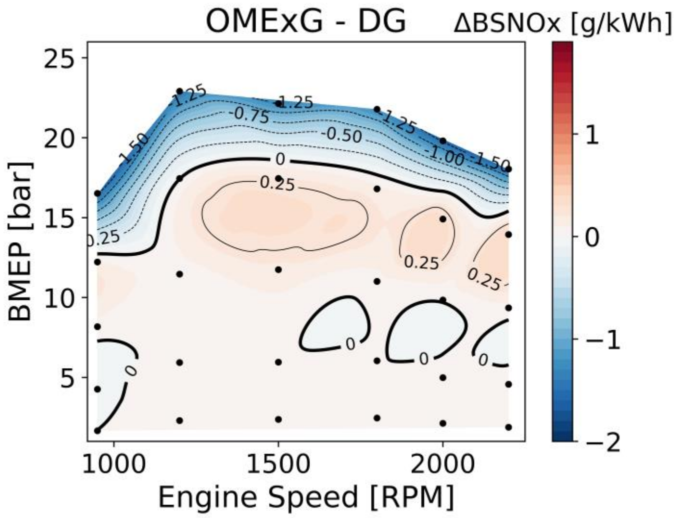
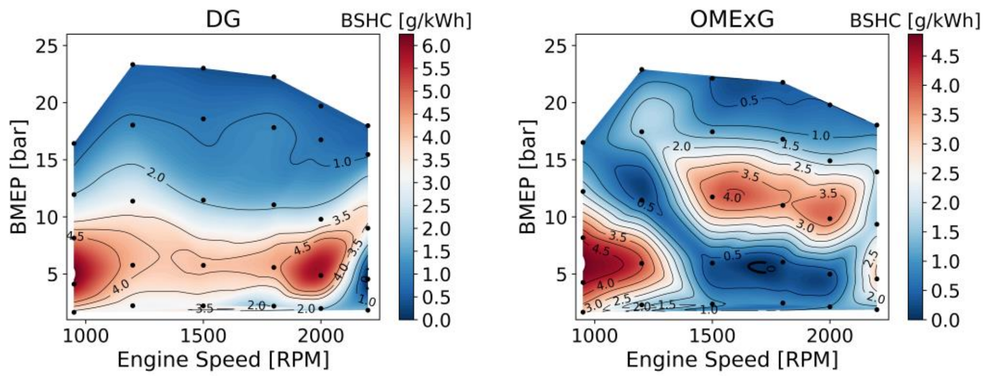


Figure 20. Brake specific NOx difference map between OMEx-gasoline and diesel-gasoline.

Figure 21 and Figure 22 depict the unburned hydrocarbons (HC) results for both calibrations as well as their absolute difference. It can be seen that the highest HC results are found at the fully premixed combustion zone. During the compression stroke, part of the fuel is directed to the piston gaps due to the flow motion inside the cylinder. The fuel that enters into these regions cannot be oxidized since the equivalent diameter is less than the critical one, meaning that the oxidation reactions will be inhibited due to the heat losses. It is expected that the emission of this contaminant should correlate with the quantity of premixed fuel injected. To validate this assumption, the premix energy ratio (PER) maps are illustrated in Figure 23. It is noted that at low load conditions (10% load), the premixed energy ratios are zero, indicating that only diesel is used. For engine loads up to 50%, the PER values are steeply increased, reaching maximum values of 80-90%, depending on the high reactivity fuel. The use of OMEx allows to increase the PER due to its higher cetane number, which enables to maintain the mixture inside stable ignitable conditions. As previously discussed, the engine load increase required a modification in the injection strategy to set a delayed start of injection and lower PER, i.e., an increase in the diffusive combustion mode. Consequently, the unburned hydrocarbons emissions are decreased as the losses by the fuel trapped in the piston crevices are inhibited.



(a) (b)

Figure 21. Brake specific unburned hydrocarbons maps for (a) diesel-gasoline and (b) OMEx- gasoline.

The comparison between the HC results from both calibrations (Figure 22) shows that there are contrasting trends that depends on the engine load. It is suggested that this can be coupled to the EGR levels used at each condition and the reactivity of the fuel [50]. For low to medium loads, both fuels can provide EUROVI NOx and soot levels due to the premixed combustion, which is based on high levels of dilution. In this sense, the major modification is caused by the increase of the mixture reactivity by the OMEx addition, decreasing the unburned hydrocarbons. By contrast, since no soot was verified using OMEx, the NOx emissions were decreased by means of higher EGR levels, as it can be realized in Figure 24. This higher dilution levels, and consequent lower temperature, also impacts the oxidation rates. Consequently, the unburned hydrocarbons are increased.

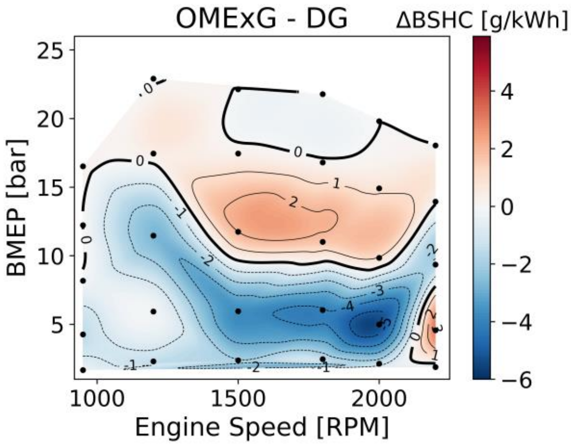


Figure 22. Brake specific unburned hydrocarbons difference map between OMEx-gasoline and diesel-gasoline.

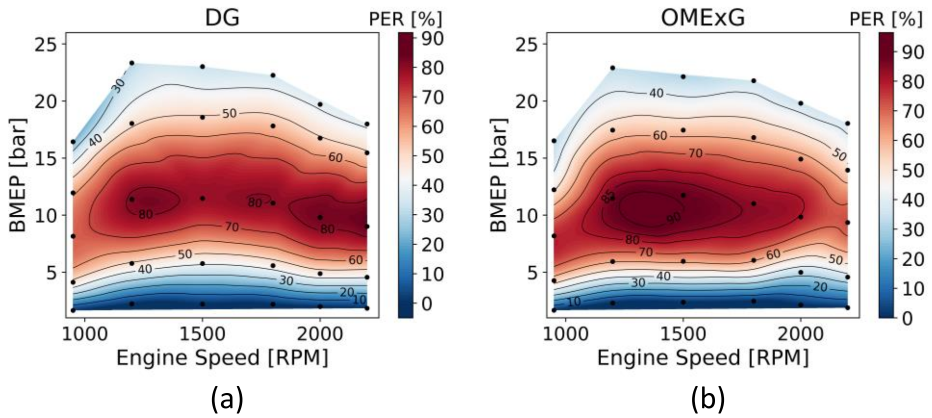
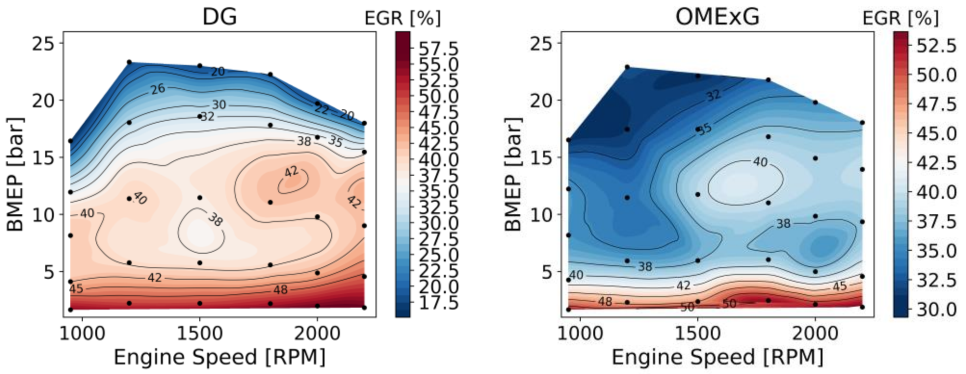


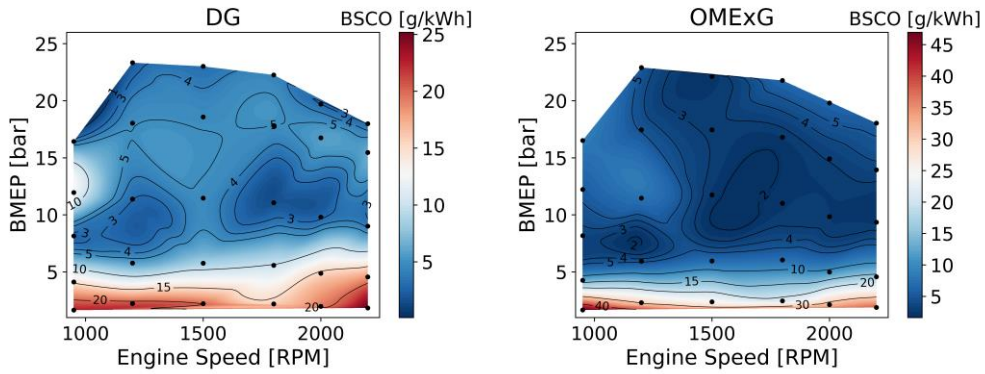
Figure 23. Premixed energy ratio maps for (a) diesel-gasoline and (b) OMEx- gasoline.



(a) (b)

Figure 24. Exhaust gas recirculation maps for (a) diesel-gasoline and (b) OMEx- gasoline.

Lastly, the CO emissions are depicted in Figure 25 for the diesel-gasoline and OMEx-gasoline calibrations. This emission seems to be mostly affected at low load conditions, where its values can be as greater as 40 g/kWh in the case of using OMEx as HRF. These conditions are characterized by low temperature with EGR levels higher than 50%, which are suitable to inhibit the oxidation reactions. It should be remarked that the unburned hydrocarbon emissions do not have a similar trend than those of the CO due to the combustion strategy used for these low load conditions. As stated above, the PER values are zero, indicating that no gasoline is injected for these conditions. Therefore, the combustion is based on a partially premixed combustion with early HRF injections. Nonetheless, the small fuel amount and the low injection pressure does not provide enough momentum to reach the cylinder and piston walls. In this manner, the HC and CO are decoupled from these operating conditions.



(a) (b)

Figure 25. Brake specific carbon oxide for (a) diesel-gasoline and (b) OMEx- gasoline.

Figure 26 depicts the difference map between both calibrations. It can be stated that the fuel modification has low impact on the CO emissions in most of the calibration map. The most significant changes are perceived at low load conditions, where the OMEx fuel produces higher CO concentrations. This phenomenon can be related to the required modifications to achieve the EUVI NOx levels. The higher global reactivity realized by using OMEx leads to higher heat release peaks, increasing the maximum temperature at the combustion chamber. Therefore, the NOx formation is enhanced, requiring additional increases in the EGR rates compared to those of the conventional diesel-gasoline calibration, thus increasing the CO emissions.

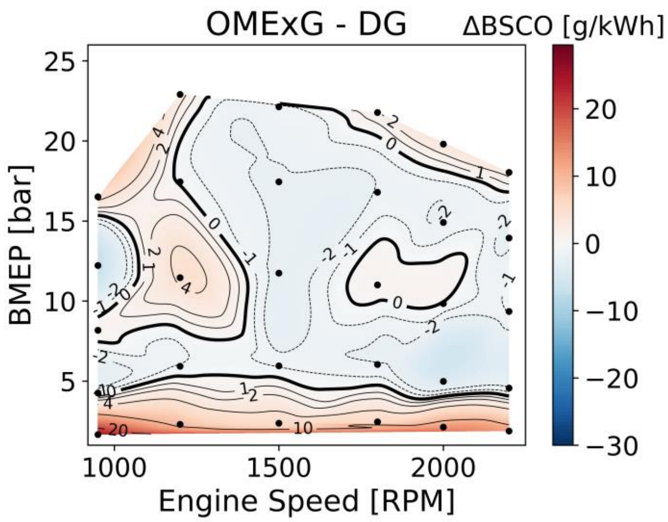


Figure 26. Brake specific carbon monoxide difference map between OMEx-gasoline and diesel-gasoline.

4. Conclusions

This work evaluated the use of the dual-mode dual-fuel combustion in a stock multi-cylinder engine platform as a pathway to fulfill the current and future emissions legislations in terms of NOx, soot and CO_2. To do this, a full map calibration was carried out following a specific calibration methodology to obtain the best fuel consumption with lower NOx and soot as always as possible. First, the use of conventional fuel for both HRF (diesel) and LRF (gasoline) was investigated aiming to enhance the market penetration of the concept from the use of drop-in fuels. The results allow to conclude that:

- This setup can mitigate both NOx and soot emissions whenever a fully premixed combustion can be employed.
- From engine loads higher than ≈65%, the mechanical constraints (pressure gradients, turbocharger boundary conditions…) prevent to follow the previous premixed strategies.
- From 65% to full load the injection strategy should be modified towards a diffusive combustion rather than a fully premixed one. This modification impairs both NOx and soot at different levels, exceeding the EUVI steady-state targets.

Second, the use of OMEx as a high reactivity fuel was assessed to evaluate the impact of its chemical composition on the dual-fuel combustion process. The most interesting remarks can be summarized as follows:

- The use of OMEx allows to decrease the soot emission to values not detectable by the current measurement device.
- The NOx emissions can be reduced by means of the EGR increase, without impacting the engine-out soot levels.
- The first two points allow to optimize the air management system, reducing the pumping losses. Moreover, the combustion process can be better phased according to the operating conditions since the soot restrictions are not present anymore. This set of benefits allows to improve the engine efficiency up to 3% compared to the previous D-G calibration.

In this sense, it can be concluded that the DMDF combustion concept can be an alternative to realize reductions in NOx and soot, helping to decrease the final truck cost. It is suggested that the cost benefits from simplifying the ATS and mitigating the use of urea can be superior to those of adding a port fuel injection system injection system to the vehicle. In addition, the use of OMEx provides even higher benefits with respect to these contaminants, allowing to fulfill the EUVI steady-state limits for all the operating conditions. It should be considered that OMEx can be extracted from the CO_2 in the environment, presenting a lower CO_2 footprint than the conventional fuels. This can be summed up with the better efficiency values than diesel-gasoline, resulting in CO_2 savings in the WTW approach. Nonetheless, OMEx-G is not overall superior than D-G since it can be evidenced the higher unburned products in a wide range of the operating map. The most critical point is the higher CO concentration at low load conditions that can be a challenge for conventional diesel oxidation catalyst. Moreover, oxygenated fuels are prone to produce significant number of nanoparticles during the combustion process. These points should be investigated in future works.

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Abbreviations

ASTM: American Society of Testing and Materials

ATDC: After Top Dead Center

CAD: Crank Angle Degree

CA50: Crank angle at 50% mass fraction burned

CDC: Conventional Diesel Combustion

CO: Carbon Monoxide

CR: Compression Ratio

DI: Direct Injection

DPF: Diesel Particulate Filter

ECU: Electronic Control Unit

EGR: Exhaust Gas Recirculation

EVO: Exhaust Valve Open

FSN: Filter Smoke Number

HC: Hydro Carbons

HCCI: Homogeneous Charge Compression Ignition

IMEP: Indicated Mean Effective Pressure

IVC: Intake Valve Close

IVO: Intake Valve Open

LTC: Low Temperature Combustion

MCE: Multi Cylinder Engine

OEM: Original Equipment Manufacturer

ON: Octane Number

PFI: Port Fuel Injection

PPC: Partially Premixed Charge

PRR: Pressure Rise Rate

RCCI: Reactivity Controlled Compression Ignition

RoHR: Rate of Heat Release

SOC: Start of Combustion

SCE: Single Cylinder Engine

SCR: Selective Catalytic Reduction