Extending the potential of the Dual-Mode Dual-Fuel combustion towards the prospective EURO VII emissions limits using Gasoline and OMEx

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Abstract

Dual-mode dual-fuel (DMDF) combustion strategy has been corroborated to be a potential combustion mode to achieve ultra-low NOx and soot emissions, as requested by the future regulations on internal combustion engines. In addition, the synthetic fuels have arisen in the last years for overcoming the problem of total emissions of CO_2 over the fuel life cycle. In the case of DMDF, poly-oxymethylene dimethyl ether (OMEx) has been found a promising alternative to diesel when combined to gasoline. In this sense, the OMEx-gasoline combination promotes ultra-low NOx and zero-soot emissions while maintaining the engine performance and acceptable levels of other regulated emissions. The main objective and novelty of this work is to experimentally evaluate the potential of this architecture to reach post-EURO VI NOx emissions levels at engine-out conditions by means of a dedicated engine calibration. Moreover, to evaluate the implications of using synthetic fuels for the European objectives in terms of CO_2 impact considering different driving conditions scenarios by means of a virtual vehicle model. The results show that the OMEx-gasoline DMDF engine can reach engine-out NOx emissions below 0.2 g/kWh with zero-soot emissions in all engine operating map and the impact on fuel consumption is lower than 4% with respect to conventional diesel operation. Also, the H2030 target of 30% CO_2 reduction can be achieved in a well-to-wheel base.

Keywords

Reactivity controlled compression ignition; Dual-fuel combustion; EURO VII; OMEx; emissions reduction

1. Introduction

The increasing demand and production of vehicles based on internal combustion engines (ICE) led to a concern on pollution. With the time, it became a matter of fact the impact of ICEs on air pollution and global warming, and it has been more than 40 years since government started a plan to reduce and regulate the impact of vehicle emissions by setting a maximum amount of certain contaminants that a vehicle can produce. This maximum allowed production rate has become more and more stringent over the years, especially for heavy duty-engines as they are considered an important source of emissions and contributors to the global warming [1].

Currently there are several standards depending on each country or region (the EPA 2010 in USA, the EURO VI in Europe or the China VI in China are some examples of different emissions regulations), but they follow a common trend despite the possible lag from one to another in their entrance into application. Most of them focus on the progressive reduction of a set of common pollutants that are considered as dangerous for our health or contributors to the air quality drop and global warming [2].

Specifically, the most important pollutants are: nitrogen oxides (NOx), that contribute to ground-level formation of ozone that can cause respiratory problems; carbon monoxide (CO), a specie that arises as product of an incomplete reaction, specially under rich conditions, which is a gas that when inhaled can enter the bloodstream and reduce the capacity to transport oxygen with the health issues that this carries; unburned hydrocarbons (HC), which are mainly a portion of the fuel that did not burn inside the cylinder or through the aftertreatment system (ATS) and once emitted as part of the exhaust gases contributes to the formation of ozone like NOx, the formation of particulates in the ambient and can cause several illnesses related to irritation or even cancer; and particulate matter (PM) or soot, that is basically solid carbon-based particles that can form during combustion and that have a very dangerous effect on human health, like premature death, cancer or cardiovascular problems. Moreover, carbon dioxide (CO_2) is one of the main products of complete combustion, like water (H_2O), and even if its impact on human health is not as severe as the other pollutants, it is (like water) a contributor to the greenhouse effect and therefore to global warming [3][4].

As can be inferred from the effects of these pollutants, it is necessary to reduce the amount produced by engines and strict regulations are needed. This work will focus on the European emissions standard for heavy-duty engines and its projection in the near future. Given that a heavy-duty engine can be used in more than one vehicle/configuration, testing is performed on engines alone, rather than on complete vehicles, and limits are expressed in terms of grams per kilowatt-hour (g/kWh) [5]. The evolution of the limits imposed by the European legislation are well known and defined until its latest and current version, the EURO VI [6], which does not only stablish the maximum production rate of the major pollutants but also dictates the procedure for the homologation of the vehicle and ensure the correct operation of the ATS, like the use of the World Harmonized Steady Cycle (WHSC) and the World Harmonized Transient Cycle (WHTC) among other measures [7][8].

Considering the progressive update of the European regulation and given that it has been seven years since the EURO VI came into force, after many updates and modifications it is normal that manufacturers and researchers, together with governmental entities start to talk about post-EURO VI regulation. This tentative EURO VII will be officially presented in 2021, but initial reports point out that the main concerns on this version will be the reduction of NOx emissions (a reduction of 50% from EURO VI limits at 0.2 g/kWh and a specific maximum for NO_2) and controlling particulate matter of small size (particulate numbering from 23 to 10 nm) [9]. Nonetheless, more updated studies may report different estimations of the future legislation, meaning that a definitive regulation still needs discussion and agreement between the different parties [10][11].

Another objective of the European Union that is interrelated to the severe limitation on vehicle emissions aims for a substantial reduction of the global CO_2 emissions. More specifically, the objectives are a 15% reduction by 2025 and a 30% reduction by 2030, both relative to the reported emissions in the current period from 2019 to 2020. These targets are also known as Horizon 2025 and Horizon 2030 respectively and aims for a reduction in the use of fossil fuels and a more responsive and ecological energy production and management [12].

With this scenario, manufacturers and researchers need to be able to reach such exigent limitations. Up to now they relied on combustion strategies based on high and ultra-high EGR (exhaust gas recirculated) rates (up to 50%) [13] and an ATS that could include a DOC (diesel oxidizer catalyst) to reduce HC and CO, an SCR (selective catalytic reducer) to reduce NOx, and a DPF (diesel particulate filter) to trap soot emissions [14]. This system can result to be complex and expensive, and sometimes even insufficient, as some manufacturers had to opt for a dual SCR architecture with a single-stage or double-stage urea dosing system to achieve the desired targets [15][16][17]. Furthermore, a complex ATS can also be difficult to integrate with the engine. The more elements it has, the greater the backpressure it will produce in the exhaust pipe, reducing the power output of the engine or increasing the fuel consumption to reach the desired specifications. To reach the projected EURO VII requirements, this strategy would need to evolve to a triple SCR architecture in the ATS as concluded in some reports [18], but this would imply a very high cost for the manufacturer and the fuel consumption reduction (and CO_2 emissions) would have to be compromised.

Even though this scenario seems to be very dramatic, the solution is not impossible. Instead of relying on the conventional diesel combustion (CDC), researchers conclude that it is necessary to move to what are known as low temperature combustion strategies (LTC). The LTC are combustion modes in which the combustion process is controlled to occur in the form of low-temperature flames that drastically reduce the amount of NOx produced, as well as premixing the fuel to reduce locally rich regions where soot can be produced. Among the different concepts of LTC there are some that are very promising like the HCCI (Homogeneous Charge Compression Ignition), TSCI (Thermally stratified Compression Ignition), PPCI (Partially Premixed Compression Ignition) or RCCI (Reactivity Controlled Compression Ignition), and all of them are potentially able to reach higher brake thermal efficiencies (BTE) that contribute to reduce fuel consumption and therefore, CO_2 emissions. Not only that, but they are also being investigated using alternative fuels with lower carbon content and higher oxygen content that enhance the oxidation of the fuel (lower soot) and produce less CO_2 [19].

There is a particularly promising LTC concept based on the RCCI strategy denominated Dual-Mode Dual-Fuel (DMDF) brought by CMT-Motores Térmicos in which the reactivity of the mixture is controlled by a near-homogeneous load of a low reactivity fuel (LRF) introduced through the port fuel injection system (PFI) and a stratified charge of a high reactivity fuel (HRF) introduced with the direct injection system (DI) [20].

The DMDF combustion concept has been tested and evaluated and the results show that it is able to achieve EURO VI engine-out NOx and soot levels using gasoline as LRF and diesel as HRF. In this way, it is possible to remove, or at least reduce the sizing, the SCR and the DPF from the ATS reducing manufacturing costs and a low impact of its implementation [22].

Additionally, the DMDF concept has been tested using alternative e-fuels like Methanol and synthetic diesel, with the most promising results obtained using OMEx (a mixture of polyoxymethylene dimethyl ethers, also known as POMDE or OME, ranging mainly from OME3 to OME5) as HRF, which has non-sooting properties as well as a lower CO_2 impact thanks to its cleaner production using CO_2 obtained from direct air capture (DAC) [23]. The OMEx fuel, in compensation for its smaller lower heating value (LHV) is able to burn more efficiently, augmenting the BTE and reducing the equivalent brake specific fuel consumption in addition to reaching EURO VI engine-out NOx and soot emission levels in all the engine map [24].

The objective of this work is to further investigate the potential of OMEx as HRF in the DMDF concept by reaching the proposed limits of NOx emissions for what will be the future EURO VII standard without the need of an SCR to reduce NOx emissions or a DPF for reducing exhaust soot. As far as the authors are aware, little or no work has been carried out to reach post-EURO VI levels of NOx and soot without the aid of a complex ATS, therefore the novelty of this work. Given the current uncertainty about what will be stablished as the EURO VII and after consulting with some manufacturers of the industry, in this work it is considered only the NOx limitation with a limit of 0.2 g/kWh on the WHSC and 0.23 g/kWh on the WHTC. To do so, a complete map calibration has been produced experimentally and a series of numerical evaluations have been carried out using the commercial software GT-Power to mimic real driving conditions. Finally, an analysis of the CO2 impact of this architecture is evaluated to corroborate the feasibility of this architecture to fulfil H2025 and H2030 targets.

2. Materials and Methodology

2.1. Engine characteristics

The experimental work has been performed on an 8L multi-cylinder production engine modified to include a new fuel supply circuit for the 6 PFI necessary to inject the LRF in the intake manifold, a low-pressure EGR circuit to confer more flexibility on the control over combustion process and emissions, and new pistons with an optimized geometry to reduce the engine compression ratio from 17.5:1 to 12.75:1 to ensure an adequate operation and stability at higher loads. Due to the presence of particulate solid matter in the EGR that goes through the low-pressure EGR circuit, it is necessary to install a DPF to protect the turbocharger.

Table 1. Engine characteristics.

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]	235 @ 2100 rpm
Rated torque [Nm]	1200 @ 1050-1600 rpm

2.2. Test cell description

For its safe and adequate operation, the engine is mounted on an AVL active dynamometer that is operated using its own software platform AVL PUMA Open from which the engine speed is commanded. The same platform is employed for recording averaged pressures and temperatures at different relevant points of the system, fresh air mass flow using an air flow meter at the intake and average fuel consumption using two gravimetric fuel balances used for LRF and HRF respectively.

Given the complexity of the system by having two EGR circuits and two injection systems, an external graphic interface was developed using LabView to use a NI PXIe 1071 board to command the DI system and PFI system separately, as well as different control parameters like injection pressure, positioning of different valves (EGR valves and back pressure valves) and VGT rack position. For referencing the injection timing, the LabView interface is connected to an encoder that works with a resolution of 0.2 crank angle degrees (CAD).

The external control platform also includes readings and recordings of different instantaneous magnitudes like the in-cylinder pressure of all six cylinder, as well as the measurements from external devices dedicated to emissions measurements. For regulated emissions, two measurement devices are used: a five-gas Horiba MEXA-7100 DEGR analyser [25] for measuring species like NOx, CO, HC O2 and CO2, and an AVL smoke meter 415 [26] for the smoke content.

Using all the measurements of instantaneous and averaged magnitudes, an in-cylinder thermodynamic analysis is performed in two separate steps. The first one is an online analysis implemented in the LabView control system based on apparent heat release to provide a more specific analysis of the combustion during the operation of the engine. The second step consists of a more detailed thermodynamic analysis using the recorded measurements. This post-processing is done using the CALMEC software developed at CMT, that consists of a set of pre-calibrated models to perform a detailed 0D analysis of the combustion [27].

A detailed schematic distribution of all the equipment included in the test cell can be found in Figure 1, and the specific models and relevant accuracies of the measuring devices are included in Table 2.

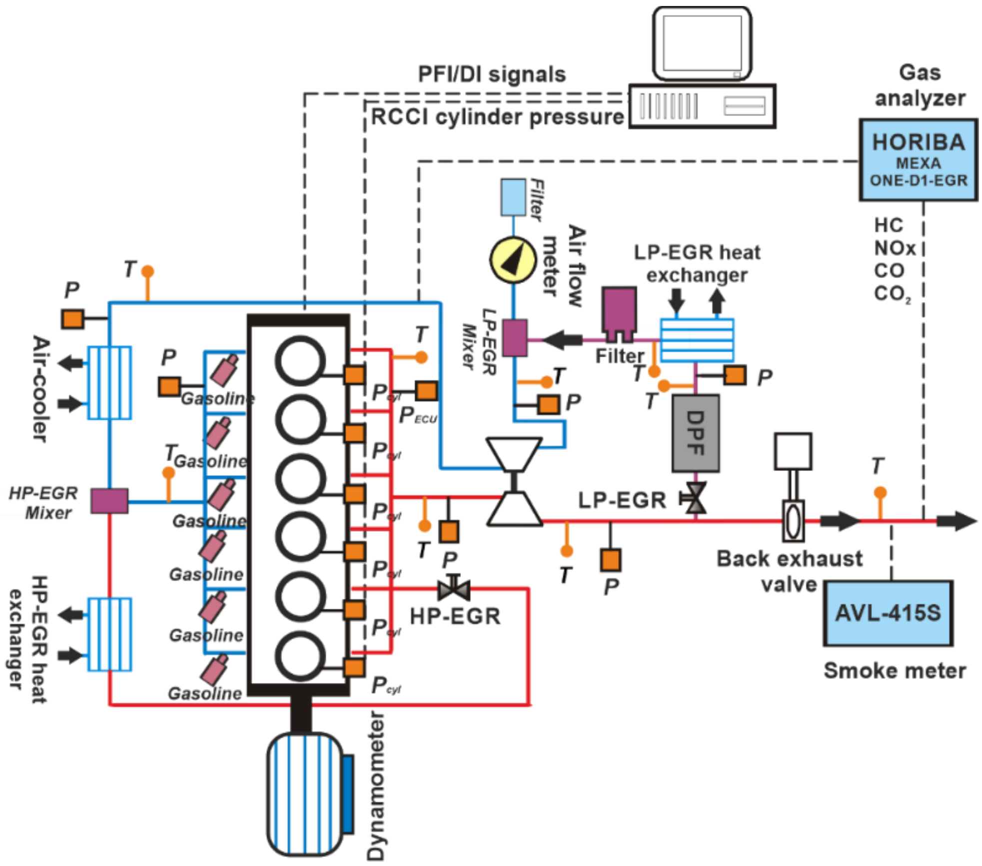


Figure 1. Test cell diagram.

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

Variable measured	Device	Manufacturer / model	Accuracy
In-cylinder pressure	Piezoelectric transducer	Kistler / 6125C	±1.25 bar
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 analyser	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%

2.3. Fuels and injection systems characteristics

In this work, OMEx is used as HRF and gasoline as LRF. The main physicochemical properties of these fuels are summarized in Table 3. The properties of commercial diesel are also included in Table 3 as some results will be compared to previous published results where only commercial diesel and gasoline were used.

Table 3. Physical and chemical properties of gasoline and the different high reactivity fuels evaluated.

	EN 590 diesel	EN 228 gasoline	OMEx
Density [kg/m^3] (15 °C)	842	720	1067
Viscosity [mm^2/s] (40 °C)	2.93	0.545	1.18
Cetane number [-]	55.7	-	72.9
Carbon content [% m/m]	86.2	84.4	43.6
Hydrogen content [% m/m]	13.8	15.6	8.82
Oxygen content [% m/m]	0	0	47.1
RON [-]	-	95.6	-
MON [-]	-	85.7	-
Lower heating value [MJ/kg]	42.44	42.4	19.21
Vapor pressure [hPa] (T=40 °C)	1-10	450-650	32

The charge formation and stratification control of the dual-fuel system is controlled by injecting the LRF through the PFI located at the intake of each cylinder to form a homogeneous base air-to-fuel mixture during the admission phase and then inject the HRF through the DI system during the compression stroke. By modifying the amount of each fuel, the start of injection (SOI) of the HRF and the injection pressure of the common rail to have a certain control over the mixing rate it is possible to obtain the desired load reactivity distribution. For a concise characterization of both injection systems the properties of both injector types are included in Table 4.

Table 4. Characteristics of the different injectors.

DI Injector		PFI Injector	
Actuation Type [-]	Solenoid	Injector Style [-]	Saturated
Steady flow rate @ 100 bar [cm^3/min]	1300	Steady flow rate @ 3 bar [cm^3/min]	980
Included spray angle [°]	150	Included Spray Angle [°]	30
Number of holes [-]	7	Injection Strategy [-]	single
Hole diameter [μm]	177	Start of Injection [CAD aTDC]	340
Maximum injection pressure [bar]	2500	Maximum injection pressure [bar]	5.5

2.4. Methodology for experimental evaluation

A systemic calibration procedure is applied to a total of 30 operating points distributed through all the operating range of the engine. In terms of engine speed, it ranged from 950 to 2200 rpm and load increase was divided into 10%, 25%, 50%, 75% and 100% load relative to the maximum nominal power output at each engine speed.

For the injection strategy, the DMDF combustion concept has been implemented by combining gasoline as LRF and OMEx as HRF modifying the injection timing and number of injections to control the fuel blend stratification level. As depicted in Figure 2, at low and medium loads, it implements a more homogeneous-like or partially stratified injection strategy, while at higher loads it implements a more diesel-like diffusive combustion where it performs better and can achieve higher power rates [21].

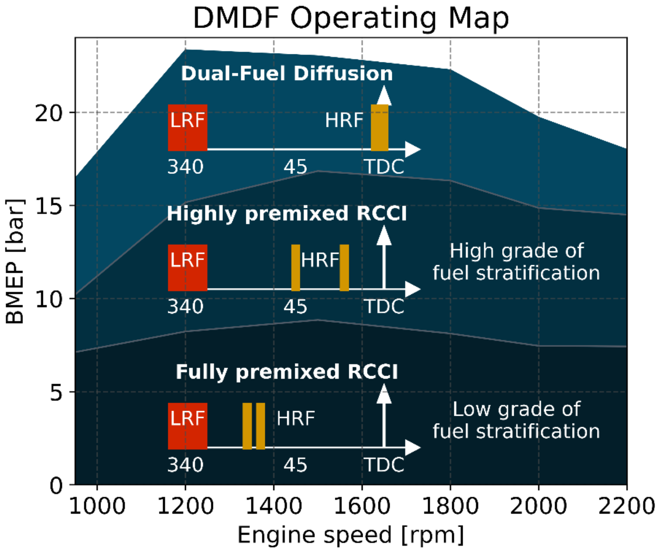


Figure 2. Conceptual injection strategy of the dual-mode dual-fuel combustion concept.

The starting point to develop the calibration aimed to reach NOx and soot levels beyond the EUVI limits (EURO VII) is based on the EU VI calibration developed in a previous study using gasoline as LRF and OMEx as HRF [24]. The constraints considered to avoid mechanical problems in the engine were the same that were used for the EU VI calibration (in-cylinder P_{max}< 180 bar, PRR< 17.5 bar/CAD).

To ensure that an adequate result is achieved in every measured point, a systematic procedure is followed to reach the proposed emissions limits while ensuring the target power output with an optimum specific fuel consumption. The step-by-step procedure mainly depends on adjusting the premixed energy ratio (PER), defined as the fraction of energy introduced through the PFI with respect to the total (Equation (1)), the EGR rate and the injection timing with an adequate tuning of the VGT position to ensure proper boosting levels. This methodology is very similar to the one presented in a previous study from the authors, whose main details can be found in Benajes et al. [20]. The summary of the calibration procedure is graphically described in the diagram presented in Figure 3.

$$\mathrm{PER}\ (\%) = \frac{\dot{\mathrm{m}}_{\mathrm{LRF}} \cdot \mathrm{LHV}_{\mathrm{LRF}}}{\dot{\mathrm{m}}_{\mathrm{LRF}} \cdot \mathrm{LHV}_{\mathrm{LRF}} + \dot{\mathrm{m}}_{\mathrm{HRF}} \cdot \mathrm{LHV}_{\mathrm{HRF}}} \cdot 100 \qquad (1)$$

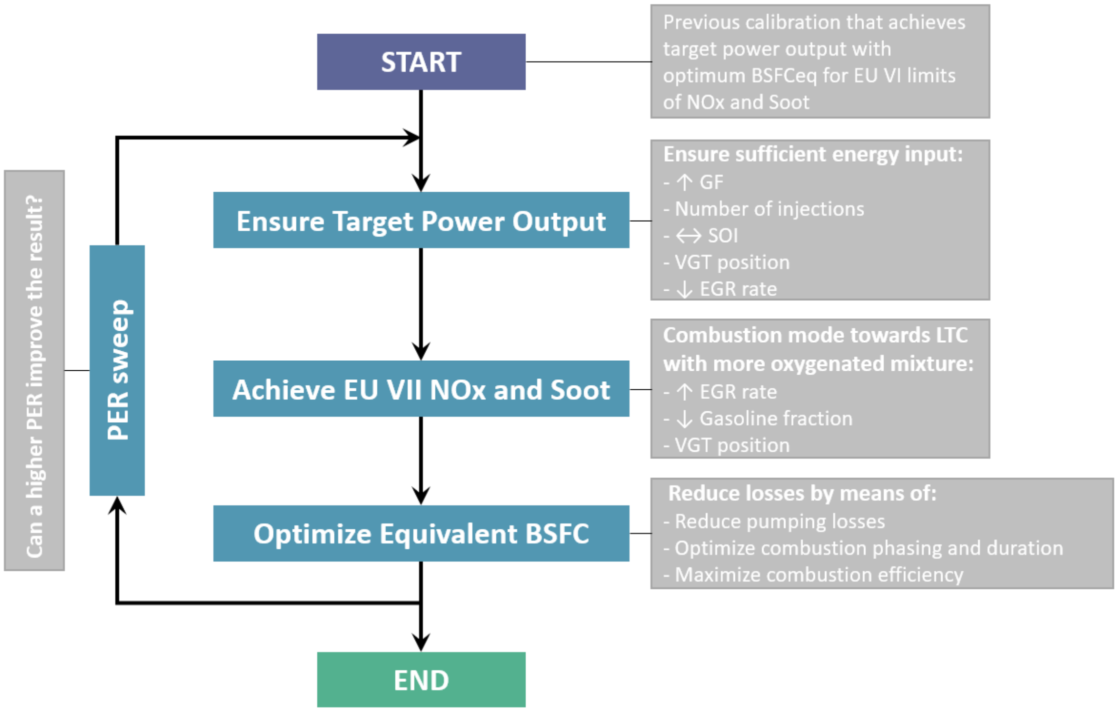


Figure 3. Summary of calibration strategy.

2.5. Methodology for numerical evaluation

An evaluation and comparison of the potential of the EU VI and EU VII calibration maps is carried out under different driving cycles using the software GT-Power from Gamma Technologies® [28]. The different modules and models included in this platform allow to build up a complete vehicle model.

The vehicle structure and weight are modelled to determine the required power to maintain the imposed velocity profile. The models include different factors as friction with the road, aerodynamic drag force of the vehicle and internal friction of the main components. The transmission included in the model converts the required axle speed and power required by the vehicle into the power and crankshaft speed that the engine must provide. From the experimental measurements, the engine is simulated on a map-based model in which the inputs of power and engine speed required from the vehicle dictate the instantaneous emissions and fuel consumption.

As the medium-duty engine tested is used for transportation applications, the vehicle model setup includes a truck model based on the Volvo FE 350 with a twelve-gear gearbox. Experimental data of this vehicle was provided for the validation of the numerical model and the model was dimmed capable of predicting total fuel consumption with an accumulated error lower than 5% even for worst case scenarios as shown in Figure 4, while emissions could have slightly higher dispersion depending on the species [29].

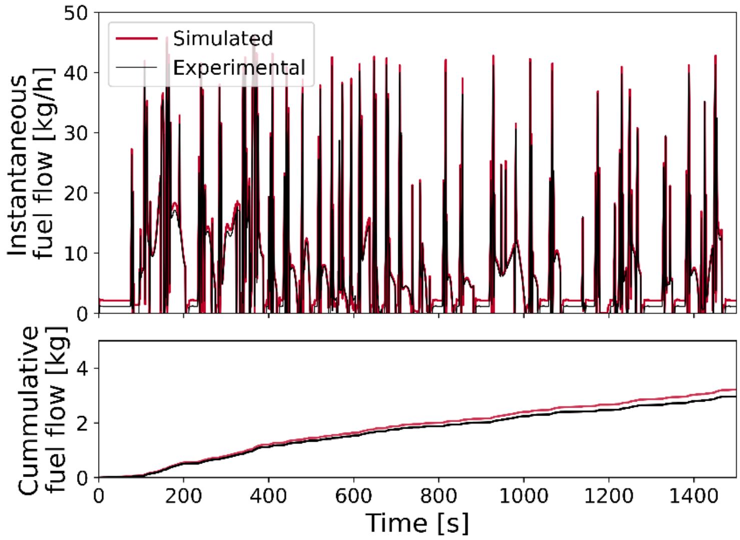


Figure 4. Validation of the vehicle model against experimental data. Obtained from [22].

The final model structure is shown in Figure 5, and more details can be found in [22]. Additionally, the main parameters that define the model are summarized in Table 5, including characteristics from the truck and the gearbox.

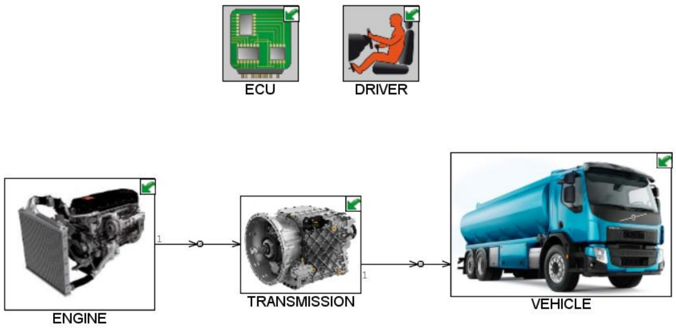


Figure 5. General structure of the vehicle model.

Table 5. Vehicle model main features.

Vehicle model characteristics	
Engine displacement volume [cm^3]	7700
Engine control	Map-based from experimental results
Vehicle mass [kg]	7035
Cargo mass [kg]	8982.5
Frontal area [m^2]	6.89
Tires size [mm/%/inch]	295/80/22.5
Number of axles	3
Number of wheels	10
Vehicle wheelbase [m]	5
Final Drive Ratio [-]	3.08
Gear ratio (from 1st to 6th gear) [-]	14.94, 11.73, 9.04, 7.09, 5.54, 4.35
Gear ratio (from 7th to 12th gear) [-]	3.44, 2.7, 2.08, 1.63, 1.27, 1

Regarding the vehicle velocity profile imposed on the simulation, four different driving cycles are used, including three realistic driving cycles obtained from measurements of real truck routes covering the scenarios of urban areas, highway, a combination of both, and the world harmonized vehicle cycle (WHVC) used for homologation[29]. The instantaneous altitude and vehicle velocity profiles are included in Figure 6, and more information can be found in [31]. For a more extensive evaluation and comparison of the different calibrations, the truck will be evaluated with payloads of 0%, 50% and 100% of its maximum allowed payload, and the combination of WHVC at 50% payload is taken as reference corresponding to the homologation conditions [30].

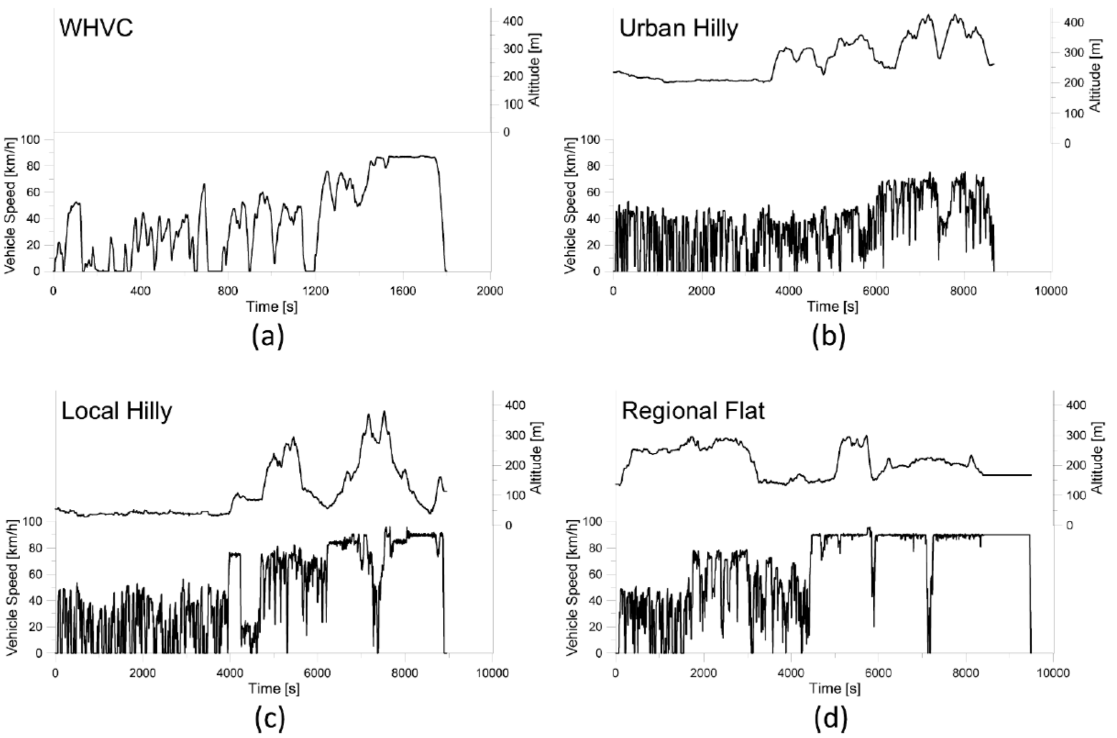


Figure 6. Instantaneous profiles of the different driving cycles used for the numerical evaluation [31].

3. Results and discussion

This section contains the results of the EURO VII calibration obtained following the procedure described in subsection 2.4, as well as the differences relative to the previous DMDF calibration aimed to reach the EURO VI limits for NOx and soot emissions at engine-out levels using OMEx-Gasoline. For a clearer understating of the results, the changes in the main calibration parameters that determine the combustion mode are presented first, followed by the changes of relevant combustion parameters. After this, engine performance and engine-out emissions are presented and discussed. At the end, some notes on the limitations found on the applicability of this concept are included.

3.1. Calibration parameters

As it was mentioned in the explanation of the calibration methodology (subsection 2.4), one of the most important parameters for the DMDF calibration is the PER, which provides a near homogeneous fuel-air mixture that serves as a base to later promote an in-cylinder reactivity stratification by means of the HRF injection. In LTC modes, the more homogeneous the mixture is, the greater the NOx reduction is as local hot regions are avoided where high-temperature reactions enhance the formation of nitrogen oxides [32]. Given this basis, it is predictable that a higher PER would be preferred on the DMDF combustion mode calibration to reach the EURO VII NOx emissions limits. In Figure 7a, it can be seen how the engine power output is increased mainly based on the LRF input to maintain a LTC with low NOx production, therefore PER increases with the load demand up to a medium load. Beyond this medium load region, the pressure gradients characteristic of a premixed and almost-homogeneous combustion are greater than the imposed mechanical safety limits and a lower PER is induced, making the calibration more dependent on the HRF, which is used to better control the combustion start, its duration and its phasing. Normally, this would mean that power is obtained at the expense of greater NOx and soot emissions, but the usage of OMEx as the HRF will result in certain benefits in this matter as will be reported later. In perspective with the EURO VI calibration, a clear trend towards more premixed combustion is obtained in all the engine map, having the greatest PER increase on the medium-high load region as observed in Figure 7b.

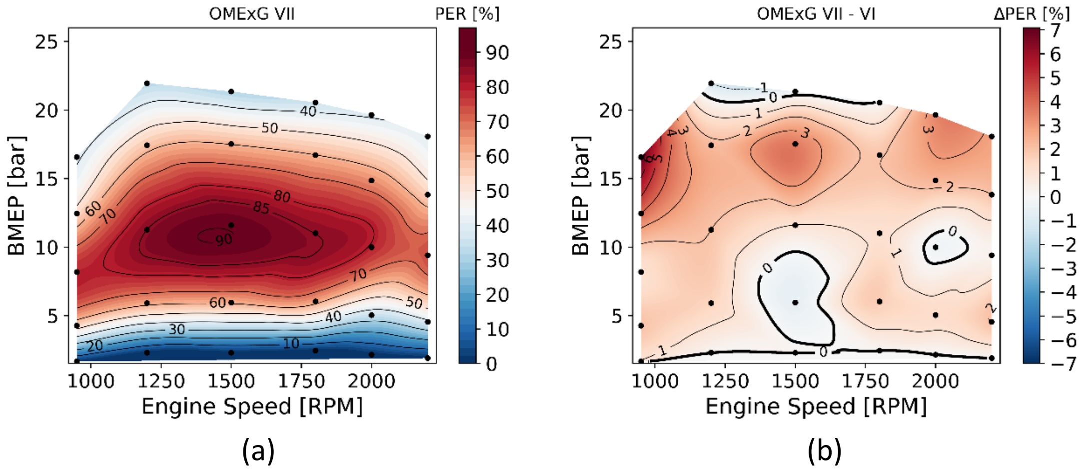


Figure 7. PER distribution along the engine operating map (a) in absolute value for the EU VII calibration and (b) difference with the previous EU VI calibration.

The other most common variable used to reduce the in-cylinder temperatures to promote lower NOx production rates is the EGR. In Figure 8a, it is shown how the greatest EGR rates are used at very low loads together with lower PER (Figure 7a) to control the combustion duration of the HRF and the in-cylinder peak temperatures. At medium loads, a moderate-to-high EGR levels are used to reduce the pressure gradients arising from the premixed combustion, and finally, the lowest EGR usage is found at full load, where the fuel-to-air mixture results to be relatively very rich and an excessive amount of EGR would result on quenching of the combustion, lower power output and excessively delayed combustion. Nonetheless, the global EGR percentage is relatively greater than that in other combustion modes thanks to the flexibility of having both EGR circuits. This fact, together with the flexibility conferred by the dual-fuel combustion system, ensures ignition and adequate progression of combustion despite the high EGR in the cylinder. This allows to achieve higher EGR levels maintaining moderate intake temperatures and oxygen content and still have a sufficiently reactive combustion process to obtain the desired power output and fuel consumption with acceptable emission levels. Contrary to the trends shown for the PER, the EGR rate difference between both calibrations does not show a clear trend in Figure 8b, with only just slight adjustments done depending on the operating point. It is necessary to remark that when using the DMDF concept aiming to reach the potential EURO VII engine-out NOx emissions levels, the amount of total mass of nitrogen oxides is very small, and this is very sensitive to EGR rates. The generally greater PER combined with small EGR adjustments is sufficient to adapt to in-situ conditions and reduce specific NOx emissions by halve of those obtained with the EURO VI calibration.

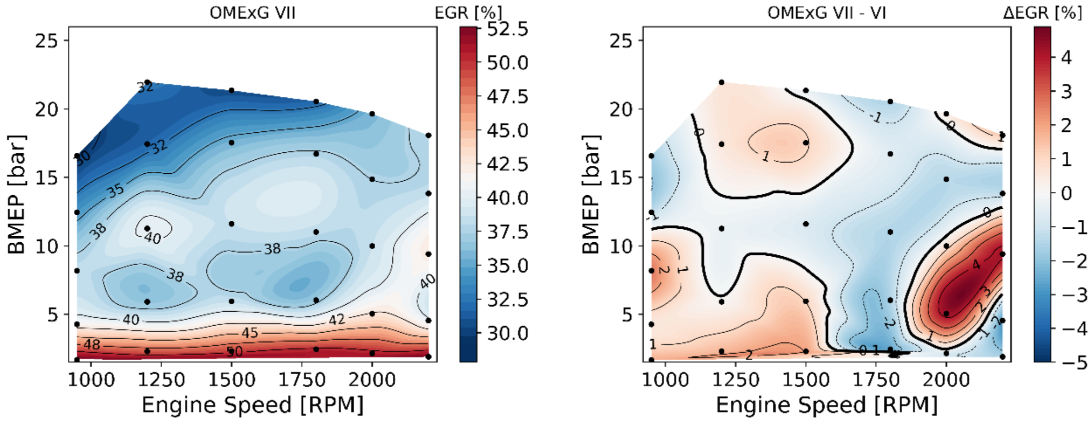


Figure 8. EGR level distribution along the engine operating map (a) in absolute value for the EU VII calibration and (b) difference with the previous EU VI calibration.

Given the changes introduced in important parameters of the dual-fuel combustion like the PER and EGR, one can suspect that significant adjustments on the injection settings would be needed. Figure 9 shows the differences in the SOI between both calibrations. Only at very low loads some minor adjustments of the injection timing were done to ensure that the combustion phasing was maintained, and the engine performance and power output were not affected by the changes in PER and EGR. More specifically, these changes in the injection configuration appear only where important changes in the EGR tuning appear or at very low load, where the PER is zero as only HRF is being injected. This proofs that using OMEx as a HRF enables to achieve prospective EURO VII levels of NOx and soot with small changes in the calibration.

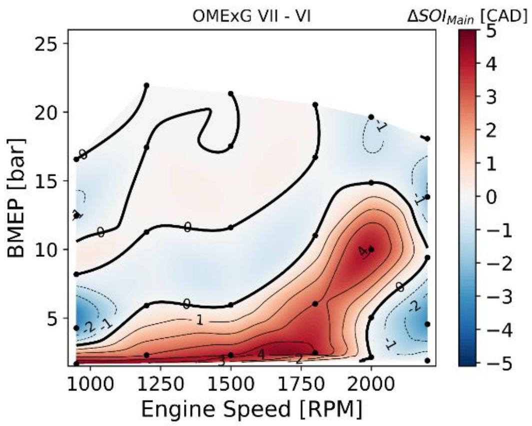


Figure 9. Difference in SOI of the main injection between EI VII calibration and the previous EU VI calibration.

3.2. Combustion characteristics

The capability of reaching lower NOx production levels is a direct consequence of the combustion properties and the in-cylinder thermodynamic evolution. As already explained, by having a more homogeneous mixture and lower temperatures in the combustion chamber, hot local regions, where NOx production rates are greater, are avoided. This low temperature combustion mode is easily reached on the DMDF combustion mode with the use of the low pressure EGR circuit, which allows to have very high EGR rates with a low impact on the turbine performance. By increasing the low pressure EGR fraction, it is possible to maintain the boosting levels with a colder intake temperature (the low pressure EGR circuit includes a heat exchanger to cool down the exhaust gas to maintain the volumetric efficiency of the engine and the compressor), as shown in Figure 10.

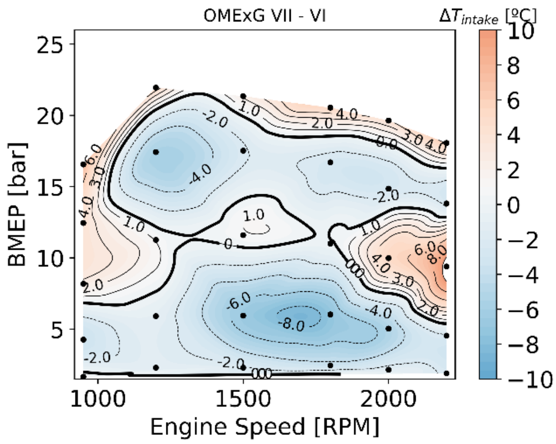
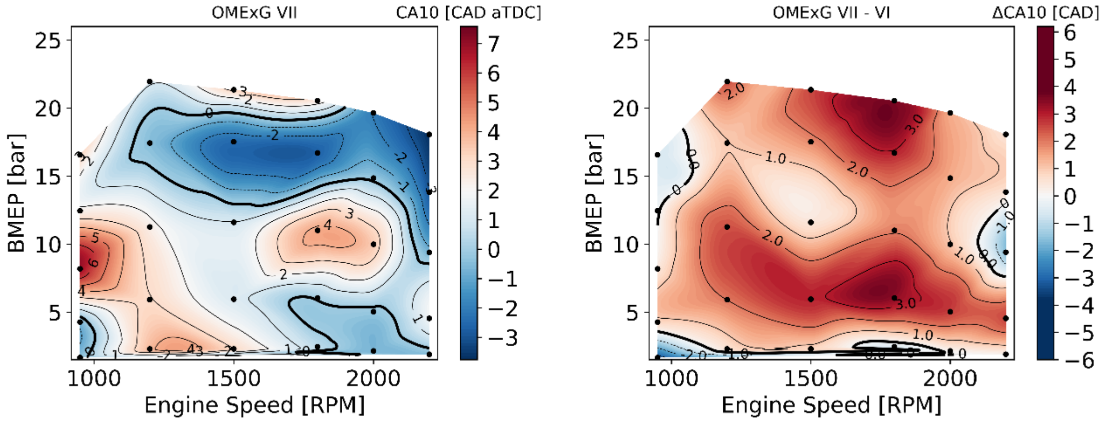


Figure 10. Difference in intake temperature between the EU VII calibration and the previous EU VI calibration.

The use of low pressure EGR allows to have lower in-cylinder temperatures that enable to increase the PER and have a more homogeneous mixture without surpassing the imposed pressure gradient limit, as was presented in the previous section. The use of higher PER with lower temperatures results, generally, in a more delayed autoignition and slower combustion process due to the lower reactivity of the in-cylinder mixture. These characteristics can be confirmed looking at Figure 11 and Figure 12, which show the CA10 (crank angle degree at which 10% of the fuel has been burned) as an indicator of the start of combustion (SOC) and CA90-CA10 as indicator of the combustion duration.

In Figure 11a, the evolution of the CA10 along the engine operating range clearly shows two differentiated regions. The first one covers up to 60% load, where a more premixed combustion strategy is used with varying stratification levels depending on the operating point. In this region, the fuel load is introduced earlier in the compression stroke and it will slowly react until the combustion process starts on its own, which results in a longer ignition delay. At higher loads, a more diffusive combustion strategy is employed, with the HRF injection occurring nearer to the top dead center (TDC). The turbulence and kinetic energy induced through the spray together with the higher local concentrations of the HRF act as an enhancer of the start of combustion, forcing earlier ignitions, which explains how at high loads the combustion consistently starts slightly before the TDC [33][34][35].

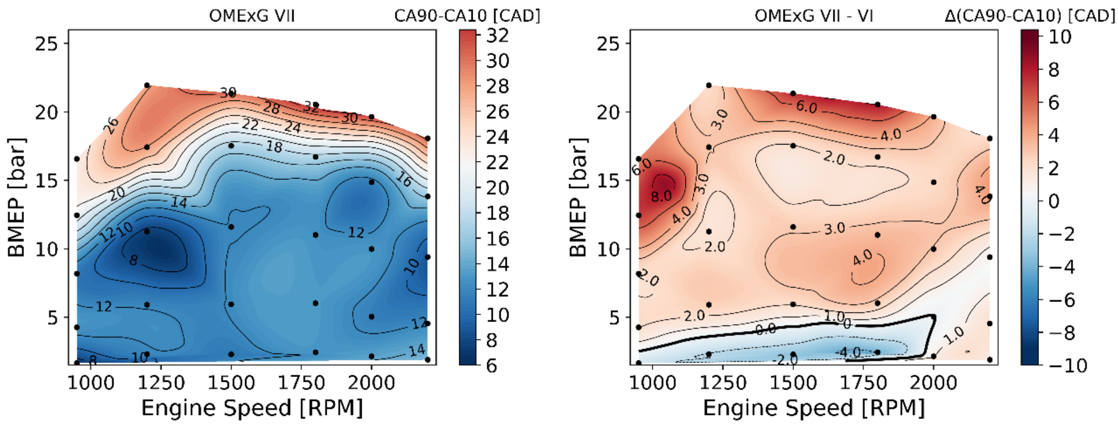
On a thermo-chemical basis, the variable with the greatest impact on the start of combustion is the temperature. The direct consequence of having colder in-cylinder conditions is that the SOC is delayed. In Figure 11b, it can be seen how the combustion process has been delayed between 1 to 3 CAD as a general trend on the complete operating map.



(a) (b)

Figure 11. Start of Combustion (CA10) distribution along the engine operating map (a) in absolute value for the EU VII calibration and (b) difference with the previous EU VI calibration.

The combustion duration is dependent on the reactivity of the mixture that at the same time depends on the temperature and the in-cylinder fuel mixture composition. By increasing the PER of the mixture, the LRF takes more relevance on the combustion process and the reactivity of the global in-cylinder mixture is lowered, which is the main contributor of the longer combustion duration. Additionally, because of the delayed SOC, combustion takes place later in the expansion stroke and temperatures are much lower as combustion advances compared to the previous calibration. As shown in Figure 12, these two coupled effects of lower reactivity and lower temperatures prolong the combustion duration by 3 to 6 CAD in all the operating map as compared to the EUVI calibration. It can be observed that the greatest increase of the combustion duration appears at full load conditions, where a more diffusive combustion strategy is utilized.



(a) (b)

Figure 12. Combustion duration distribution along the engine operating map (a) in absolute value for the EU VII calibration and (b) difference with the previous EU VI calibration.

To complete the analysis of the combustion properties under the presented calibration for the prospective EURO VII NOx limit, the CA50 is analysed. This parameter can provide a global overview of not only where the combustion occurs but also on the impact that this will have on the engine performance and the fuel-to-work conversion efficiency. Given that the starting point of this work had an already optimized calibration for which the CA50 was already optimized under certain constrictions, the possible deviations from the EU VI combustion phasing can provide a significant improvement or penalty on the engine performance and fuel consumption. In Figure 13, it can be appreciated how the delayed and slower combustion process with the EU VII calibration translates on a later combustion phasing compared to the EU VI results. A later combustion phasing means that the indicated efficiency of the engine is lowered and a certain penalty on engine performance must be expected. Nonetheless, the amount of said penalty cannot be directly extrapolated from this observation as for every operating condition, the optimum CA50 is different when changing the boundary conditions as will be seen in the following section.

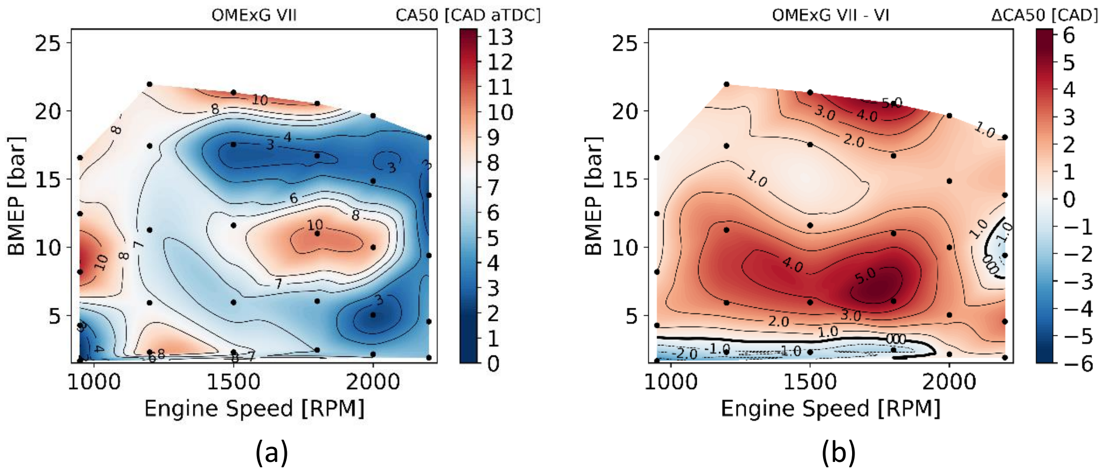


Figure 13. Combustion phasing (CA50) distribution along the engine operating map (a) in absolute value for the EU VII calibration and (b) difference with the previous EU VI calibration.

Lastly, Figure 14 shows a direct comparison between the rates of heat release (RoHR) under different operating conditions utilizing both calibration maps to give a better understanding on how the combustion process has been affected. Three points have been selected to be representative of the three different combustion modes of the DMDF strategy: 50% load at 1800 rpm, 75% load at 1800 rpm and 100% load at 1800 rpm.

As shown in Figure 14, at 50% load the DMDF combustion strategy still allows for a highly premixed combustion, resulting in a near homogeneous charge with very light fuel stratification that produces a Gaussian-shaped heat release. At 75% engine load, the richness of the in-cylinder mixture can result in high pressure gradients or detonation if it is homogeneously mixed. Thus, to control the burning rate, a higher degree of fuel stratification is induced with more delayed injections of the HRF. Under these stratified conditions, different burning rates correspond to different mixture reactivities, and said trend is observed on the heat release rate slope. At the full load condition, it is very difficult to maintain the pressure gradient constrain and still obtain the target power output with a premixed combustion, so a diffusive combustion process is promoted. The main characteristics of this combustion are noticeable on its corresponding RoHR curve, where a fraction of the blend ignites as a premixed combustion, and then the burning rate is kept with the constant supply of fuel. At any of these conditions, it is appreciable the same trend when moving from the EURO VI to the new prospective EURO VII calibration: the start of combustion takes place slightly later as a consequence of the lower temperatures and the slopes at the rising side of the curve are considerably lower due to the higher content of LRF. Also, the rate of heat release peaks are lower, and the maximum temperatures reached during combustion are sensibly lower for the proposed EURO VII calibration to avoid high NOx production rates.

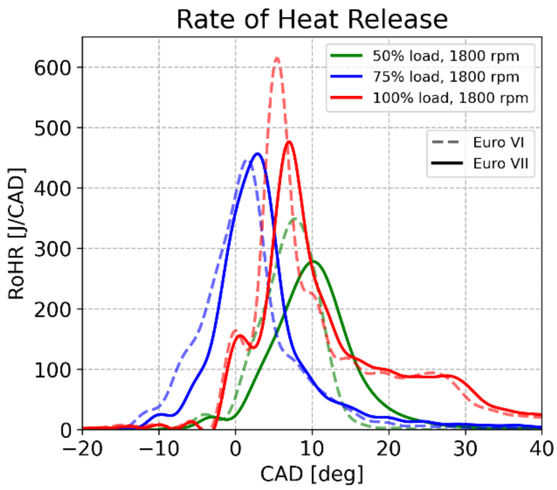


Figure 14. Representative rates of heat release of the three characteristic combustion modes in the DMDF concept for the EURO VI and EURO VII calibrations.

3.3. Performance and emissions in steady-state conditions

In the analysis of the OMEx-Gasoline combustion characteristics under the prospective EURO VII calibration, some notes were already done about the impact that a delayed and longer combustion would have on performance. More specifically, the first performance parameter affected by this is the gross indicated efficiency (η_i or GIE) that is defined as the capacity to obtain work from the energy stored in the fuel when it burns during the closed side of the piston cycle. Figure 15 shows the change in gross indicated efficiency between both OMEx-Gasoline calibrations, making clear the relationship between combustion duration, combustion phasing and the combustion performance. It can be observed how in the regions where combustion duration was longer and the CA50 was delayed substantially from the reference of the EURO VI calibration, the GIE is reduced significantly. Only at very low loads where only HRF is used a small benefit in performance can be noticed. This is consequence of the higher reactivity and low cetane number (CN) of OMEx that allows to have a good combustion phasing and burning rate despite the lower pressures and temperatures in the cylinder only by adjusting the injection settings without damaging the combustion start or the production of emissions like NOx or soot.

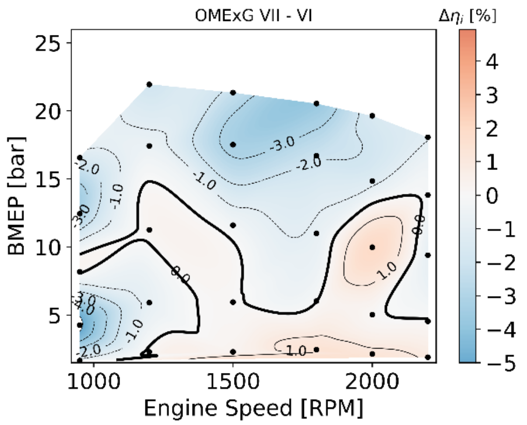


Figure 15. Difference in gross indicated efficiency between the EURO VII calibration and the EURO VI calibration.

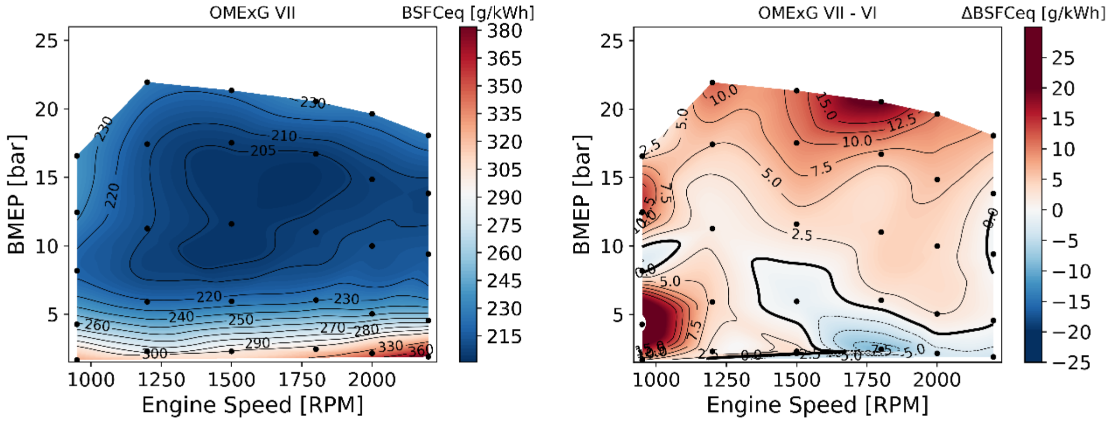
To this reduced GIE it is still necessary to add the effect of pumping losses and mechanical losses to obtain the BTE that is defined as the amount of useful power obtained from the fuel energy. This thermal efficiency can also be expressed as a specific fuel consumption in the form of brake specific fuel consumption (BSFC), but for a fairer comparison in terms of specific fuel consumption when changing the PER with different fuel combinations (the LHV of the global fuel mixture is modified) the equivalent brake specific fuel consumption ($BSFC_{eq}$) was defined as in Equation (2), where the fuel consumption is normalized taking into account the energy density referred to a reference fuel. In this case, the reference LRF is gasoline, and the reference HRF is diesel.

$$\mathrm{BSFC_{eq}}\ (\mathrm{g/kWh}) = \frac{\dot{m}_{\mathrm{LRF}} \cdot \frac{\mathrm{LHV_{LRF}}}{\mathrm{LHV_{LRF,ref}}} + \dot{m}_{\mathrm{HRF}} \cdot \frac{\mathrm{LHV_{HRF}}}{\mathrm{LHV_{HRF,ref}}}}{\mathrm{P_{engine}}} \qquad (2)$$

The final distribution if this equivalent BSFC as well as its increment relative to the EURO VI calibration are represented in the maps of Figure 16. The trends shown in Figure 16b are the same obtained for the GIE, meaning that the drop in GIE has greater relevance that the changes in pumping caused by the modification of the EGR distribution or the mechanical losses. Certain reduction in fuel consumption is noticeable at very low loads where the use of HRF allows for a better optimization maintaining low NOx emissions, but significant increment in specific fuel consumption appears at the lowest tested engine speeds and at full load conditions.

This significant reduction in performance at low engine speeds has been attributed to limitations of the turbocharger due to very low energy available for the turbine and the turbocharger is not able to provide enough boosting pressure. It was concluded that a redesign of the turbocharger with a smaller turbine and a bigger compressor could result in better performance at low engine speeds. At high load conditions the limitation was found to be the fuel pump for the DI system. Due to the high fuel flows required, the duration of the injections was too long, and combustion duration was excessive even if the pump was working at its maximum rail pressure of 2000 bar. The end of the injection is well into the expansion stroke which hinders the extraction of work from a portion of fuel injected at the end of the injection process. A fuel system with higher injection pressure capacity could reduce injection duration and provide some improvement at this region.

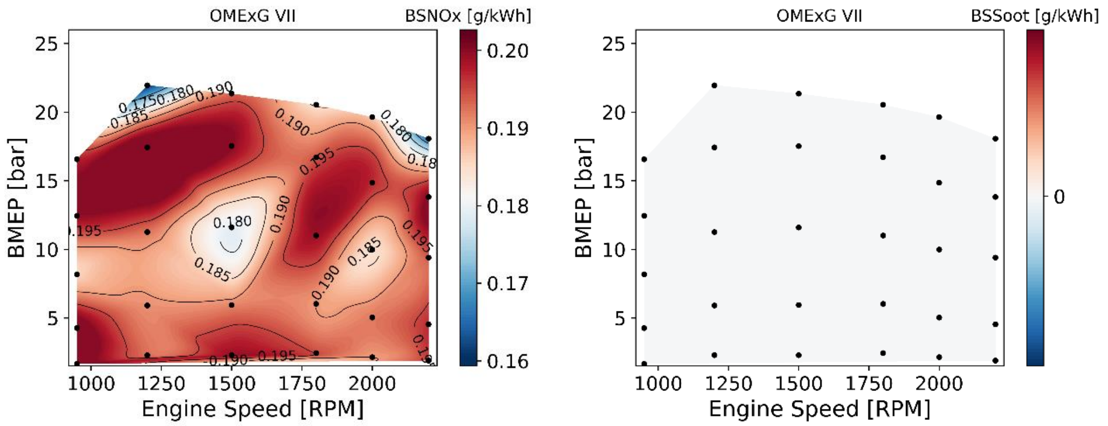
The increment on fuel consumption at the medium-load region ranges from 1% to 4%, which could be an acceptable penalty given that NOx emissions are halved. To reach a conclusion on whether this penalty on fuel consumption along the operative range of the engine is acceptable or not, real driving conditions must be considered.



(a) (b)

Figure 16. Equivalent BSFC distribution along the engine operating map (a) in absolute value for the EU VII calibration and (b) difference with the previous EU VI calibration.

In terms of emissions, Figure 17 clearly demonstrate that both targets of brake specific NOx emissions (BSNOx) below 0.2 g/kWh and brake specific soot emissions (BSSoot) below 0.01 g/kWh are fulfilled in all the engine map. The use of OMEx has already reported important benefits in achieving zero-soot combustion thanks to its oxygen content and its molecular structure that enhances the direct oxidation of carbon atoms instead of allowing the formation of polycyclic aromatics that lead to soot [36]. This condition allows to increase the EGR rates or PER without much concern about soot formation and adds great flexibility to the system to achieve the desired level of NOx emissions. The fact that the use of OMEx on the DMDF combustion mode permits to reach post-EURO VI levels of NOx without soot emissions for any engine operating condition opens the possibility of simplifying the ATS of the vehicle by removing the SCR in charge of NOx reduction and could potentially compensate the costs of adding a low-pressure EGR circuit.



(a) (b)

Figure 17. NOx emissions (a) and Soot emissions (b) of the DMDF OMEx-Gasoline calibration for the proposed EURO VII limits.

Of course, if the proposed levels of EURO VII for NOx and soot are reached by means of delaying the combustion process and worsening the combustion efficiency itself, there must be a penalty in terms of unburned hydrocarbons and CO. The long combustion process enters more into the late part of the expansion stroke where pressure and temperature go down to a non-flammable region leaving a part of the fuel that does not burn completely or reactions that do not finish adequately. In the case of CO, the oxidation of CO is a reaction several times slower than that of other important species that appear during the combustion process. By delaying the combustion, this reaction has less available time to oxidize the CO before the end of the cycle, leaving great amounts of CO in the exhaust gases. Additionally, the increase in PER also increases the amount of fuel that goes into local regions where quenching of the combustion process can occur, like the piston crevices, producing an additional increase on CO and HC. In Figure 18a can be appreciated how the increase of unburned hydrocarbons is greater at lower load conditions where low temperatures and high amounts of EGR are utilized, while the increase of CO shown in Figure 18b does not only occur at low load conditions but also in those zones where combustion duration is longer and unfinished reactions are prone to appear.

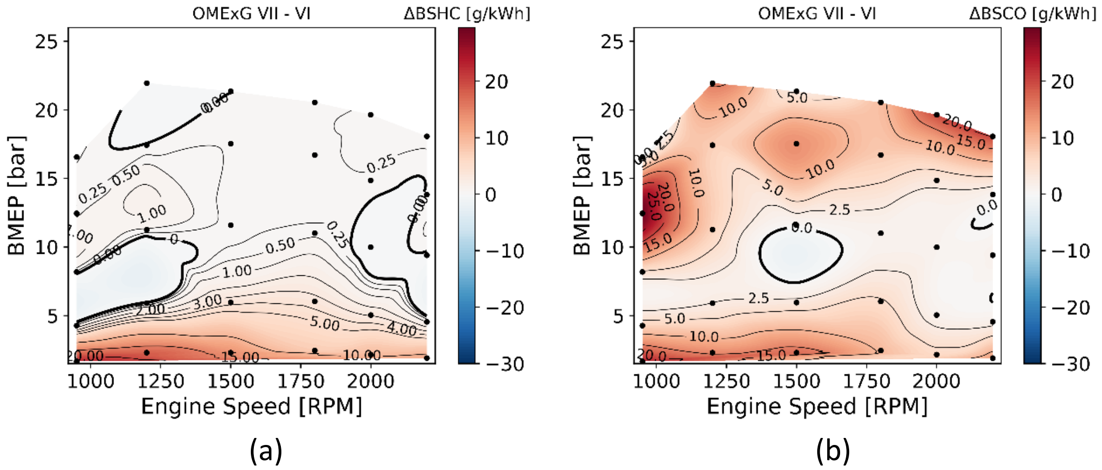


Figure 18. Unburned hydrocarbons emissions (a) and CO emissions (b) increments of the DMDF OMEx-Gasoline calibration for the proposed EURO VII calibration with respect to the previous EURO VI calibration.

The EURO VI calibration already produced emissions of CO and HC over the limit stablished, and the proposed EURO VII calibration has an increase over the previous calibration that can reach near 20 g/kWh. These levels of HC and CO are extremely high and very far from the homologation limits, and the necessity of an ATS able to deal with these amounts of emissions is of utmost importance. Some studies carried out in the past showed that conventional DOC systems for diesel engines were able to reduce great amounts of these contaminant and have a total tailpipe emission under EURO VI limits [37], but the great increase in the total mass of CO and HC as well as the change in composition of the exhaust gases as a consequence of using an oxygenated fuel like OMEx can be a valid argument to not accept that a conventional DOC is still a valid solution. It is necessary to carry out a validation of the conclusion reached in the past under the new boundary conditions and evaluate if it is necessary to size a new DOC accordingly to the increase in the amount of pollutants that it has to work with.

4. Impact on real application scenario

In order to evaluate the impact of the variations on the engine map during driving conditions, a vehicle model has been used to simulate four different driving cycles under three different cargos (0%, 50% and 100% load).

The simulations have been carried out for the four engine calibrations: OEM conventional diesel combustion, EURO VI DMDF diesel-gasoline calibration, EURO VI DMDF OMEx-Gasoline calibration and EURO VII DMDF OMEx-Gasoline calibration. The results are shown case-by-case as a percentage variation always taking the conventional diesel calibration as a reference. In all cases, the results are compared to the EURO VI limits for transient operation, and the EURO VII limits considered for NOx emissions.

4.1. Analysis of pollutants and fuel consumption

Figure 19 shows the reduction of NOx emissions during the different driving conditions and under the different levels of cargo load. The results with a black circle are those that reach global emissions under EURO VI limits, and a red circle was used for the cases fulfilling the EURO VII limit proposed for this work. It can be seen how the diesel-gasoline calibration can perform adequately at almost every case with low or medium cargo load. However, in those cases where it is necessary to reach the higher power output region in the map, where a more diffusive combustion strategy is employed, it is not able to reach the 0.46 g/kWh that the EURO VI normative requires. In the case of OMEx, both calibrations succeeded in reaching their respective limits of NOx emissions at any cargo load or driving cycle. This grants the possibility of removing the SCR from the ATS an reduce the engine manufacturing costs, as well as in the urea consumption for its operation.

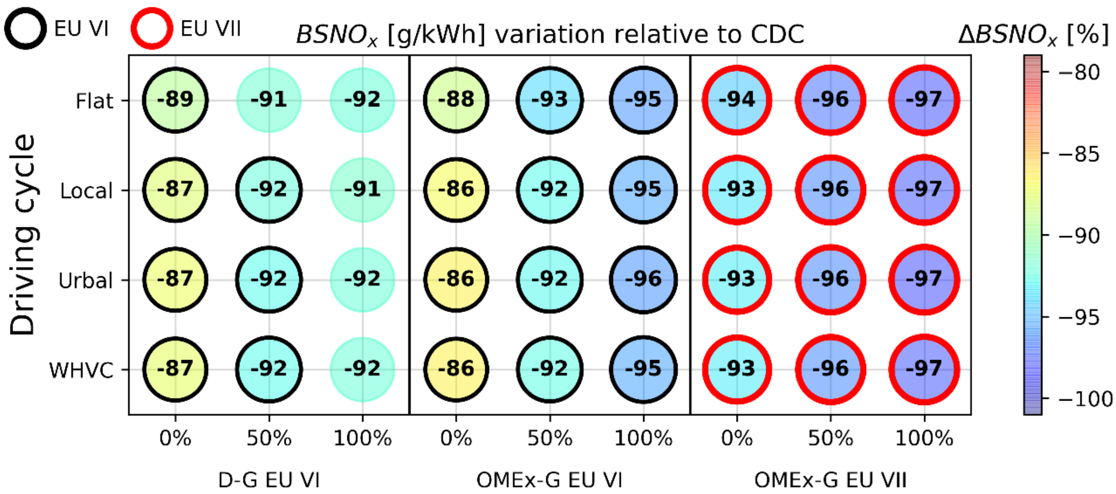


Figure 19. Variation in BSNOx emissions under different driving conditions.

Something similar occurs with the soot emissions reported in Figure 20. In this case, it was considered that no substantial reduction in the soot limit would be applied in the incoming EURO VII normative and only the EURO VI limit was applied in the form of a black circle in those cases fulfilling this limitation. The diesel-gasoline calibration reported benefits versus CDC in terms of soot emissions only in the low-load region of the map, therefore conventional fuels are only able to reach EURO VI limits at conditions where not high engine load is requested frequently. For OMEx it is trivial to reach any of the imposed limits thanks to its zero-soot properties. Nonetheless, the possibility of some soot emissions appearing under real transient operation or the presence of particles not measured by the smoke meter like ultra-fine particles needs a more dedicated study.

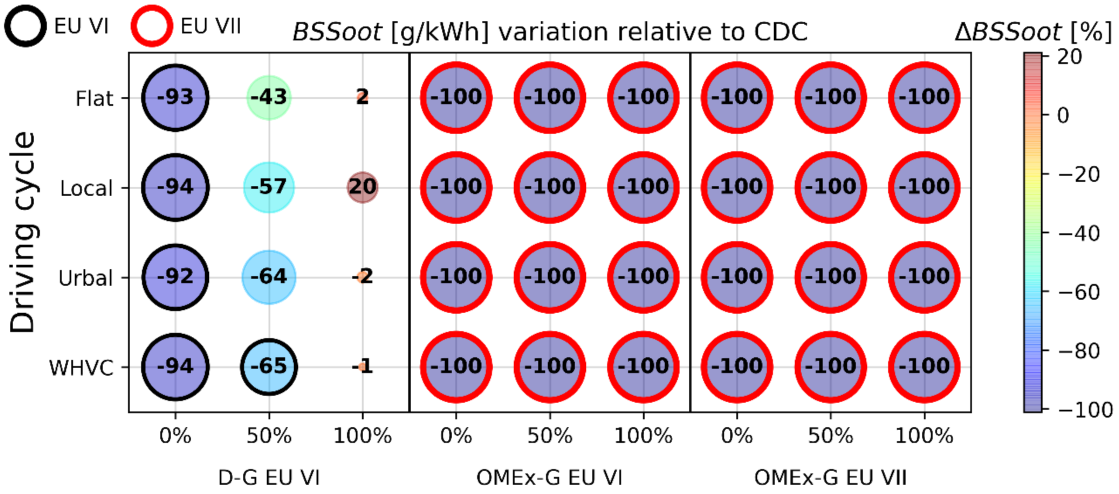


Figure 20. Variation in BSSoot emissions under different driving conditions.

For the case of global emissions of HC and CO during driving conditions, it important to consider that the reference calibration for CDC is for the stock engine with a CR of 17.5:1, while the DMDF calibration have been obtained for the modified configuration of 12.75:1 of CR. This reduction in CR implies that the high temperatures at which the HC is almost completely burned are no longer available and the low in-cylinder temperature with the possible lower combustion efficiency results in an unavoidable increase of HC emissions. Also, the EGR rates have been more than doubled after including the low-pressure EGR circuit and the high EGR rates significantly increases the CO emissions compared to the CDC configuration. For these reasons it is important to account for the increases in the emissions of these pollutant to see the impact that this might have in the consideration of the necessary ATS.

Figure 21 and Figure 22 depict the increase in BSHC and BSCO respectively. For any of the DMDF scenarios evaluated in this work would be possible to reach EURO VI levels without including an ATS in the system. Depending on the driving cycle, the truck payload and the fuel combination of the DMDF strategy, the emissions of HC range from 3 to 80 times greater than that of the CDC calibration, while for CO they range from 7 to 30 times greater than CDC. These numbers can be very alarming, but these relative numbers are compared against a calibration that already fulfilled EURO VI emissions accepted levels of HC and CO by far. For the diesel-gasoline DMDF it was already demonstrated that a conventional DOC could deal with these high levels of combustion inefficiencies and fulfil EURO VI in most of the driving scenarios [37]. If this fact is introduced in the analysis, and it is assumed that said performance of the DOC can be maintained for the OMEx-gasoline DMDF cases, it can be appreciated how most of the cases could be reduced to enter EURO VI normative. Almost all of the results with the

OMEx-gasoline DMDF calibration for EURO VI are in the same range as the diesel-gasoline DMDF calibration and it is not farfetched to say that the same ATS evaluated for the DMDF with conventional fuels could reach very similar performances. This extrapolation is not applicable for the OMEx-gasoline calibration for EURO VII given that HC level at high load conditions and CO levels at low load conditions are significantly greater than those of the diesel-gasoline DMDF calibration. A new DOC design with an appropriate sizing for this application would result in an enhanced performance capable of reaching EURO VI normative. Nonetheless, the authors consider that it is an important task for the future to evaluate the performance of conventional DOCs when using OMEx, as well as providing a more detailed analysis of the sizing of the required DOC.

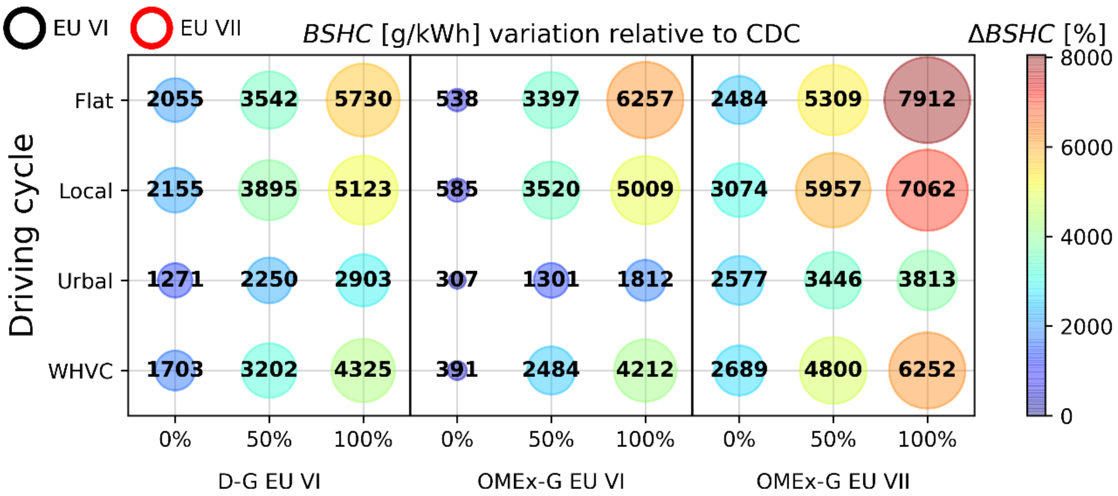


Figure 21. Variation in BSHC emissions under different driving conditions.

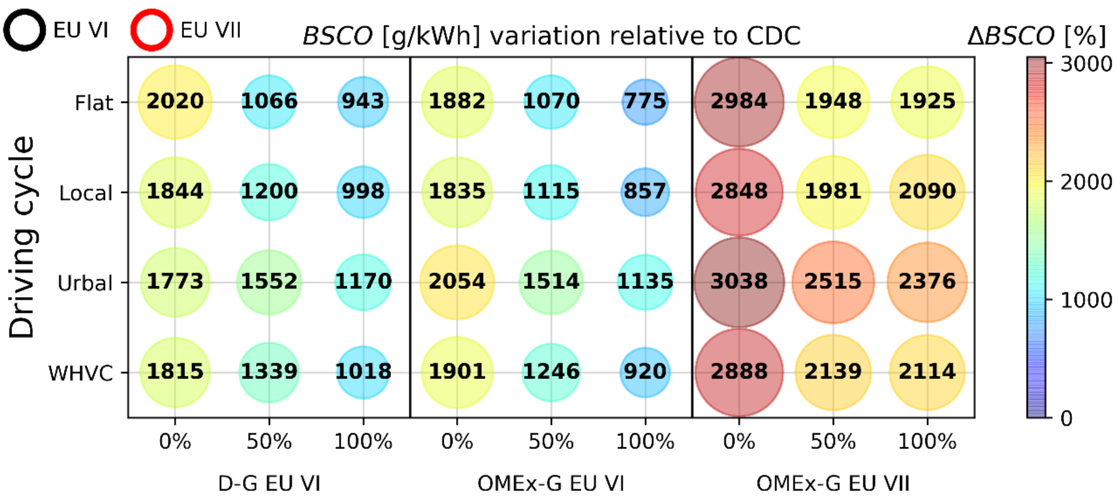


Figure 22. Variation in BSCO emissions under different driving conditions.

To analyse the global fuel consumption, the variations in equivalent BSFC are represented in Figure 23 as an indicative of how efficiently the OMEx is being utilized in the engine to obtain power with respect to the reference fuels. In this sense, the diesel-gasoline shows a small increment with respect to the conventional diesel case, but the great advantage in NOx and soot emissions at low loads gives the diesel-gasoline DMDF engine a great potential for de-rated engines and low-to-medium duty applications. With respect to the EURO VI DMDF OMEx-gasoline calibration, a consistent improvement in efficiency is found under low and medium cargo loads, while very little impact is appreciated at high cargo loads for any driving cycle. This more global efficient utilization of the fuel combined with the advantages in terms of ATS simplification makes the OMEx a very attractive substitute of diesel as HRF in DMDF applications. When utilizing OMEx for the proposed EURO VII calibration, the benefits at high load start to disappear mainly due to the fuel consumption increase at full load conditions, consequence of the hardware limitations Contrary to this, at those conditions with no cargo a certain improvement from the EURO VI calibration is observed. This unintuitive improvement appears because of the small region where only OMEx is used at very low engine loads, allowing for a better performance even when NOx emissions are being reduced by half. This result should not be considered as a global improvement as it is dependent of the vehicle and the selected gear shift strategy, and a specific analysis is necessary for each application. Despite this, the general impact in performance of moving towards a post-EURO VI calibration with a full engine map fulfilling engine-out NOx and soot emissions still results in better performance than when using conventional fuels for the DMDF combustion strategy.

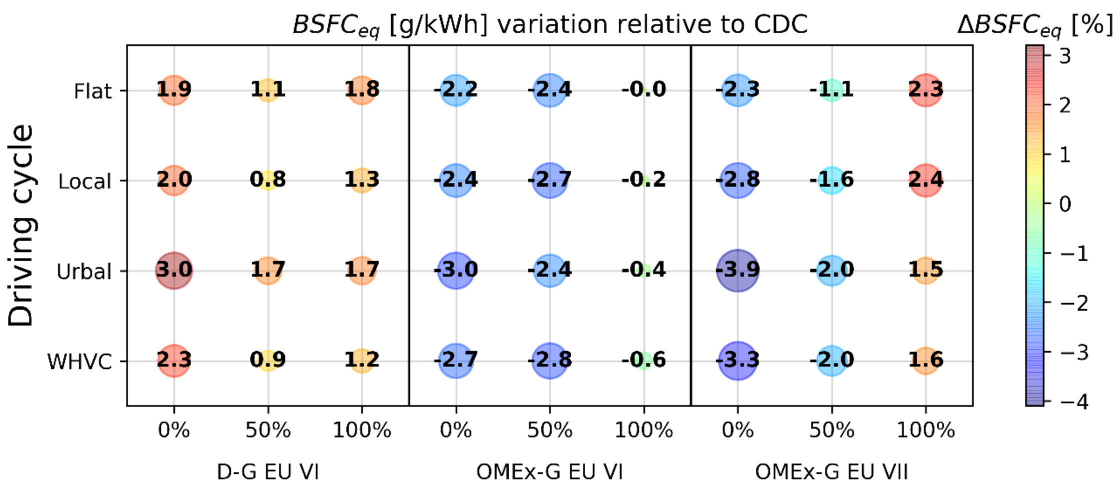


Figure 23. Variation in equivalent BSFC under different driving conditions.

4.2. Analysis of CO_2 emissions

The European Union aims to reduce the CO_2 emissions by targeting the average CO_2 production per fleet [12] as a measure to reduce greenhouse gases emissions. Currently, the regulation on CO_2 emissions only considers tailpipe emissions (also known as Tank-to-Wheel or TtW CO_2 emissions) measured in terms of grams of CO2 emitted per vehicle ton and kilometre travelled ($gCO_2/t\cdot km$), and it will keep like this for the incoming target of Horizon 2025 that requests a 15% reduction in global fleet emissions of CO_2.

Since the engine is not equipped with ATS, the CO_2 emissions data collected during the engine experiments is referred to engine-out conditions. Thus, a direct comparison between the different calibrations at tailpipe level, as the normative requests, cannot be carried out. To do this, the tailpipe CO_2 emissions are estimated from the fuel consumption obtained from the simulation, which has an error below 4% as compared to the experimental measurements. Then, considering the carbon content of each fuel, and assuming its complete oxidation, an accurate estimation of the tailpipe CO_2 emissions can be obtained. This hypothesis is consistent with previous works of the research group, which showed that a conventional ATS can deal with the CO and HC emissions from the DMDF combustion mode [37].Thus, considering the average fuel composition, the CO_2 production for each fuel are considered to be 3.17 g of CO_2/g of fuel for diesel, 3.09 g of CO_2/g of fuel for gasoline and 1.6 g of CO_2/g of fuel for OMEx, and the variations on the global consumption of HRF and LRF due to changes in the fuel fraction along the engine map, the tailpipe CO_2 emissions are estimated.

In Figure 24, the global impact on CO_2 emissions is represented as a percentual variation with respect to the conventional diesel calibration that is considered to be representative of the current fleet. For the diesel-gasoline DMDF calibration slight benefits can be appreciated, but in general the impact is almost negligible. Contrary to this, the low LHV of OMEx makes necessary to increase the fuel consumption in terms of mass, and the total CO_2 emissions are increased significantly despite the lower carbon content of OMEx. This produces a clear increasing trend of the CO_2 production, especially in the cases with no cargo or full cargo that are cases for which the engine needs to operate mostly either at very low engine loads or at very high engine loads (where the PER is lower and the fraction of OMEx is greater).

These results are not working on the direction for the Horizon 2025, so that additional measures have to be considered to enable the application of a DMDF OMEx-Gasoline on real vehicles. In this sense, previous works have demonstrated that to the hybridization is a good method to reduce the fuel consumption (and therefore CO_2 emissions) and can become a good solution to have a platform with high power rates and high autonomy with zero-soot emissions and very low NOx emissions with a simplified ATS for the ICE [31].

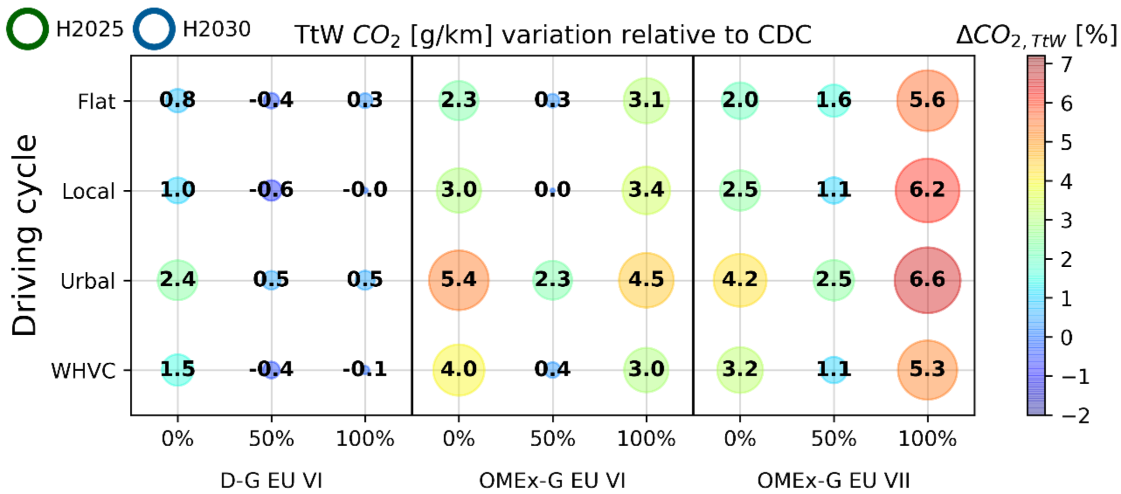


Figure 24. Variation in TtW emissions of CO_2 under different driving conditions.

It is important to note that the CO_2 emissions analysis in terms of TtW is not completely representative of the real CO_2 emissions of a technology using a synthetic fuel like OMEx. Synthetic fuels are produced by combining CO_2 and H_2. For a clean implementation of this synthesis, several technologies DAC, or carbon capture and sequestration (CCS) allow to obtain the CO_2 required to produce the fuel from the ambient or from direct emissions in industrial activities. Hydrogen can be obtained from electrolysis or steam methane reforming utilizing methane as a vector of hydrogen, and all the energy necessary for this process can be obtained from renewable energy sources like solar panels or wind turbines [38]. After this, several techniques can be used to formulate OMEx or other synthetic fuels like Methanol catalytic conversion and Fischer-Tropsch process [39].If this production process is well organized, the final CO_2 emissions during the fuel production can even be negative and compensate the CO_2 emissions during the fuel combustion. The fuel life cycle analysis can result in CO_2-neutral emissions and become an important contributor to the reduction of CO_2 emissions to the atmosphere. By considering the production process, the Well-to-Tank (WtT) CO_2 emissions of the fuel can drastically change the benefits of an e-fuel in terms of contributing to the reduction of greenhouse emissions. Some discussions are being carried out in the actuality to change how the normative for the Horizon 2030 considers the CO_2 emissions by including also the production process.

Data provided by the fuel supplier of this work suggests that, depending on the production process, the use of OMEx could result in a reduction 80% to 90% in terms of Well-to-Wheel (WtW) CO_2 emissions with respect to conventional fuels like diesel or gasoline. If this information is introduced in the previous analysis assuming the best-case scenario to compare the simulations in terms of WtW emissions, Figure 25 shows how easily the 30% CO_2 emissions reduction target (green circle) is achieved with OMEx as HRF in a DMDF engine.

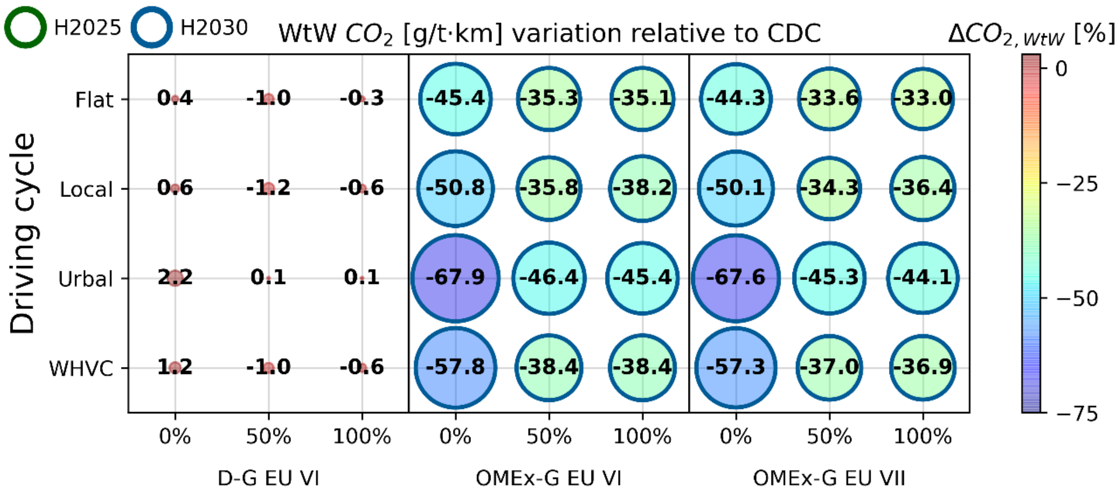


Figure 25. Variation in WtW emissions of CO2 under different driving conditions.

5. Conclusions and future work

This work demonstrates the potential of the dual-mode dual-fuel combustion mode with the novelty of using gasoline as LRF and OMEx as a HRF e-fuel to reach the potential NOx and soot post-EURO VI emissions levels at engine-out conditions. To do so, a complete engine calibration, starting from a previous calibration for EURO VI, has been developed in a multi-cylinder engine. The following main conclusions were reached from the analysis of the results found in this work:

- The DMDF combustion mode with gasoline as LRF and OMEx as HRF is able to reduce to half of the current EURO VI NOx emissions limit while still maintaining a zero-soot operation in all of the engine map.
- The impact on the calibration settings necessary to reach the targets of NOx and soot is very low.
- A consistent penalty in terms of equivalent BSFC was observed due to delaying the combustion process and lowering the indicated efficiency.
- An excessive increase of CO and HC emissions arising from reducing combustion performance make necessary to experimentally evaluate the performance of conventional systems for the oxidation of these substances.

After producing a complete calibration map, a numerical evaluation of the results was performed using GT-Power under realistic vehicle specifications and different driving conditions including the effect of the cargo load. For this evaluation, the original EURO VI calibrations of diesel-gasoline DMDF and OMEx-gasoline DMDF, as well as the obtained post-EURO VI OMEx-gasoline DMDF calibration have been compared against conventional diesel engine operation. From this numerical evaluation, it was possible to conclude that:

- Diesel-gasoline DMDF has a small penalty in BSFC and is only able to fulfil EURO VI NOx and soot limitations at low cargo load.
- The increase in thermal efficiency when using OMEx results in a consistent reduction of equivalent BSFC compared to CDC. The higher the cargo load, the smaller the benefit in equivalent BSFC.
- The global impact of the equivalent BSFC penalty when reaching post-EURO VI levels of NOx and soot is especially important at full cargo load. This has been associated to a hardware limitation at full load conditions.
- Both calibrations utilizing OMEx as HRF can maintain a zero-soot combustion while fulfilling their respective limitation in NOx emissions in all evaluated scenarios.

An additional analysis of the impact of using OMEx in terms of CO_2 emissions was carried out assuming that an adequate ATS system would be implemented to oxidize the high levels emissions of CO and HC. This evaluation was performed for both, TtW and WtW emissions of CO_2, and the conclusions were:

- The effect of having lower LHV for the HRF increases the absolute fuel mass consumed and is not compensated by the fact of having lower carbon content in the fuel. Therefore, the TtW CO_2 emissions are significantly increased.
- To reach the Horizon 2025 target, measures related to direct reduction of fuel consumption must be implemented, like the vehicle hybridization.
- The use of OMEx as a synthetic fuel can become a promising solution to reach the H2030 CO_2 reduction target on Well-to-Wheel basis.

During this work some limitations on the hardware of the engine and some hypothesis that may not be applicable any longer have been found. As a way of improving and further investigating the capabilities of the DMDF utilizing OMEx as HRF, the following tasks are proposed to be evaluated in the future.

- Dedicated analysis on the limitations imposed by the DI fuel supply system. A common rail system with higher pressure capacity that can reduce the injection duration significantly should be tested.
- Dedicated analysis on the limitation of the turbocharger at low engine speeds. A different combination of turbine and compressor dedicated for this combustion mode with high EGR rates could improve the engine performance in this region of the operating map.
- Test conventional DOCs to corroborate if current levels of CO and HC coming from both calibrations with OMEx can be reduced below EURO VI limits with low impact on the ATS.

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7. References

[1] Frey, H. C. (2018). Trends in on-road transportation energy and emissions. Journal of the Air and Waste Management Association, 68(6), 514–563. https://doi.org/10.1080/10962247.2018.1454357

[2] Barbosa, F. C. (2020). Heavy Duty Diesel Emission Standards Regulation Evolution Review-Current Outcomes and Future Perspectives. SAE Technical Papers, January. https://doi.org/10.4271/2019-36-0174

[3] Fan, Y. Van, Perry, S., Klemeš, J. J., & Lee, C. T. (2018). A review on air emissions assessment: Transportation. Journal of Cleaner Production, 194, 673–684. https://doi.org/10.1016/j.jclepro.2018.05.151

[4] Winkler, S. L., Anderson, J. E., Garza, L., Ruona, W. C., Vogt, R., & Wallington, T. J. (2018). Vehicle criteria pollutant (PM, NOx, CO, HCs) emissions: how low should we go? Climate and Atmospheric Science, 1(1). https://doi.org/10.1038/s41612-018-0037-5

[5] Bonnel, P., Kubelt, J., & Provenza, A. (2011). Heavy-Duty Engines Conformity Testing Based on PEMS. https://doi.org/10.2788/56535

[6] EU: Heavy-Duty Truck and Bus Engines. (2019). Accessed on September, 2020. https://dieselnet.com/standards/eu/hd.php.

[7] World Harmonized Stationary Cycle (WHSC). Emission Test Cycles: World Harmonized Stationary Cycle (WHSC). (2008). Accessed on September, 2020. https://dieselnet.com/standards/cycles/whsc.php.

[8] World Harmonized Transient Cycle (WHTC). Emission Test Cycles: World Harmonized Transient Cycle (WHTC). (2007). Accessed on September, 2020. https://dieselnet.com/standards/cycles/whtc.php.

[9] Noble, A (2017). Next Generation. EURO 7 Update. Transport Engineer.
[10] Transport & Envrionment. (2020). Road to Zero: the last EU emission standard for cars, vans, buses and trucks. 1–19.
[11] "ACEA Position Paper-Views on Proposals for Euro 7 Emission ..." Accessed January 11, 2021. https://www.acea.be/uploads/publications/ACEA_Position_Paper-Views_on_proposals_for_Euro_7_emission_standard.pdf.
[12] EU. (2019). Regulation (EU) 2019/1242 of the European Parliament and of the Council of 20 June 2019 Setting CO2 emission performance standards for new heavy-duty vehicles and amending Regulations (EC) No 595/2009 and (EU) 2018/956 of the European Parliament. Official Journal of the European Union, L 198(April), 202–240. https://eur-lex.europa.eu/eli/reg/2019/1242/oj
[13] Venu, H., Subramani, L., & Raju, V. D. (2019). Emission reduction in a DI diesel engine using exhaust gas recirculation (EGR) of palm biodiesel blended with TiO2 nano additives. Renewable Energy, 140, 245–263. https://doi.org/10.1016/j.renene.2019.03.078
[14] Reşitoğlu, I. A., Altinişik, K., & Keskin, A. (2015). The pollutant emissions from diesel-engine vehicles and exhaust aftertreatment systems. Clean Technologies and Environmental Policy, 17(1), 15–27. https://doi.org/10.1007/s10098-014-0793-9
[15] Helbing, C., Köhne, M., Kassel, T., Wietholt, B., Krause, A., Lohre, L., Gerhardt, N., & Eiglmeier, C. (2020). Volkswagen ' s TDI-Engines for EURO 6d – Clean Efficiency for Modern Mobility. 41st Internationales Wiener Motorensymposium.
[16] Daimler. (n.d.). Changeover to new emissions standard: New Mercedes-Benz passenger cars comply with the EURO 6d-TEMP standard as a minimum. Retrieved November 09, 2020, from https://www.daimler.com/innovation/diesel/nox-reduction.html
[17] Demuynck, J., Favre, C., Bosteels, D., Bunar, F., Spitta, J., & Kuhrt, A. (2019). Diesel Vehicle with Ultra-Low NOx Emissions on the Road. SAE Technical Papers, 2019-September(September). https://doi.org/10.4271/2019-24-0145
[18] Posada, F., Isenstadt, A., & Badshah, H. (2020). Estimated Cost of Diesel Emissions-Control Technology To Meet Future California Low NOx Standards in 2024 and 2027. May. https://theicct.org/sites/default/files/publications/HDV-emissions-compliance-cost-may2020.pdf
[19] Jeevahan, J., Mageshwaran, G., Joseph, G. B., Raj, R. B. D., & Kannan, R. T. (2017). Various strategies for reducing Nox emissions of biodiesel fuel used in conventional diesel engines: A review. Chemical Engineering Communications, 204(10), 1202–1223. https://doi.org/10.1080/00986445.2017.1353500
[20] Benajes, J., García, A., Monsalve-Serrano, J., & Villalta, D. (2018). Exploring the limits of the reactivity-controlled compression ignition combustion concept in a light-duty diesel engine and the influence of the direct-injected fuel properties. Energy Conversion and Management, 157(July 2017), 277–287. https://doi.org/10.1016/j.enconman.2017.12.028
[21] García, A., Monsalve-Serrano, J., José Sanchís, E., & Fogué-Robles, Á. (2020). Exploration of suitable injector configuration for dual-mode dual-fuel engine with diesel and OMEx as high reactivity fuels. Fuel, 280(June), 118670. https://doi.org/10.1016/j.fuel.2020.118670

[22] García, A., Monsalve-Serrano, J., Lago Sari, R., & Gaillard, P. (2020). Assessment of a complete truck operating under dual-mode dual-fuel combustion in real life applications: Performance and emissions analysis. Applied Energy, 279(August), 115729. https://doi.org/10.1016/j.apenergy.2020.115729
[23] García, A., Monsalve-Serrano, J., Villalta, D., Lago Sari, R., Gordillo Zavaleta, V., & Gaillard, P. (2019). Potential of e-Fischer Tropsch diesel and oxymethyl-ether (OMEx) as fuels for the dual-mode dual-fuel concept. Applied Energy, 253(July), 113622. https://doi.org/10.1016/j.apenergy.2019.113622
[24] Benajes, J., García, A., Monsalve-Serrano, J., & Sari, R. (2020). Clean and efficient dual-fuel combustion using OMEx as high reactivity fuel: Comparison to diesel-gasoline calibration. Energy Conversion and Management, 216(May), 112953. https://doi.org/10.1016/j.enconman.2020.112953
[25] Systems, E. M. (2019). Emission Measurement Systems - HORIBA. 1–3. https://www.horiba.com/en_en/products/by-segment/automotive-test-systems/emissions-measurement-systems/ . Accessed in September, 2020.
[26] AVL. (2005). Smoke Value Measurements with the Filter-Paper-Method (Issue June).
[27] Payri, F., Olmeda, P., Martin, J., & Carreño, R. (2014). A New Tool to Perform Global Energy Balances in DI Diesel Engines. SAE International Journal of Engines, 7(1), 2014-01–0665. https://doi.org/10.4271/2014-01-0665
[28] Gamma Technologies. (2019). GT-POWER Engine Simulation Software | Gamma Technologies. https://www.gtisoft.com/gt-suite-applications/propulsion-systems/gt-power-engine-simulation-software/ . Accessed in March, 2020.
[29] García, A., Monsalve-Serrano, J., Rückert Roso, V., & Santos Martins, M. E. (2017). Evaluating the emissions and performance of two dual-mode RCCI combustion strategies under the World Harmonized Vehicle Cycle (WHVC). Energy Conversion and Management, 149, 263–274. https://doi.org/10.1016/j.enconman.2017.07.034
[30] European Parliament And The Council Of The European Union. (2018). Regulation (EC) No 595/2009 of The European Parliament and of the Council of 18 June 2009 on type-approval of motor vehicles and engines with respect to emissions from heavy duty vehiclec (EURO VI) and amending Regulation (EC) No 715/2007 and Directive 20. 52(68), 48–119. http://data.europa.eu/eli/reg/2009/595/oj
[31] García, A., Monsalve-Serrano, J., Martinez-Boggio, S., Gaillard, P., Poussin, O., & Amer, A. A. (2020). Dual fuel combustion and hybrid electric powertrains as potential solution to achieve 2025 emissions targets in medium duty trucks sector. Energy Conversion and Management, 224(August), 113320. https://doi.org/10.1016/j.enconman.2020.113320
[32] Krishnamoorthi, M., Malayalamurthi, R., He, Z., & Kandasamy, S. (2019). A review on low temperature combustion engines: Performance, combustion and emission characteristics. Renewable and Sustainable Energy Reviews, 116(February), 109404. https://doi.org/10.1016/j.rser.2019.109404
[33] Desantes, J. M., Arrègle, J., López, J. J., & Cronhjort, A. (2006). Scaling laws for free turbulent gas jets and diesel-like sprays. Atomization and Sprays, 16(4), 443–473. https://doi.org/10.1615/AtomizSpr.v16.i4.60

[34] Desantes, J. M., Lopez, J. J., Garcia, J. M., & Pastor, J. M. (2007). Evaporative diesel spray modeling. Atomization and Sprays, 17(3), 193–231. https://doi.org/10.1615/atomizspr.v17.i3.10
[35] Desantes, J. M., García-Oliver, J. M., Pastor, J. M., & Pandal, A. (2016). A comparison of diesel sprays CFD modeling approaches: DDM versus σ-γ Eulerian atomization model. Atomization and Sprays, 26(7), 713–737. https://doi.org/10.1615/AtomizSpr.2015013285
[36] García, A., Gil, A., Monsalve-Serrano, J., & Lago Sari, R. (2020). OMEx-diesel blends as high reactivity fuel for ultra-low NOx and soot emissions in the dual-mode dual-fuel combustion strategy. Fuel, 275(February), 117898. https://doi.org/10.1016/j.fuel.2020.117898
[37] Benajes, J., Garcia, A., Monsalve-Serrano, J., & Sari, R. (2018). Evaluating the Efficiency of a Conventional Diesel Oxidation Catalyst for Dual-Fuel RCCI Diesel-Gasoline Combustion. SAE Technical Papers, 2018-Septe, 1–13. https://doi.org/10.4271/2018-01-1729
[38] Desantes, J. M., Molina, S., Novella, R., & Lopez-Juarez, M. (2020). Comparative global warming impact and NOX emissions of conventional and hydrogen automotive propulsion systems. Energy Conversion and Management, 221, 113137. https://doi.org/10.1016/j.enconman.2020.113137
[39] Gierlich, C. H., Beydoun, K., Klankermayer, J., & Palkovits, R. (2020). Challenges and Opportunities in the Production of Oxymethylene Dimethylether. Chemie Ingenieur Technik, 92(1–2), 116–124. https://doi.org/10.1002/cite.201900187

Abbreviations

ATS: Aftertreatment System

BMEP: Brake Mean Effective Pressure

BTE: Brake Thermal Efficiency

BSCO Brake Specific CO Emissions

BSFC: Brake Specific Fuel Consumption

BSFCeq: Equivalent Brake Specific Fuel Consumption

BSHC Brake Specific HC Emissions

BSNOx: Brake Specific NOx Emissions

BSSoot Brake Specific Soot Emissions

CAD: Crank Angle Degree

CA10: Crank Angle for 10% of burned fuel mass

CA50: Crank Angle for 50% of burned fuel mass

CA90: Crank Angle for 90% of burned fuel mass

CCS: Carbon Capture and Sequestration

CDC: Conventional Diesel Combustion

CN: Cetane Number

CO: Carbon monoxide

CO2: Carbon dioxide

DAC: Direct Air Capture

D-G: Diesel-Gasoline fuel blend

DI: Direct Injection

DMDF: Dual-Mode Dual-Fuel

DOC: Diesel Oxidizer Catalyst

DPF: Diesel Particulate Filter

EGR: Exhaust Gas Recirculated

GIE: Gross Indicated Efficiency

H2025: Horizon 2025

H2030: Horizon 2030

H2O: Water

HC: Unburned hydrocarbons

HCCI: Homogeneous Charge Compression Ignition

HRF: High Reactivity Fuel

ICE: Internal Combustion Engine

LHV: Lower Heating Value

LRF: Low Reactivity Fuel

LTC: Low-Temperature Combustion

MON: Motor Octane Number

NOx: Nitrogen Oxides

OME/POMDE: Polyoxymethylene Dimethyl Ether

OMEx: OME3-5 Mix

OMEx-G: OMEx-Gasoline fuel blend

PER: Premixed Energy Ratio

PFI: Port Fuel Injection

PM: Particulate Matter

PPCI: Partially Premixed Compression Ignition

PRR: Pressure Rise Rate

RoHR: Rate of Heat Release

RON: Research Octane Number

RCCI: Reactivity Controlled Compression Ignition

SCR: Selective Catalytic Reducer

SOC: Start of Combustion

SOI: Start of Injection

TDC: Top Dead Centre

TSCI: Thermally Stratified Compression Ignition

TtW: Tank-to-Wheel

VGT: Variable Geometry Turbine

WHSC: World Harmonized Stationary Cycle

WHTC: World Harmonized Transient Cycle

WHVC: World Harmonized Vehicle Cycle

WtW: Well-to-Wheel

WtT: Well-to-Tank