Methanol and OMEx as fuel candidates to fulfill the potential EURO VII emissions regulation under dual-mode dual-fuel combustion

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Abstract

Recent investigations show that there are different combustion strategies that promise to achieve higher efficiencies in internal combustion engines. Advanced combustion modes such as reactivity controlled compression ignition (RCCI) and dual-mode dual-fuel (DMDF) have proven to be able to achieve low NOx and soot engine-out emissions while being able to operate over the complete engine map. On another front, intensive research has been done in the fuels field. Oxygenated fuels, like oxymethylene dimethyl ethers (OMEx) and methanol, are of special interest because of their potential to reduce the soot emissions, while allowing to adjust parameters, such as EGR, to higher values to also reduce NOx emissions and avoid other problems like excessive in-cylinder peaks of pressure. In this research, the effects of diesel-methanol and OMEx-gasoline fuels were studied on a dual-mode dual-fuel (DFDM) multi-cylinder engine at 1800 rpm for various loads (25, 50, 80 and 100%). To do so, a dedicated calibration to optimize the brake thermal efficiency while trying to maintain NOx and soot emissions under EURO VI limitations was applied. Then, the combustion characteristics, performance and emissions results are compared to a base diesel-gasoline case. Boundary conditions (intake pressure, temperature and air mass) for each fuel combination where similar, with the exception of the premixed energy ratio, which was on average 20% lower for diesel-methanol. Each fuel combination was, weighted against each other by means of an merit function, where the fuel combination with the lowest value has the closest approximation to ideal $BSFC_{eq}$, BSNOx, BSSoot, BSCO and BSHC targets. Results show that diesel-methanol presents no significant differences with respect to diesel-gasoline in terms of equivalent $BSFC_{eq}$ and NOx, while the substitution of diesel by OMEx in the dual-mode dual-fuel combustion has the potential of, although penalizing HC and CO emissions, not only achieving the EURO VI NOx (0.4 g/kWh) limits but also the future potential EURO VII (0.2 g/kWh) as well because of a negligible soot production that allows using EGR levels of up to 50%.

Keywords

Dual fuel combustion; OMEx; Methanol; e-Fuels; EUVII emissions

1. Introduction

Emissions regulations must evolve through time and become stricter to try to mitigate the steep increase of greenhouse gases emissions and pollution because of the negative effects that they have on human health [1] and the environmental wear that they cause [2]. Regulations are a way to, firmly, incentivize manufacturers to develop new strategies in order to avoid environmental pollution and emissions . The transport sector has an important contribution to these emissions; in Europe, this sector makes up almost a quarter of the Greenhouse gases and is the main cause of pollution in the cities [3]. Although electric vehicles are generally considered cleaner on per use emissions and, as such a solution to the reduction of emissions and compliance with regulations, they are not exempt from polluting the air [4]. Therefore, is in the best interest of the automotive industry to continue researching and improving the internal combustion engine (ICE) technology to fulfill the new restrictions. Especially, since ICE vehicles are expected to be the main source of transportation in the foreseeable future [5].

Recently, several investigations on the engine hardware [6] [7], combustion concepts [8] [9], and electric powertrain coupling [10] [11] have made considerable progress in achieving the objectives of reducing emissions without penalizing the performance. However, these advancements by themselves will not be enough in the long term. To aid in this task the development and investigation of new fuels, and strategies to use such fuels in a way that enables the combustion engines to take advantage of most of the energy available, is of the upmost importance. In this regard, new fuels have appeared with properties like better more oxygenation [12] [13], carbon neutrality [14], more energy content or electrical recharge ability [15]. Highly oxygenated fuels like methanol and OMEx are interesting case studies to appreciate the enhancements and downsides that their properties contribute when used coupled with combustion concepts like reactivity controlled compression ignition (RCCI) and dual-fuel dual-mode (DFDM) [8] [16]; thus, they will be the focus of this paper.

Previous research has indicated that using oxygenated fuels, like methanol and OMEx, can effectively reduce the soot formation in compression ignition (CI) engines [17] [18]. Methanol, in particular, has been considered as a replacement for conventional diesel because it burns at lower temperatures, producing, in principle, less NOx [19]. It is a single carbon renewable fuel that can be produced in different ways: from fossil fuels, biomass and carbon capture and utilization schemes (the latter two, both renewable energy sources) [20]. It has a higher oxygen content, enthalpy of vaporization and octane number than gasoline, characteristics that can enhance reactivity gradients and extend the operational load range [21]. Its simpler molecular structure reduces the formation of soot [22], as the fuel reacts faster with oxygen, of which it has a higher content. In this study, methanol is used as the low reactivity fuel (LRF) for a DFDM concept in tandem with diesel as the high reactivity fuel (HRF). Several studies using methanol as an additive for diesel or as a blend [23] [24], found that smoke, CO and unburned HC emissions are decreased, but NOx increased as the methanol proportion in the blend increased. Works on low temperature combustion (LTC) modes with methanol, although scarcer, can also be found. In [25], Li et al. studied the optimization of diesel-methanol at low loads with RCCI and direct dual fuel stratification (DDFS), [26] compared diesel-methanol and PODE-methanol fuels in RCCI combustion

and [27] also researched methanol for RCCI by both direct injection (DI) and port injection, finding that it has the potential of achieving ultra-low NOx through port injection.

On the other hand, Oxymethylene dimethyl ethers (OMEx) are a family of synthetic fuels formed by the chemical structure $CH_3O(CH_2O)_xCH_3$ where x can take values from 1 to 5 [28], modifying the size of the molecule. The larger molecules (x=3-5) can be used for compression-ignition engines due to their very high cetane numbers [14]. These e-fuels lack C-C bonds and possess many oxygen atoms, which promote negligible soot emissions [29]. Methanol and formaldehyde are intermediate steps in its production [30] and the synthetization process demands lower electrical energy for its production than other e-fuels [29]. Previously, OMEx used as a HRF with gasoline as a LRF has been compared to diesel-gasoline in dual-fuel combustion [31] [32] [33], resulting in ultra-low soot values throughout all the engine map. The low soot values allowed to calibrate the engine to also have high EGR levels, that reduce NOx emissions, and higher efficiency due to lower pumping losses.

This work, aims to compare the combustion, performance and emissions of diesel-methanol and OMEx-gasoline under DFDM, on a multi-cylinder engine working at 1800 rpm and 25, 50, 80 and 100% load to evaluate if they have the potential to reduce emissions (NOx and soot) while maintaining performance to fulfill EURO VI emission limits and beyond (EU VII) without significantly increasing fuel consumption nor the requirements for aftertreatment systems.

2. Materials and methods

The following section outlines the steps followed and the equipment employed during the investigation. In particular, the engine, test facilities and the trial procedures will be presented.

2.1. Engine characteristics

Tests were performed in a multi-cylinder, four stroke, 8L, compression ignition engine modified to be able to operate under a dual-fuel combustion strategy. Among its modifications from the commercially available unit, six port fuel injectors (PFI) were installed to inject the low reactivity fuel (LRF) in the intake manifold. Moreover, to avoid excessive peak pressure gradients at high loads, the compression ratio (CR) was reduced from 17.5:1 to 12.8:1. To do so, the piston bowl was optimized to be more compatible with the RCCI mode. Finally, a low pressure (LP) exhaust gas recirculation (EGR) system was added to deliver EGR and maintain the mass flow in the turbine, whilst lowering the temperature of the EGR gasses. Table 1 describes the engine main characteristics.

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

2.2. Description of the test cell

The main measurement devices are represented in the test cell facility scheme (Figure 1). Fully premixed combustion requires a high level of charge dilution, for this, a LP EGR system was incorporated to the stock engine. Additional water filters were included before the compressor to remove the water condensates formed in the EGR line caused by the temperature drop [35]. The average values for temperature and pressure were recorded at the locations represented in the scheme shown in Figure 1. A Kistler 6125C cylinder pressure transducer was equipped to each individual cylinder to allow the detection of possible cylinder-to-cylinder dispersions on the different gaseous quantities: air, EGR and LRF.

The data was collected using an AVL 364 encoder with a of 0.2 crank angle degree (CAD) resolution. A Horiba MEXA-7100 DEGR analyzer measured the O_2, CO, CO_2, NOx and HC at the exhaust. To quantify the smoke emissions by extracting a subset of the exhaust gas and passing it through a filter paper, an AVL 415S smoke meter was used. With this device, the blackening of the filter paper indicates the soot content in its own unit filter smoke number (FSN) [34]. Instantaneous fuel consumption for both the HRF and LRF was measured by two AVL 733 S balances. Air mass flow was measured with an Elster RVG G100 sensor. An AVL PUMA interface was utilized for the acquisition of low frequency data, at a rate of 10 Hz, and controlling the engine speed. Engine torque was user regulated. A NI PXIe 1071 board controlled all injection systems (high reactivity and low reactivity) and recorded high frequency signals; it was able to process pressure data and perform a heat release analysis in real time. The board allows the monitoring of the main combustion metrics and heat release rate (HRR) profile while connected to a controller system developed in the research facility. It also allows the control of the back-pressure valve and the LP EGR system. Table 2 displays the main elements of the test cell with their respective accuracy.

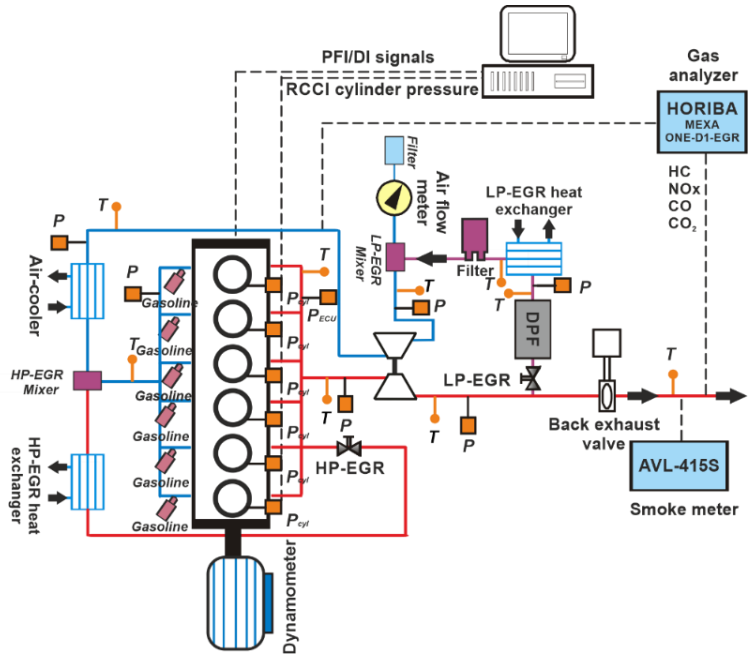

Figure 1. Test cell scheme.

Table 2. Instrumentation information.

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 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%

2.3. Fuels and injection systems characteristics

Dual-fuel configurations require injection systems adapted to the premise of having to inject both a high reactivity fuel (HRF) and a low reactivity one. This research focuses on OMEx-Gasoline (OMEx-G) and diesel-methanol as the main fuel combinations to perform analysis on. The characteristics of the fuels discussed are described in Table 3.

Table 3. Fuel physical and chemical properties.

	EN 228 gasoline	EN 590 diesel	Methanol	OMEx
Density [kg/m^3] (T= 15 °C)	720	842	792	1067
Viscosity [mm^2/s] (T= 40 °C)	0.545	2.929	0.58	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	-	109	-
MON [-]	85.7	-	100	-
Lower heating value [MJ/kg]	42.4	42.44	19.93	19.04

As previously mentioned, a PFI system was adapted to the engine to be able to inject the LRF (gasoline or methanol); said system operates at 5.5 bar. The stock injection system was used to inject the HRF (OMEx, in the case of OMEx-G tests, and diesel, for diesel gasoline tests)through direct injection (DI). The same Labview software controlled the injection timing and fuel mass for both the HRF and LRF. Table 4 describes the most important information of the fuel injection systems used in the research facility.

Table 4. Characteristics of the direct and port fuel injectors.

Direct injector		Port fuel 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. Calibration methodology description

Implementation of a dedicated calibration methodology was done to stay within predefined constraints, i.e. the upper exhaust pressure limit imposed by the size of the turbocharger. A multiple-injection strategy was used to allow more degrees of freedom controlling the combustion and, therefore, achieving the desired targets. The first calibration step sought to achieve the desired load operating under dual-fuel conditions while maintaining the pressure gradient under 15 bar/CAD and the maximum pressure below 185 bar. For that purpose, the engine was started at conventional diesel combustion (CDC) conditions. Then, the LRF percentage was incremented to reach the maximum premixed energy ratio (PER); this step was iterated over after the last step of the calibration loop. Higher PER values indicate that the LRF governs the combustion process and excessive pressure rise rates (PRR) and high cylinder pressure can be seen. Consequently, to avoid mechanical problems it is necessary to decrease the PER by delaying the start of injection (SOI) of the HRF, attempting to shift the injection towards after top dead center (TDC). Finally, the indicated mean effective pressure coefficients of variations are targeted to values lower than 4% to procure a stable combustion. When

the load was accomplished, and all the constraints are satisfied, the next step was commenced.

The successive phase involves the compliance with the regulations of EURO VI for soot and NOx. To comply with EURO VI regulations for particulate matter (PM) of soot, a limit of 0.01 g/kWh was imposed. The limit is intended to ensure the quantity of PM emissions are below EURO VI; however, it does not ensure that soot (in terms of PM) will be below EURO VI. It does guarantee tests with soot contents above this limit will also be uncompliant wit EURO VI. After the load setting step, four different scenarios can be presented: either NOx and soot are below (0.4 g/kWh and 0.01 g/kWh respectively), NOx is below EUVI limit and soot is above 0.01 g/kWh, NOx is over the EUVI limit and soot is below 0.01 g/kWh, or both pollutants exceed the limits.

Depending on the aforementioned scenarios, different methods are recommended to reach the limit values. If NOx are higher than EUVI, there are several alternatives to follow: 1) increase the EGR concentration to increment the dilution levels, 2) shift the combustion closer to the compression stroke by delaying the SOI in order to reduce the temperatures during the combustion. Otherwise, when soot values are higher than the specified limit, the in-cylinder oxygen concentration should be increased and the mixture improved. This can be done with higher injection pressures aimed at improving the penetration and evaporation sprays; reducing the EGR concentration and increasing the degree of dilution; increasing the inlet pressure; and using earlier SOIs to promote longer mixing times thereby reducing the zone-rich equivalence ratio.

High load conditions have a bigger chance of exceeding both NOx and soot limits, because part of the HRF burns diffusively and the more elevated temperatures favor the NOx formation. Strategies must therefore be used to reduce NOx and soot, such as increasing the EGR concentration, increasing the oxygen concentration at the intake (by closure of the turbine), moving SOI forward to an earlier stage and increasing the injection pressure. These strategies usually have adverse effects on soot and NOx; increasing EGR could reduce the NOx but enhance the soot formation due to the decrease of the air to fuel equivalence ratio, and earlier SOIs and injection pressure could lead to higher in-chamber temperatures and the subsequent increase in NOx formation. Hardware flexibility is explored to achieve emission target, however, when this cannot be done, constraints can be set to less strict parameters to reach the desired engine load.

The final step of the calibration aims to reduce the main losses that occur during the conversion of the chemical energy from the fuel into mechanical energy, while maintaining the emission limits achieved in the previous step. In order to provide the required amount of EGR and reduce pumping losses, the position of the variable geometry turbine (VGT) is modified; then, a sweep of the LP and HP EGR is made, while also varying the VGT position to provide the same amount of air and EGR, but with a better turbine efficiency and lower backpressure in the engine. The other processes of this step aim to achieve optimal combustion phasing, improved combustion efficiency and minimal pumping losses by evaluating the effect of small modifications on the PER, SOI and EGR.

Figure 2 presents a scheme of the calibration methodology steps presented above. To ensure the calibration does not coincide with a local minimum, and that the initial high PER assumption is true, the PER is evaluated again at the end of the loop.

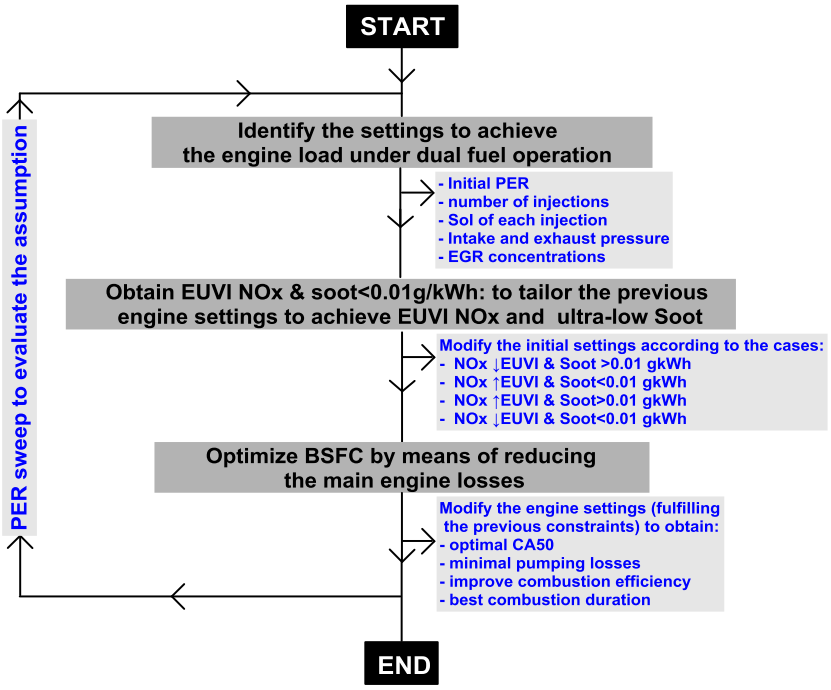

Figure 2. Engine calibration loop for each fuel blend.

Because this work compares different fuels, with significantly different lower heating values (LHV), fuel consumption results are compared on an equivalent basis. The selection of this method seeks to evaluate the conversion efficiency of the fuels and exclude the impact of the LHV. This is done by defining the equivalent brake specific fuel consumption ($BSFC_{eq}$) using the lower heating value of diesel as reference, and accounting for the fuels in terms of mass and their respective LHV.

$$BSFC_{eq}[g/kWh] = \frac{\dot{m}_{HRF} \cdot \left(\frac{LHV_{HRF}}{LHV_{diesel}}\right) + \dot{m}_{LRF} \cdot \left(\frac{LHV_{LRF}}{LHV_{gasoline}}\right)}{P_b} \quad (1)$$

3. Results and discussion

The current section discusses the results for each of the different fuel combinations evaluated. Four engine loads (25%, 50%, 80% and 100%) were tested at 1800 rpm. The combustion, performance and emissions results are analyzed to grasp the outcome of the different tested fuels (diesel-methanol and OMEx-gasoline) compared to previously obtained data for the base case, diesel-gasoline. This is done in order to evaluate the benefits and drawbacks of replacing gasoline by methanol as LRF and OMEx as HRF, due to the effect of the different fuel characteristics.

3.1. Boundary conditions

During the calibration of the operating points, the focus was to achieve the set targets while respecting the emission constraints (NOx and soot) and minimizing the power losses associated to the peripheral systems. Hence, injection pressure, while being one the parameters that could be adjusted, was always tried to be maintained at

the lowest possible value. Nonetheless, when the load increases, the injection pressure had to be increased to be able to achieve the required amount of fuel in reasonable injection duration values. Even thought, the calibration procedure allowed the intake conditions to vary between the fuel combinations. As it can be seen in Figure 2, the differences in terms of air management conditions between fuels are not that high, being the PER, because its relation to the physical properties of the fuels, the outlier.

Diesel-methanol intake pressure values are quite similar to those of diesel-gasoline for all the operating conditions. However, this fuel combination has a higher air mass flow (up to 11.5%) due to the cooling effect occurring during the fuel vaporization because of the higher latent heat of vaporization of methanol. Contrary to this, the air mass flow for the OMEx-gasoline case shows a reduction of about 6.2% percent with respect to the diesel-gasoline base case. At medium loads, the intake pressure for OMEX-gasoline is lower than the other fuel combinations. The higher oxygen content of OMEx can be attributed as partial responsible for this fact. In addition, since less air mass flow would be needed to achieve the same oxygen values during the calibration procedure, the intake pressure can be reduced to minimize the pumping losses, as will be discussed in further sections.

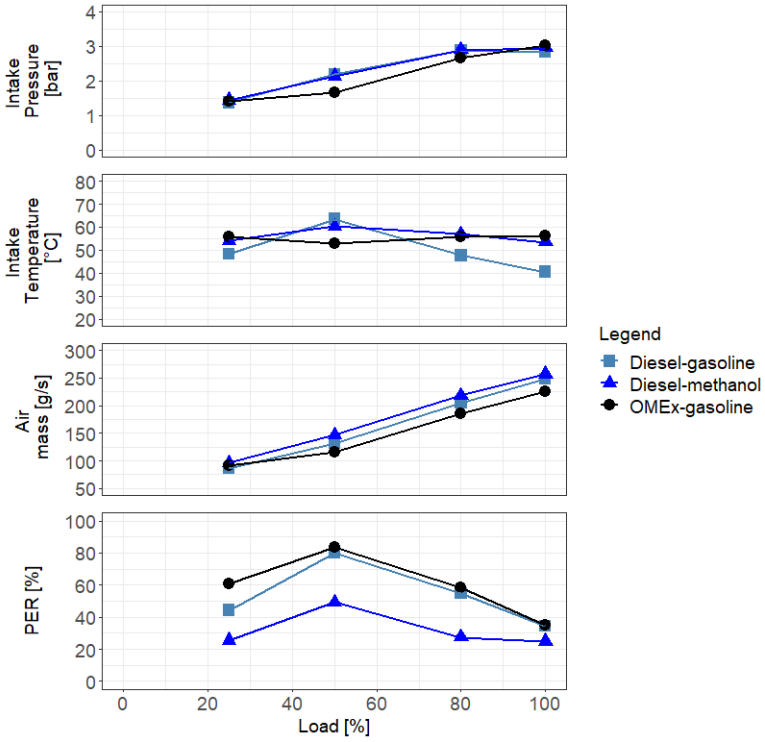

Figure 2. Boundary conditions at the intake manifold.

Table 5 presents the combustion control parameters that, even if their variations are not the primary subject of study of this paper, they can have an important effect on the combustion, performance, and emission characteristics of the tested points. Therefore, it is of importance to at least take them into consideration when discussing further results.

Table 5. Operating point parameters.

Fuels	Diesel-gasoline				Diesel-methanol				OMEx-gasoline			
Load [%]	25	50	80	100	25	50	80	100	25	50	80	100
Speed [rpm]	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800	1800
EGR [%]	38.1	43.5	35.0	20.4	41.1	38.3	32.8	19.7	38.5	39.7	38.4	33.3
Inj. Pres. [bar]	800	600	2000	2000	800	800	2000	2000	600	600	1200	2000
SOI main [CAD bTDC]	22	50	8	7	16	17	8	11	29	26	10	8
TOI main [μs]	692	546	889	1323	763	776	1231	1516	952	799	1497	2359
SOI pilot [CAD bTDC]	32	60	-	-	30	32	-	-	44	46	30	-
TOI pilot [μs]	692	546	-	-	763	776	-	-	952	799	676	-
LRF amount [%]	44.3	79.9	54.8	34.2	42.3	67.3	44.7	41.5	41.0	69.9	38.4	19.5

294

295 **3.2. Combustion results**

296 Figure 3 shows the comparison of the in-cylinder pressure and heat release rate
297 profiles at all loads for the different fuel combinations tested under the RCCI strategy at
298 1800 rpm. It is worth mentioning that at low-to-medium load, a fully premixed
299 combustion occurs, while at high-to-full load a dual-fuel diffusive combustion takes
300 place. The analysis will be done by comparing each fuel combination versus the diesel-
301 gasoline reference. This will help to highlight the differences that arise when substituting
302 diesel by OMEx and gasoline by methanol.

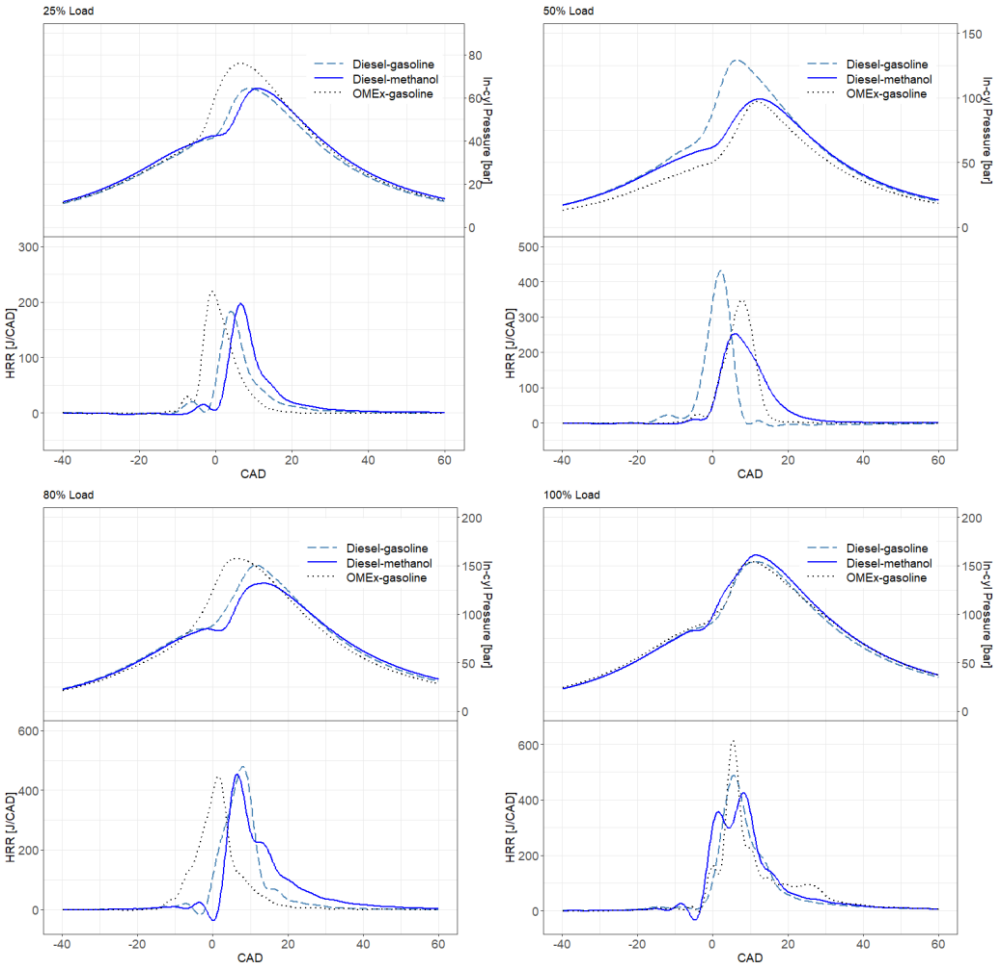

Figure 3. Effect of diesel-methanol and OMEx-G on the in-cylinder pressure and heat release rate for the different operating points.

As shown in Figure 3, Diesel-methanol has a delayed combustion from low to high loads (the phenomenon is reversed at full load), compared to its diesel-gasoline counterpart. Such a consequence can be assigned to the differences in reactivity between gasoline and methanol, where gasoline has lower RON and, therefore, the start of combustion (SOC) takes place early (once the autoignition of the diesel occurs). Additionally, methanol has a higher RON than gasoline, meaning it can withstand a bigger compression before igniting. The delayed combustion of the methanol allows greater time for the fuel mixture once the HRF has been injected, and thus, a more homogeneous charge is obtained. At full load the trend might be reversed due, in part, to the tendency to pursue a combustion closer to CDC with multiple injections in order to be able to avoid the excessive rise rates common of RCCI at high loads [35], and therefore the combustion is more dependent on the HRF. From low to high loads, methanol leads to a more advanced SOI_{main}, to supply the necessary energy into the cylinder and achieve the calibration loads. Methanol used as LRF leads to longer combustion durations than diesel-gasoline, which is likely to increase the CO production as will be seen in a later section. This longer combustion can occur due to the lower PER used with methanol, as was proved by [26]. The 50% burn crank angle (CA50), shown in figure 4 further reflects the trends seen for the SOC; being interesting to notice how at full load the diesel-methanol case catches up with the diesel gasoline one. The explanation for this can be attributed to the more significant role at this load of the HRF.

Substituting the gasoline by methanol, a significant reduction of the peak pressure can be seen at medium-to-high loads. This benefit is however not maintained at low and full load. At full load, as was previously stated, the HRF starts playing a more important role on the combustion phasing and characteristics, while at low load the premixed combustion of the lower reactivity methanol promotes in-cylinder pressures that are quite similar or smaller than those found with gasoline. As will be seen in further sections, methanol is conducive to a reduction of the maximum pressure gradient, precisely because of that reduced reactivity.

PERs are lower for diesel-methanol because methanol has a significantly lower LHV than gasoline. This, in turn, can cause a reduction of the maximum in-chamber temperature [36], phenomenon that occurred for all the operating points using diesel-methanol, except the 100% load. These low PER values lead to a larger diffusive phase of the combustion especially at 25% and 80% loads, as shown in Figure 3.

OMEx presents important changes in LHV and reactivity compared to diesel (it has a lower LHV and a higher cetane number). The higher cetane number promotes in-cylinder local fuel mixtures with higher cetane number than diesel, which helps to reduce the combustion delay, as shown in the cases of 25% and 80% load. Moreover, it is useful to highlight that at 25% load the higher reactivity of OMEx results in shorter combustion duration, as can be appreciated in Figure 4. The heat release rate for this load shows a low temperature heat release (LTHR) (as does as well the 50% load operating condition). Contrarily, for the other loads, as the HRF becomes more predominant in the combustion phasing, the combustion duration of OMEx is longer, despite having a higher fuel reactivity, because it has a lower LHV than diesel. The 50% load condition is unusual in that an important delay of the start of combustion is observed. It is considered that this effect is originated by the lower heating value and the requirement of bigger fuel quantities to achieve the necessary energy for the combustion. Another result of the properties changes of OMEx is that its injection rates are considerably longer to achieve the energy levels necessary to reach the load conditions.

The full and 50% load cases do not present increases regarding the in-cylinder pressure curve against the diesel-gasoline one. In fact, the maximum pressure is almost the same at full load and is lower by 24.8% at medium load. Notwithstanding, low and medium-high load are comprised of increments of 18% and 3.7%, respectively.

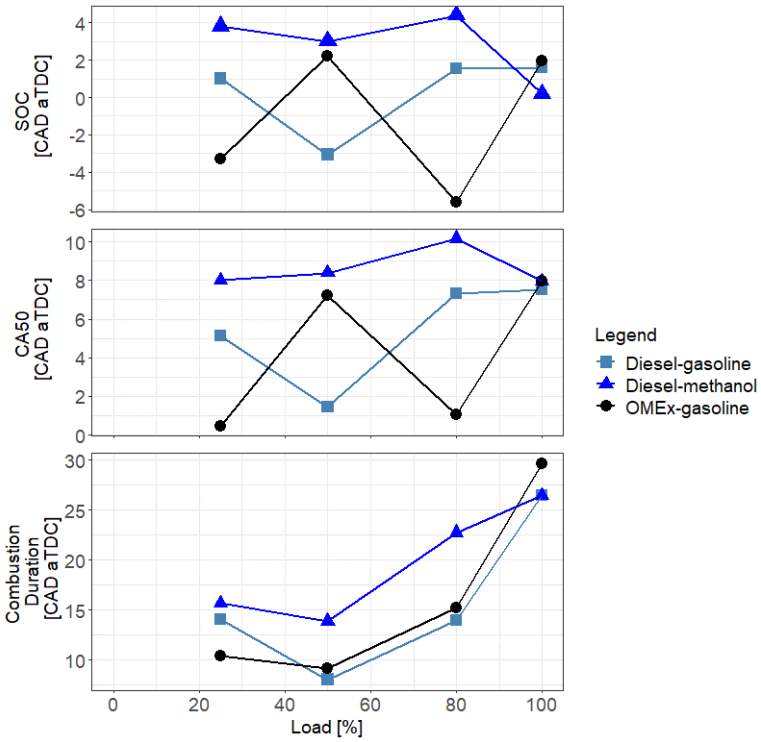

Figure 4. Effect of diesel-methanol and OMEx-gasoline on the start of combustion and combustion duration.

3.3. Performance results

The calibration process used was intended, among other objectives, to reduce the unacceptable pressure rise rates and peak cylinder pressures that can potentially damage the engine or cause excessive noise. This section evaluates the performance results of the studied fuels to evaluate its advantages and disadvantages. As shown in Figure 5, at low-to-medium load, diesel-methanol presents slight reductions in $BSFC_{eq}$, specifically 0.16% and 0.42%, as well as lower pressure gradients. In this sense, the octane number is a feasible cause since a higher value for the number indicates that methanol can be compressed further without igniting, acting as a damping property of the pressure rise rate (PRR). The reduction of the PRR is in agreement with prolonged combustion duration and is a desirable feature to reduce the mechanical damage and increased engine noise. PMEP for diesel-methanol is 0.35 bar over the diesel-gasoline calibration at the lowest load tested; this trend is reversed at higher loads, but the methanol-originated pumping losses do not add demand for additional turbocharger requirements. From Figure 5, it can be seen that using methanol has lower efficiency than gasoline, but recalling the huge difference of LHV that these two fuels possess can explain this phenomenon, as gasoline has a bigger energetic content. In terms of fuel consumption, it is seen that for both cases, substituting a fuel from the diesel-gasoline base case carries some fuel consumption penalties (around 40% more). By this reason, the fuel consumption was compared in terms of $BSFC_{eq}$, as explained in section 2.4, to exclude the effect of the LHV and compare the fuels by their fuel conversion efficiency instead of just the amount of fuel. Then, it can be said that OMEx produces an important decrease in $BSFC_{eq}$, since its energy conversion seems to prove to be better. On, non-equivalent, BSFC criteria, the OMEx is up to 176 g/kWh higher than the diesel case. OMEx also shows elevated pressure gradients compared to diesel-gasoline and diesel-

methanol, which can be attributed to the higher reactivity of the fuel. OMEx has higher pumping losses than diesel-methanol, but lower than diesel-gasoline as the required pumping mean effective pressure is lower. This coincides with findings reported in [16].The efficiency of the OMEx-gasoline case is inferior to that of diesel-gasoline, which can be appointed to the smaller LHV of OMEx and the reduction of the engine power, which is a direct consequence of its higher oxygen content [37]; the same assumption can be made for methanol. As with diesel-methanol, PMEP values are lower with OMEx, however this change is not as significant as with methanol.

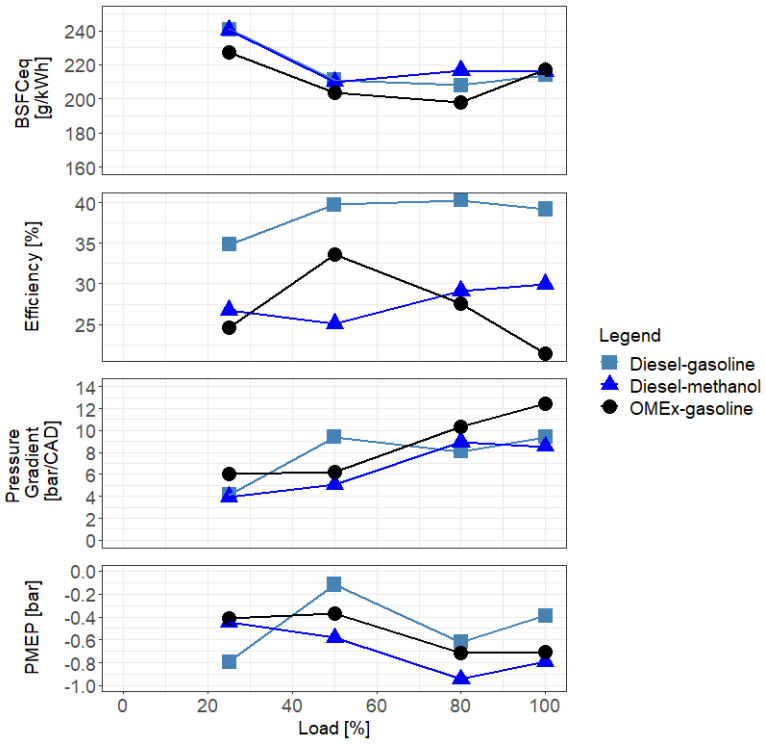

Figure 5. Effect of diesel-methanol and OMEx-G on performance parameters: BSFCeq, Efficiency, Pressure Gradient and PMEP.

3.4. Emission results

Regarding the emission parameters shown in Figure 6, the diesel-methanol blend results in increased NOx emissions for all the low-to-medium load operating conditions compared to the diesel-gasoline operation. On this point, is important to highlight that these values are very close to the EURO VI limits, so even though the values for methanol are higher, the difference does not exceed 0.05 g/kWh above the EURO VI limit. The extended combustion duration discussed in section 3.2 increases the residence time at high temperatures, crucial factor in the thermal mechanism of NOx formation. At full load, the NOx emissions levels from the diesel-methanol operation are lower by 2%. This reduction in NOx can be due to the fact that at higher loads, the combustion is more heavily influenced by the HRF which for both the base case (diesel-gasoline) and the diesel-methanol case is diesel. Hence, this slight difference in emissions could be a combination of different factors and not a direct consequence of the fuel used, since diesel is the fuel that governs both combustion conditions at this load.

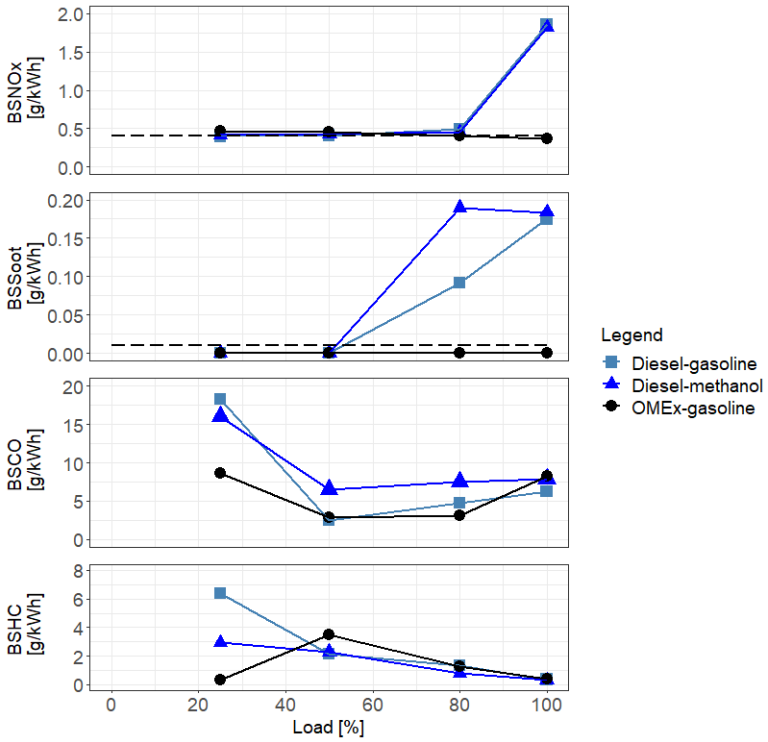

Figure 6. Effect of diesel-methanol and OMEx-G on emissions parameters: BSNOx, BSSoot, BSCO and BSHC.

HC emissions for diesel-methanol are similar to that of diesel-gasoline, at medium-to-full load, because of the reduction of premixing. So, even although the combustion duration is longer for the methanol, its late combustion timing balances out the effect of the residence times, thus allowing the reduction of incomplete combustion. The low load condition completely inverts this trend, with diesel methanol having over 6 g/kWh of unburned HC more than diesel-gasoline. Soot emissions are higher, mainly at 80% of engine load. This is somewhat counter-intuitive since the methanol has higher oxygen content than gasoline and that is expected to reduce soot formation. Then, other possible factors that can influence soot formation must be inspected, being the PER and the reduced percentage of LRF good candidates to explain this. To further explain the previous point, the percentage of methanol injected through the PFI, and its low soot properties, are not enough to counterbalance the effects of the soot formation.

The NOx emissions tend to increase as the ignition advances, which is the case for the medium-to-low load conditions for the OMEx-gasoline. However, the NOx values product of the calibration procedure remain close to the EURO VI limits. This is consequence of that the OMEx burning produces virtually no soot emissions due to the high oxygen concentration. Therefore, the air management system can be controlled to provide the adequate EGR levels to inhibit the NOx pollutants formation. Even at full load, where the diesel-gasoline exceeds by far the legislation limits, the OMEx values remained under it. Additionally, OMEx has a higher cetane number, which allows a higher power output [38] , thus, it is possible to reach the required load with lower NOx and soot emission than diesel-gasoline. At full load, a counter effect of the lower NOx is an increase of unburned HCs, but compared to the NOx improvement is quite small (0.06 g/kWh). The 50% load also shows more elevated HCs than the diesel-diesel gasoline case, and 0.05 g/kWh more of NOx emissions. For the remaining loads, the HC emissions

are lower with OMEx. CO emissions for medium-to-full load are quite similar to diesel-methanol, while at 25% load a reduction of 9.7 g/kWh is observed.

3.5. Fuel evaluation for EURO VI

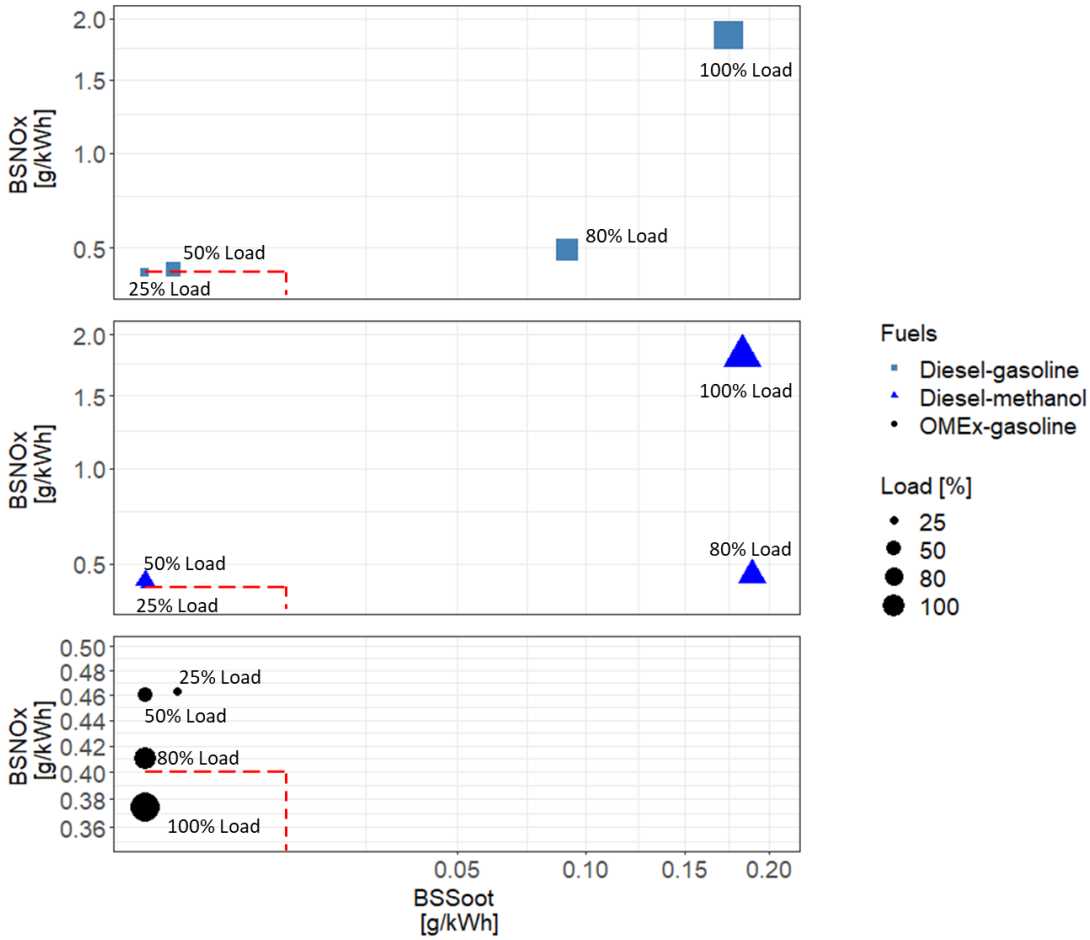

Figure 6. NOx-soot trade of for diesel-gasoline, diesel-methanol and OMEx-gasoline fuel blends. Left: All points. Right: Points that achieve soot limitations.

Conventionally, the NOx-soot tradeoff is one of the difficult challenges to face when trying to optimize ICEs to comply with regulations. Figure 6 shows the emissions commented in section 3.4, plotted in trade-off format to highlight more clearly the benefits of each fuel against the other. From Figure 6 it is evident that OMEx-gasoline has the best NOx-soot tradeoff, since is the only fuel combination for which, at least, all the operating points at all the loads have soot values under 0.01 g/kWh.

To further confirm the advantages and disadvantages of the fuel blends they were weighted against each other by means of the following function:

$$F = \sum \frac{X - X_{limit}}{X_{max} - X_{min}} \quad (2)$$

Where X represents the parameters to be evaluated ($BSFC_{eq}$, BSNOx, BSSoot, BSCO, BSHC); X_{limit} is the maximum value allowed by regulations or in the case of BSFCeq the value for diesel-gasoline; X_{max} is the maximum value for the parameter in the dataset

and X_{min} the minimum one. Emissions have been normalized with respect to EURO VI limits and $BSFC_{eq}$ by the aforementioned diesel-gasoline case.

Figure 7 illustrates the total sum of the penalization of each parameter evaluated for each fuel combination. The distribution of this penalizations by load is also distinguishable. The smaller the value of F, the closer the fuel combination is to ideal fuel and emissions performance. Then, it is evident that OMEx-gasoline is the fuel combination that better satisfies the trade-off between emissions and $BSFC_{eq}$. While diesel-methanol seems to perform the worst under the selected criterion, from sections 3.2 to 3.4 and previous research [26] is important to recall that it possesses characteristics that can help to achieve combustion targets and that could improve its desirability as a fuel when performing a complete life cycle analysis. Thus, more research on the fuel is recommended.

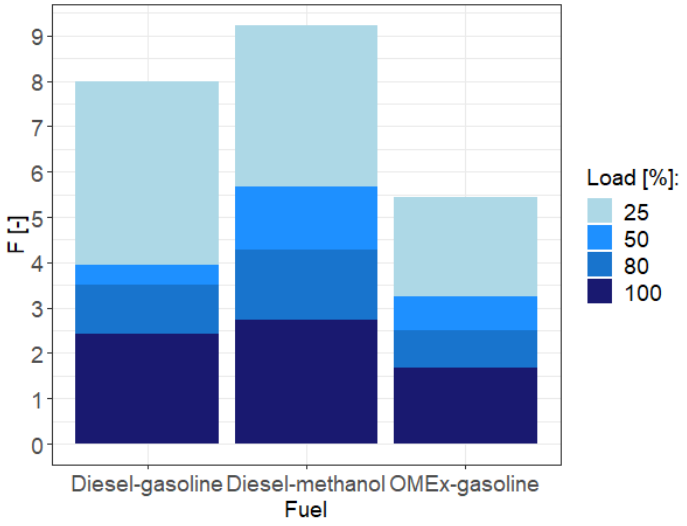

Figure 7. Penalization of the different fuel combinations by load.

3.6. Potential of OMEx to reach potential EURO VII conditions

As shown in the previous subsection, the advantages of substituting diesel by OMEx surpass the advantages of replacing the gasoline by methanol. This inspired further testing with the OMEx-gasoline combination. More specifically, an EGR sweep was performed in order to assess whether or not OMEx has the potential to fulfill the EURO VI emissions limitation for both NOx and soot emissions simultaneously -and even the potential EURO VII- without the need for after-treatment systems (ATS).

The tests were performed with constant parameters at 10, 25, 50, 75 and 100% loads to successfully isolate the effect of EGR on the combustion. In order to keep the total admitted mass constant, the turbine was set to turn together with low pressure and high pressure EGR. The respective settings are described in Table 6.

Table 6. Constant parameter for characterization of the effect of EGR on OMEx-gasoline.

	PER [%]	Pintake [bar]	Madm [g/s]	Mfuel [g/s]	SOI main [CAD bTDC]	SOI pilot [CAD bTDC]
10%	0	1.19	133	5.37	6	22

25%	59.8	1.38	154	6.5	29	44
50%	83.6	1.66	191	8.5	26	46
75%	58.3	2.66	287	15.9	10	30
100%	34.8	3.03	330	27.6	8	-
Max. dev.	±0.5	±0.01	±3	±0.1	-	-

The most straight forward solution to reduce the NOx production in the ICEs is the use of EGR. However, it is necessary to confirm if this effect is achieved without a great impact on the other emissions. Since more EGR reduces the in-cylinder temperature, inhibiting the oxidation reactions, a consequent increase for the HC and CO emissions is expected. As seen in Figure 8, in the case of OMEx, the impact of the EGR on the trade-off for NOx-HC and NOx-CO can be considered important (an increase of 2.5 g/kWh of HC for 0.7 g/kWh of NOx reduced for the former and 20 g/kWh of CO for 0.5 g/kWh reduced in NOx for the latter) the negligible effect the EGR increase generates on soot formation makes OMEx a formidable fuel to reach the emissions restriction levels. Furthermore, Figure 9 shows that increasing the EGR rate delays the combustion and increases the combustion duration. The delayed CA50 is the main cause of reduction of the engine torque.

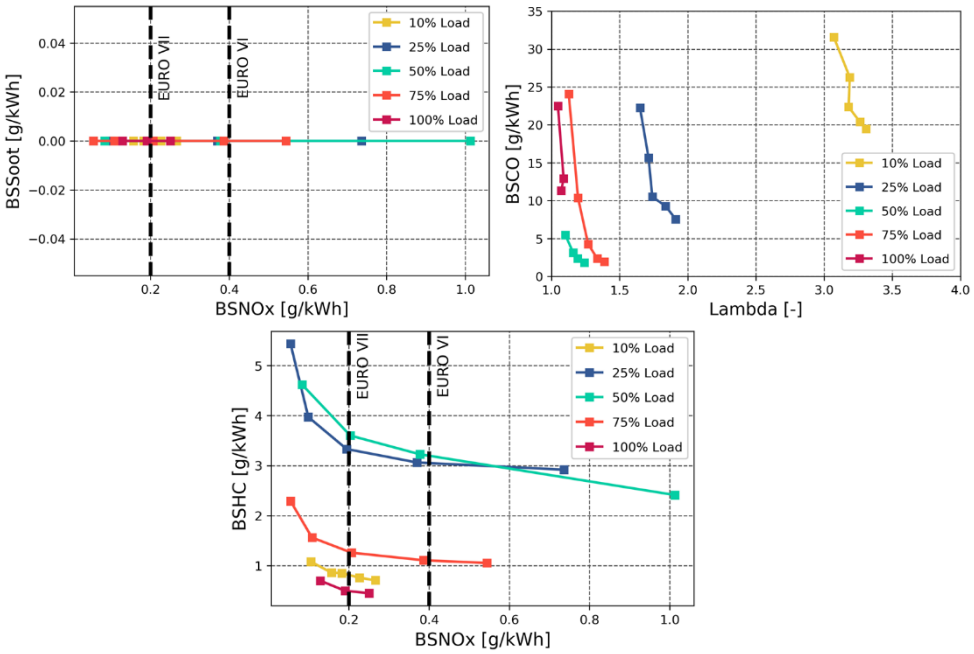

Figure 8. Engine emissions tradeoff during the EGR exploration tests. Top: BSSoot-BSNOx trade-off. Left: BSFCO-BSNOx trade-off. Right: BSCH-BSNOx trade-off.

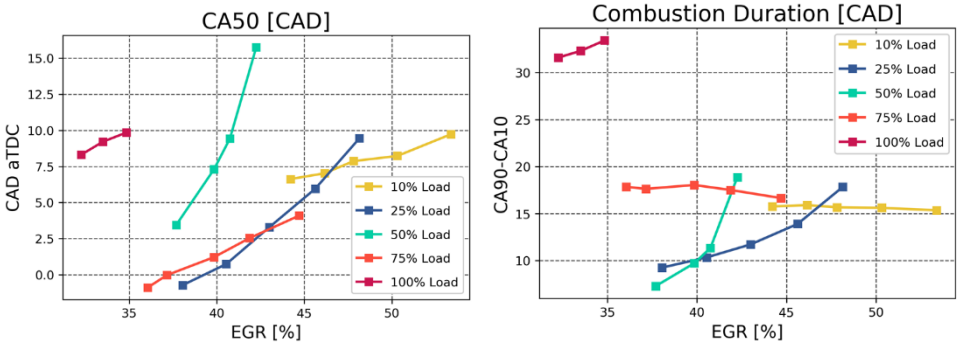

Figure 9. Combustion performance trade-off during the EGR exploration.

Figure 10 shows how increasing the EGR to reduce NOx formation also reduces the effective torque of the engine, thus affecting the fuel consumption. From the same Figure, it can be appreciated how going from EURO VI to the potential EURO VII has minimal effects on torque. Additionally, the increase on $BSFC_{eq}$ is still affordable as it does not surpass a 10% increase in the most critical case. Greater reductions than the potential EURO VII in NOx will penalize greatly the specific fuel consumption, as the injected fuel mass will have to increase to keep torque.

Finally, after evaluating of the conditions at the exhaust, it is apparent that the use of an ATS becomes necessary with the increase of the previously mentioned HC and CO emissions. Regarding the possible performance of this device, its acknowledged that it might be reduced due to the low exhaust oxygen concentration and high exhaust temperature.

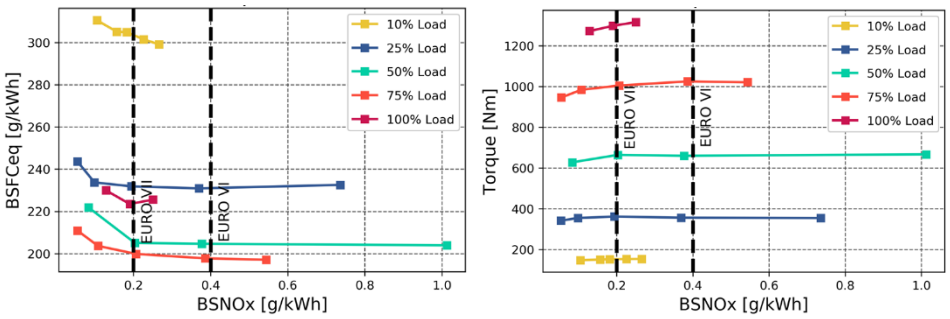

Figure 10. Engine performance tradeoff during the EGR exploration. Left: BSFCeq-BSNOx tradeoff. Right: Torque-BSNOx trade-off.

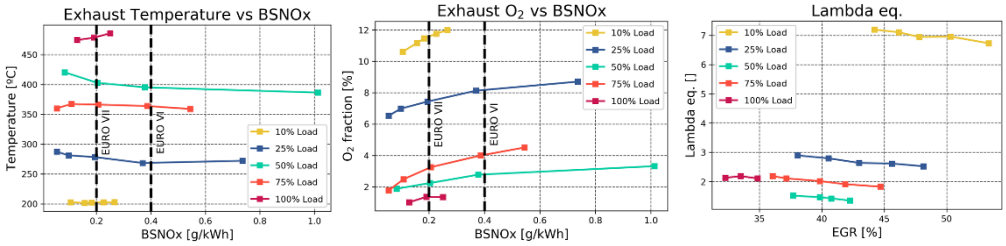

Figure 11. Exhaust boundary conditions at low NOx conditions.

4. Conclusions

This work discussed the potential of a DMDF operated engine towards the potential EURO VII emissions regulation compliance testing both diesel-methanol and OMEx-

gasoline. As a first step, the combustion was evaluated following a dedicated calibration at 1800 rpm and four different load levels for both fuel combinations tested, and comparing them in terms of combustion, performance and emissions to a base diesel-gasoline case under the same conditions. From this, the most relevant highlight are:

- Diesel-methanol DFDM has similar emissions performance as the diesel-gasoline base case in terms of NOx and HC because combustion is highly dependent on the HRF, which is diesel for both cases.
- Soot and CO emissions increase, when operating with diesel-methanol, except at 25 % load, where soot is equal, and CO is reduced by 12%.
- The use of diesel-methanol has minimal effects regarding $BSFC_{eq}$; however, fuel consumption is increased significantly because the lower LHV of methanol has to be compensated by injecting more diesel fuel and reaching a lower PER and LRF percentage to keep emissions under control.
- By using methanol, the mechanical integrity of the engine improves as the slower combustion of methanol produces lower pressure gradients and pressure peaks.
- The results for OMEx paint a better picture in terms of compliance with Euro VI regulations. When comparing the performance of the fuel against diesel-gasoline, it showed a reduction of NOx for the high-to-full load. Especially at full load, the reduction of NOx is 79.8% (from 1.85 g/kWh with diesel-gasoline to 0.37 g/kWh with OMEX-gasoline), more so, results do not exceed 0.46 g/kWh of NOx for the fuel combination.
- OMEx-gasoline soot production is the lowest of all the fuel combinations tested in this research.
- The use of OMEx instead of diesel generates BSFC increases of up to 176 g/kWh, when not compared in equivalent terms. The requirement of more fuel than its counterpart derives from its lower LHV. On the flipside, $BSFC_{eq}$ is lower for OMEx-gasoline for all loads but the 100% load, indicating a good energy conversion ratio.

From the first stage, after evaluating the superior performance of OMEx in the reduction of soot, an EGR sweep test was performed to evaluate the capabilities of reducing NOx emissions using OMEx and the impact on the other emissions and performance characteristics. It was found that:

- The OMEx-gasoline DFDM can accomplish reaching the potential EURO VII NOx levels (BSNOx=0.2g/kWh) and still be considered inside the low impact region of the tradeoff with BSFC and other regulated emissions.
- OMEx does have bigger CO and HC emissions than diesel-gasoline. This increase of CO and HC emissions makes necessary the use of an ATS, but depletion of oxygen at the exhaust may be a problem for its operation.
- The OMEx-gasoline blend produces virtually no soot; even at high EGR values because of its high oxygen content. This fact, in turn allows the reduction of pumping losses because the air management system can be calibrated without generating soot emissions above the imposed limit.

Finally, it can be concluded that the DMDF concept paired with relatively new fuels such as OMEx can help bridge the gap between the current state of ICEs and regulated

emissions, even up to the potential EURO VII for NOx. However, OMEx-gasoline has drawbacks that must be addressed before reaching the points of commercial availability, i.e. the higher CO and HC emissions coupled with reduced possible operability of an ATS. Future work might be an important tool to understand how to better solve these issues.

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Abbreviations

ATDC: After Top Dead Center

ATS: After Treatment System

$BSFC_{eq}$: Equivalent Brake Specific Fuel Consumption

BTDC: Before Top Dead Center

CAD: Crank Angle Degree

CDC: Conventional Diesel Combustion

CO: Carbon Monoxide

CO_2: Carbon Dioxide

DDFS: Direct dual fuel stratification

DI: Direct Injection

DMDF: Dual Mode Dual Fuel

EGR: Exhaust Gas Recirculation

FIS: Fuel Injection System

FSN: Filter Smoke Number

HC: Hydrocarbons

HRR: Heat Release Rate

HRF: High Reactivity Fuel

IMEP: Indicated Mean Effective Pressure

LRF: Low Reactivity Fuel

LTC: Low Temperature Combustion

LTHR: Low Temperature Heat Release

LHV: Lower Heating Value

NOx: Nitrogen Oxides

OMEx: Oxymethylene Ether

PER: Premix Energy Ratio

PMEP: Pumping Mean Effective Pressure

PFI: Port Fuel Injection

RCCI: Reactivity Controlled Compression Ignition

RON: Research Octane Number

SOI: Start of Injection

TDC: Top Dead Center

MON: Motor Octane Number

VGT: Variable Geometry Turbine

WHVC: Worldwide Harmonized Vehicle Cycle