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Utilization of Hydrotreated Vegetable Oil (HVO) in a Euro 6 Dual-Loop EGR Diesel Engine: Behavior as a Drop-In Fuel and Potentialities along Calibration Parameter Sweeps

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Abstract: This study examines the effects on combustion, engine performance and exhaust pollutant emissions of a modern Euro 6, dual-loop EGR, compression ignition engine running on regular EN590-compliant diesel and hydrotreated vegetable oil (HVO). First, the potential of HVO as a "drop-in" fuel, i.e., without changes to the original, baseline diesel-oriented calibration, was highlighted and compared to regular diesel results. This showed how the use of HVO can reduce engine-out emissions of soot (by up to 67%), HC and CO (by up to 40%), while NO_x levels remain relatively unchanged. Fuel consumption was also reduced, by about 3%, and slightly lower combustion noise levels were detected, too. HVO has a lower viscosity and a higher cetane number than diesel. Since these parameters have a significant impact on mixture formation and the subsequent combustion process, an engine pre-calibrated for regular diesel fuel could not fully exploit the potential of another sustainable fuel. Therefore, the effects of the most influential calibration parameters available on the tested engine platform, i.e., high-pressure and low-pressure EGR, fuel injection pressure, main injection timing, pilot quantity and dwell-time, were analyzed along single-parameter sweeps. The substantial reduction in engine-out soot, HC and CO levels brought about by HVO could give the possibility to implement additional measures to limit NO_x emissions, combustion noise and/or fuel consumption compared to diesel. For example, higher proportion of LP EGR and/or smaller pilot quantity could be exploited with HVO, at low load, to reduce NO_x emissions to a greater extent than diesel, without incurring penalties in terms of incomplete combustion species. Conversely, at higher load, delayed main injection timings and reduced rail pressure could reduce combustion noise without exceeding soot levels of the baseline diesel case.

Keywords: pollutant emissions; diesel engine; HVO; drop-in fuel; ECU calibration

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Citation: d'Ambrosio, S.; Mancarella, A.; Manelli, A. Utilization of Hydrotreated Vegetable Oil (HVO) in a Euro 6 Dual-Loop EGR Diesel Engine: Behavior as a Drop-In Fuel and Potentialities along Calibration Parameter Sweeps. *Energies* **2022**, *15*, 7202. https://doi.org/10.3390/en15197202

Academic Editor: Pavel A. Strizhak

Received: 31 August 2022
Accepted: 23 September 2022
Published: 30 September 2022

1. Introduction

Climate change, urban air pollution and energy sustainability have all been identified as major global concerns, with massive and direct implications for the road transport sector [1], whose primary power source continues to be the internal combustion engine (ICE) burning fossil-derived fuels [2]. Compression-ignition (CI) engines powered by conventional diesel oil account for a significant proportion of these ICEs, in both the passenger car, particularly in the European Union (EU), and heavy-duty sectors [3]. Because of their higher thermal efficiencies, they can reduce CO_2 emissions by 11 to 40% compared to gasoline counterparts [4]. However, since diesel engines consistently struggle with higher nitrogen oxides (NO_x) and particulate matter (PM), exhaust pollutant emission targets may still be difficult to meet, despite recent efforts to optimize both in-cylinder combustion [5] and after-treatment systems (ATS) [6]. Biomass-derived, diesel-like fuels are a promising way to address these issues while also reducing the environmental impact of CI engines in terms of greenhouse gas (GHG) emissions [7]. Vegetable oils, animal fats and waste cooking oils can be used as renewable feedstocks for the production of diesel fuel substitutes via various production processes that yield fuels with varying chemical compositions and properties [8].

Fatty acid methyl esters (FAME), commonly known as biodiesel fuels [9], are produced from oil-rich crops such as soy or rapeseed, using transesterification. Despite offering several advantages over petroleum-derived diesel oil (such as improved ignition, with associated reductions in CO, unburned hydrocarbons (HC) and PM emissions [10]), FAME fuel application is limited due to inconveniences associated with oxidation stability and cold flow properties [11]. Indeed, FAME may cause storage tank corrosion and ageing effects on the polymeric materials used in vehicle fuel systems [12]. Furthermore, at lower temperatures, FAME tends to form waxy crystals (which may cause problems during storage) and degrade the engine cold operability due to higher viscosity, which may affect injection behavior [13]. Because of these unfavorable properties of FAME, blending restrictions with conventional petroleum-derived diesel oil are typically set (e.g., EN590, in all EU member states, maximum 7%-vol) [14]. Nonetheless, one significant advantage of FAME is its high lubricity, which is beneficial for fuel-lubricated components in the fuel injection system [15].

A promising alternative to FAME could be hydrotreated vegetable oil (HVO), a synthetic liquid bio-fuel free of aromatics, oxygen and sulfur, made up of straight-chain paraffinic hydrocarbons (i.e., C_nH_{2n+2} alkanes) and produced by hydrotreating catalysis of triglyceride-based biomass [16] such as vegetable oils, animal fats, and waste materials [17]. Hydrotreating has several advantages over transesterification, including lower processing costs, greater feedstock flexibility as well as compatibility with conventional CI engines and fuel standards [18]: in fact, HVO can be used, either pure or blended with petroleum-based diesel in any proportion, with little to no modifications to existing CI engines [19]. The benefits of HVO over FAME have been extensively discussed in the literature [20,21], and include its relatively high heating value and cetane number, lower viscosity, lower cloud point and better cold flow properties [22]. Thanks to fewer unsaturated compounds in its chemical composition, HVO also displays better oxidation stability than FAME: aside from its excellent thermal stability, no deteriorating effects on polymeric parts of the fuel system have yet been revealed [12]. However, since HVO has poor lubricity in general, lubrication additives are required [15].

Because straight-chain alkanes have a lower activation energy to form free radicals and start the oxidation process than aromatic ring-shaped hydrocarbons with stable molecular structures, the ignition delay (ID) of HVO is generally shorter than that of conventional diesel oil [23]. As a result, HVO has an earlier start of combustion (SOC) than conventional diesel, especially at low and medium loads, whereas this effect may be weaker at higher loads [9]. Shorter ID implies a combustion with milder premixed phase, which can have an appreciable impact on combustion noise (CN) and exhaust pollutant emissions [24,25]. The majority of the available literature on the subject agrees that using HVO reduces CO, HC and PM emissions when compared to regular diesel [13,14,16,17,19,26]. The decrease in CO and HC emissions, especially at low loads (where their production is greater), is generally ascribed to higher cetane number and better ignition [14,27], whereas the absence of aromatic compounds, with their benzene rings acting as a precursor to the formation of soot in oxygen-lacking atmospheres, is widely regarded as the primary reason for PM reduction with HVO [23,28]. The literature findings on the effect of HVO on NO_x emissions, on the other hand, show no clear trends and are mostly inconsistent [14].

Despite several researchers have already investigated the effects of HVO on engine emissions and performance in the past, there is a lack of experimental studies that encompass the variety of calibration parameters potentially available on latest generation Euro 6 diesel engine platforms. Research in the vast majority of the literature is carried out on single-cylinder research engines [9], constant volume vessels [10,20], heavy-duty engines [28] or Euro 5 or older engines or vehicles [13,17,22,23,26].

The primary objective of this study is to compare the effects of HVO and diesel fuel on combustion and engine performance, first testing HVO as a "drop-in" fuel (without adjusting any ECU calibration parameter), and then varying some of the most important combustion-related parameters (start of injection, EGR, pilot injection timing and quantity, rail pressure). The effects of varying these parameters were studied at five distinct engine operating points and here illustrated at two (for the sake of conciseness), namely 1250 rpm at 2 bar of *bmep* (low load) and 2250 rpm at 15 bar of *bmep* (high load). The investigation was conducted on a 2.3-liter diesel engine for light-duty commercial vehicles applications.

2. Materials and Methods

2.1. Engine and Experimental Setup

The experimental data given below were gathered from a dedicated test campaign carried out on a fully instrumented 2.3-liter, four-stroke diesel engine prototype, provided by FPT Industrial (i.e., the OEM) and installed on a dyno test bench at the ICE Advanced Laboratory in Politecnico di Torino. Table 1 displays geometrical features as well as other parameters of the engine, whose baseline production version is adopted for modern Euro 6 light-duty commercial vehicles.

Table 1. Main technical specifications of the F1A Euro 6 engine.

Number of Cylinders	4
Displacement	2.3 l
Bore/stroke	88 mm/94 mm
Rod length	146 mm
Compression ratio	16.3:1
Valves per cylinder	4
Max power	102 kW
Max torque	400 Nm
Turbocharger	Single-stage VGT
Fuel injection system	Common rail injection system
EGR circuit type	Dual loop, water-cooled
Exhaust after-treatment system	DOC, DPF
Emission standard	Euro 6 d final

The engine under test is endowed with a high-pressure Common Rail fuel injection system with solenoid injectors, a variable geometry turbine (VGT), an intake throttle valve, an exhaust flap and a dual loop cooled EGR system that combines high-pressure (HP) and low-pressure (LP) EGR circuits. The HP circuit (also known as the short-route), which is the most widely adopted solution, collects exhaust gases from the exhaust manifold upstream of the turbine and recirculates them back into the intake manifold (downstream of the throttle valve and thus of the compressor) via either an HP EGR cooler or a parallel by-pass duct, allowing the engine to warm up faster in the latter case. An HP EGR valve can be used to control the amount of recirculated exhaust gases. The LP circuit (also known as long-route) is also included on the tested engine to allow experimental investigation of dual-loop EGR configurations. In the LP EGR circuit, exhaust gases are collected downstream of the ATS and recirculated back upstream of the turbocompressor through a dedicated LP EGR cooler. A three-way valve can be used to change the rate of exhaust gases coming from the LP EGR circuit. The ATS is made up of two components: the Diesel-Oxidation Catalyst (DOC) and the Diesel Particulate Filter (DPF), whereas the Selective Catalytic Reduction (SCR), which is present in the commercial application, was not available in the current project. A passive regeneration of the DPF is periodically required to prevent the system from clogging.

Low-frequency pressure and temperature measurements were taken at various points along the air, EGR and exhaust flow paths to provide a complete picture of engine variables. In addition, Kistler 6058A high-frequency piezoelectric transducers were used to measure in-cylinder pressure (every 0.1 °CA) in the four cylinders of the engine. An absolute pressure sensor, namely a Kistler 4007C piezoresistive transducer installed in the intake manifold, was used to reference these in-cylinder signals.

An AVL KMA fuel flowrate system was used to perform continuous measurements of the engine fuel consumption with 0.1% accuracy, while an AVL AMAi60 exhaust gas analyzer was used to measure the concentrations of NO_x/NO, HC, CO, CO_2, and O_2 both upstream and downstream of the ATS as well as the CO_2 concentration in the intake manifold. The engine-out soot emissions were measured under steady-state conditions using an AVL 415S smokemeter.

All of the above-mentioned measurement equipment was controlled by AVL PUMA Open 2, while IndiCom and AVL CONCERTO 5 were used for indicating measurements and data postprocessing, respectively.

2.2. Test Fuels

The fuels used for the experimental campaign were HVO and, as a reference, conventional petroleum-derived diesel B7 (with up to 7% biodiesel, in compliance with EN 590 regulations). ENI provided the main properties of both fuels, which are listed in Table 2. This table includes information such as the decreased density of HVO compared to diesel, its relatively high cetane number (which is mostly due to its paraffinic nature) and the complete absence of aromatics in HVO, which increases its H/C ratio. In addition to this, a more detailed analysis of the chemical composition of the tested HVO has pointed out that n-paraffines account for a total of 22.7% in mass, being n-C16, n-C17 and n-C18 the most relevant shares (9.9%, 1.7% and 9.5%, respectively).

Table 2. Diesel vs. HVO main properties.

Parameter	Unit	EN590 Diesel	HVO
Density at 15 °C	kg/m^3	830.6	777.8
Kinematic viscosity	mm^2/s	2.969	2.646
Dynamic viscosity	Pa·s	$2.47 \cdot 10^{-3}$	$2.06 \cdot 10^{-3}$
Cetane number	-	54.6	79.6
Monoaromatic	%v/v	20.1	0.50
Polyaromatic	%v/v	3.00	0
Total aromatic	%v/v	23.1	0
Flammability	°C	74.0	60.5
Lower Heating Value	MJ/kg	42.65	44.35
Hydrogen	%m/m	13.72	15.00
Carbon	%m/m	85.67	85.00
Oxygen	%m/m	0.61	0
Sulphur	mg/kg	6.50	0.53
FAME	%v/v	5.00	0.05
Approx. formula	-	$C_{13}\,H_{24}O_{0.06}$	$C_{13}H_{28}$

2.3. Exerimental Tests

The experimental test campaign was carried out on five steady-state engine operating points, four of which were deemed representative of the application of the tested engine to a light-duty commercial vehicle over a Worldwide Harmonized Light-Duty Test Cycle (WLTC), namely 1250 × 2, 1500 × 9, 1750 × 5, 2000 × 9 (expressed in terms of speed n [rpm] × *bmep* [bar]). In addition, a fifth operating point, 2250 × 15, was selected to represent the same vehicle's highway usage at constant speed (130 km/h). Experimental tests were conducted on all these five engine working points, but for the sake of conciseness, only the results pertaining to the engine operating points 1250 × 2 and 2250 × 15 will be discussed hereinafter.

The engine under test has a preliminary baseline (diesel-oriented) calibration that only uses high-pressure EGR. This baseline calibration was used as a starting point for preliminary steady-state experimental tests, which were performed to investigate the benefit pure HVO can bring as a "drop-in" fuel, i.e., using it as a completely interchangeable substitute for conventional EN590 B7 diesel oil, with no engine calibration adaptation.

In the second part of the investigation, for both fuels under test, some of the main engine calibration parameters, i.e., rail pressure (p_{rail}), electric start of injection of the main injection (SOI_{Main}), pilot 1 injection timing (DT_{PiI1}) and quantity (q_{PiI1}), HP and LP EGR valve positions, were investigated by means of suitable variable sweeps obtained through a "one-factor-at-a-time" approach, i.e., varying each of the previously listed variables throughout meaningful variation ranges, keeping the others fixed. Engine speed and torque were maintained constant at each engine operating point, regardless of changes in any of the previously listed calibration parameters, by allowing the engine testbench controller to adjust the injected fuel supply accordingly.

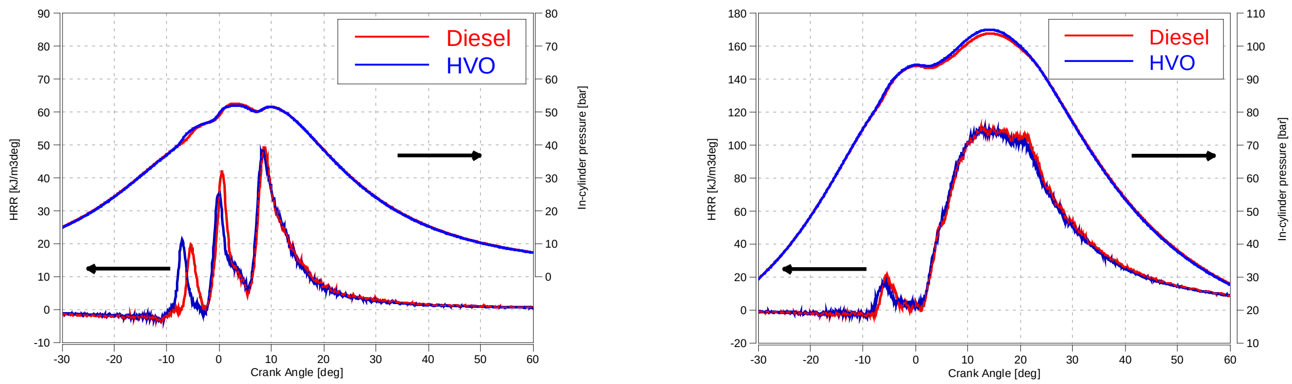
3. Results

3.1. HVO as a "Drop-In" Fuel

The first activity performed on the engine was to experimentally compare the two fuels (diesel B7 and HVO) on the selected steady-state operating points, to investigate the benefit that pure HVO can bring as a "drop-in" fuel, with no engine calibration adaptation. In the following, effects on combustion, engine performance and engine-out emissions are investigated.

3.1.1. HVO vs. Conventional Diesel Oil: Effects on Combustion

Figure 1 depicts two in-cylinder pressure traces (ensemble of 50 consecutive cycles, y-axis on the right indicated by an arrow) and the corresponding heat release rate (HRR, y-axis on the left indicated by an arrow) for the steady-state points 1250 × 2 (representative of a low-load and low-speed application) and 2250 × 15 (representative of a high-load and medium-speed application), although the study was performed for all the five selected part-load operating points previously discussed.



(**a**) (**b**)

Figure 1. In-cylinder pressure and heat release rate for diesel and HVO at 1250 × 2 (**a**) and 2250 × 15 (**b**).

Overall, the combustion of the engine running on conventional diesel or HVO is similar, most notably when the combustion of the main injection is considered. This is evident when examining the HRR peaks of the main injection depicted in Figure 1 (third and second most visible peaks on the HRR plots in Figure 1a,b, respectively), but this is true for all of the operating conditions investigated. As a matter of fact, fuel properties appear to have little effect on the combustion of the main injection regardless of the engine load. The heat produced by the pilot fuel combustion raises the temperature of the in-cylinder gases prior to the main injection, which, regardless of the specific fuel properties, ignites with a small ID upon reaching the hot burnt gases of the pilot combustion. As a result, when one or more pilot injections are used (as is standard for the calibration of latest generation Euro 6 diesel engines, except at full load), the effect of cetane number on main injection combustion is significantly reduced, resulting in a smaller difference in heat release between HVO and conventional diesel oil. Conversely, at lower load, the cetane number of the fuel shows a greater influence on the ID of the pilot injections (and thus on the mixture formation prior to ignition), most notably on the earliest one (namely pilot 2), due to relatively low in-cylinder temperatures. Indeed, the higher cetane number of HVO results in a noticeable advance of the start of combustion (SOC) of pilot 2, at 1250 × 2 (cf. Figure 1a). Furthermore, when compared to HVO, conventional diesel has a lower HRR peak during the combustion of this pilot 2 and a higher one during that of the subsequent pilot injection (namely, pilot 1). Again, this is due to the lower cetane number of conventional diesels, which may result in overmixing of some fuel from pilot 2 injection, causing it to burn later during the combustion of the following pilot 1 injection rather than during the earliest pilot shot.

3.1.2. HVO vs. Conventional Diesel Oil: Effects on Engine Performance and Emissions

Figure 2 reports, respectively, a comparison between the fuels at the steady-state points 1250 × 2 and 2250 × 15, in terms of engine-out pollutant emissions and engine performance. In particular, each figure shows engine-out soot, CO, and HC emissions, as well as *bsfc* and CN, on several y-axes stacked in function of engine-out NO_x emissions (with decreasing EGR values corresponding to increasing NO_x values). The results for all the tested points are reported in Table 3. Relative changes (in percentage) between the two fuels were considered for emissions and fuel consumption, according to Equation (1), while absolute changes were considered for brake thermal efficiency (η_u) and CN, according to Equation (2). Significant reduced values are highlighted with green color while the increased ones are reported with red color.

$$\Delta_{\%}x\ (relative\ change) = 100 \cdot \frac{x_{HVO} - x_{diesel}}{x_{diesel}} \quad (1)$$

$$\Delta x\ (absolute\ change) = x_{HVO} - x_{diesel} \quad (2)$$

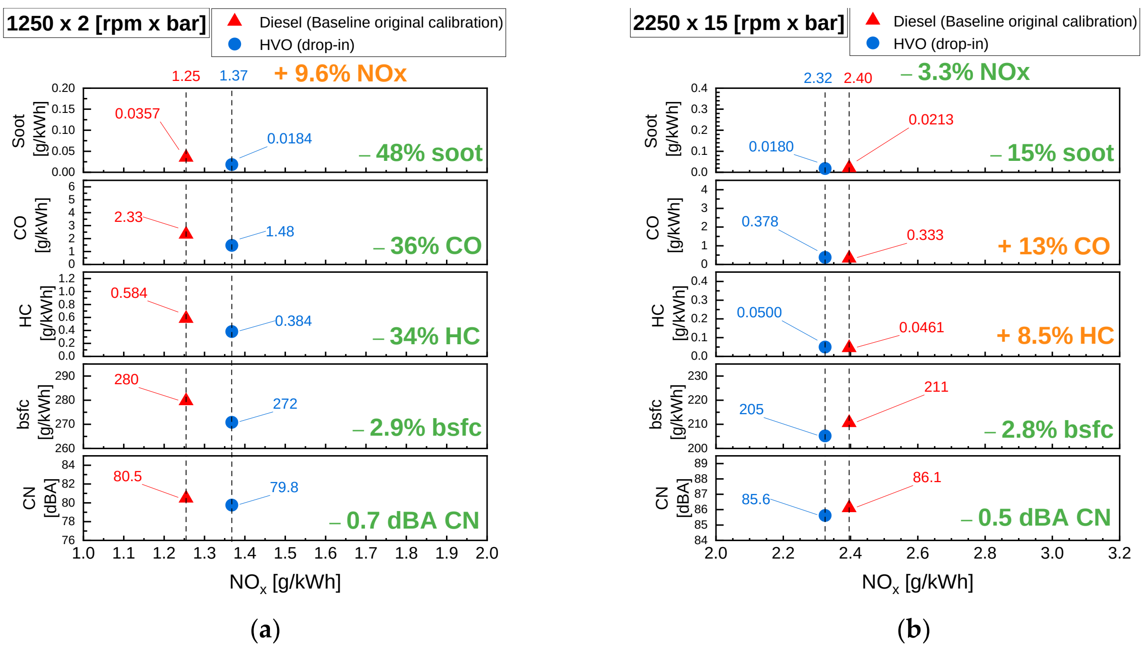


Figure 2. Emissions, fuel consumption and combustion noise for diesel and HVO at 1250 × 2 (**a**) and 2250 × 15 (**b**). Results with the baseline (diesel-oriented) ECU calibration.

Table 3. Diesel vs. HVO comparison on 5 baseline calibration points (percentage variations).

Speed × *bmep* (rpm × bar)	$\Delta_{\%}$Soot [%]	$\Delta_{\%}$CO [%]	$\Delta_{\%}$HC [%]	$\Delta_{\%}NO_x$ [%]	$\Delta_{\%}$*bsfc* [%]	$\Delta_{\%}CO_2$ [%]	$\Delta_{\%}$*vfc* [%]	$\Delta\eta_u$ [%]	ΔCN [dBA]
1250 × 2	−48	−36	−34	+9.6	−2.9	−3.5	+3.7	−1.0	−0.7
1500 × 9	−67	−33	−44	−4.8	−3.3	−4.7	+2.9	−0.2	−1.2
1750 × 5	−56	−18	−30	−6.8	−3.4	−3.1	+3.1	−0.4	−0.9
2000 × 9	−46	−15	−25	−15	−3.2	−4.5	+3.4	−0.7	−0.8
2250 × 15	−15	+13	+8.5	−3.3	−2.8	−3.5	+3.8	−1.0	−0.5

NO_x emissions measured from the engine running on HVO are generally lower than those detected when running on reference diesel fuel (albeit not by as much as for the other pollutant species, with NO_x reductions ranging from −3.3% at 2250 × 15 to −15% at 2000 × 9, as reported in Table 3). The only exception is at low load (1250 × 2), where NO_x emissions for HVO increases (+9.6%, cf. Table 3 and Figure 2a). According to the available literature on the topic, it is still unclear whether hydrotreated renewable fuels reduce or increase NO_x emissions when compared to diesel. A higher cetane number results in a shorter ID and faster combustion. However, as the current results suggest, a shorter ID does not always guarantee NO_x reduction [29], and the results may vary depending on the actual engine load conditions and calibration.

HVO reduces soot emissions for all the examined operating conditions (reductions range from −15% at 2250 × 15 to −67% at 1500 × 9, see Table 3). Soot formation is a complex phenomenon that is influenced by a wide variety of parameters such as EGR rate, ID, and fuel properties (cetane number, density, viscosity and presence of aromatic and polyaromatic compounds). Since EGR rate is nearly identical (ECU calibration remains unchanged at each engine operating point), the smoke differences between HVO and diesel could be entirely attributed to different fuel properties, changes in ID and absence of aromatic compounds. In particular, the benefits of HVO properties appear to outweigh any potential detrimental effect on soot caused by milder premixed combustion and shorter ID, especially at low load, where soot reduction is nearly halved (−48% at 2 bar of *bmep*), despite the fact that the amount of soot produced at low load is generally small, so its reduction is not particularly significant in any case. Soot, on the other hand, becomes a source of concern as load rises. Rising loads yield higher in-cylinder temperature and fuel injection pressure, which speed up the air−fuel mixing process for the whole fuel injection pattern and for both fuels, reducing ID variations in the overall combustion development, which is witnessed by very similar HRR traces in Figure 1b. Therefore, the reduction in soot (up to −67% at 1500 × 9, cf. Table 3) can be attributed primarily to virtually no aromatics in HVO and to its lower density and viscosity. The absence of aromatic compounds in a fuel generally results in less soot formation because these compounds are more likely to act as soot precursors. Moreover, HVO has lower density and viscosity compared to conventional diesel, as well as a narrower distillation temperature range (that describes the evaporation of the liquid fuel as a function of temperature). This likely enhances a faster evaporation rate and more uniform air−fuel mixture throughout the fuel cloud [13], ultimately contributing to soot formation reduction.

The results reported in Table 3 also show that HVO emits less incomplete combustion species (HC and CO) than the reference diesel fuel, which is consistent with the available literature data. The only exception is the highest load investigated (2250 × 15), but the percentage increase in CO (+13%) and HC (+8.5%) is far from being significant, because their absolute values are quite low due to the high in-cylinder temperatures involved in the combustion at this load. The most valuable HC reductions are approximately −30% at the lowest loads (1250 × 2 and 1750 × 5), while relevant CO reductions are −18% at 1750 × 5 and −36% at 1250 × 2. HVO is thought to reduce incomplete combustion emissions primarily due to its improved ignition behavior, which is attributed to its high cetane number and narrow distillation range. The distillation curve is an important indicator of the fuel suitability since fuel injection in diesel engines occur into hot in-cylinder intake charge and evaporation is important, especially at low load and during cold start operation. Fuels with low distillation curves (such as HVO) exhibit improved evaporation and mixing with the intake charge, as well as increased reactivity at low combustion temperatures, resulting in lower CO and HC emissions [30], especially at low-to-medium loads.

In addition to the previously reported reductions in engine-out pollutant emissions, HVO also reduces brake specific fuel consumption (*bsfc*) when compared to the reference diesel fuel. This is achieved at each of the operating points investigated (*bsfc* reductions range from −2.8% at 2250 × 15 to −3.4% at 1750 × 5, cf. Table 3). However, because HVO has a lower density than the reference diesel fuel (777.8 vs. 830.6 kg/m^3, a −6.35% difference), volumetric fuel consumption (*vfc*), calculated as *bsfc* divided by the fuel density, increases by about 3 ÷ 4% on average. Both *bsfc* and *vfc* are worthwhile to compare: the retail fuel market typically sells fuel in volume units (e.g., in €/l), so end-users consider *vfc* as an important parameter, whereas the engine's tank-to-wheel CO_2 emission is dependent on mass-based fuel consumption (i.e., *bsfc*), although a direct correlation between exhaust CO_2 and *bsfc* can be obtained only for fuels with the same carbon content, H/C ratio and not containing oxygen, while HVO is mainly composed of paraffins in the range of n-C15 to n-C18, and diesel consists of hydrocarbons in the range of C9–C30 [17]. In this case, the reduction in engine-out CO_2 measured by the gas analyzer (from −3.1% at 1500 × 9 to −4.7% at 1750 × 5, as reported in Table 3) is found to be quite well correlated with *bsfc* reductions. Nevertheless, well-to-wheel CO_2 emissions in the atmosphere are reported to potentially be up to 10 times lower than those of fossil fuels if HVO is used, since it is made out of renewable feedstocks that absorb CO_2 while growing [31].

As far as brake thermal efficiency is concerned, all of the tested operating points show only minor to negligible penalties when the engine runs on HVO. There does not appear to be any dependence on the specific engine load or speed regime, either: penalties in η_u with HVO range from −1% at 1250 × 2 and 2250 × 15 to −0.2% at 1500 × 9 (cf. Table 3).

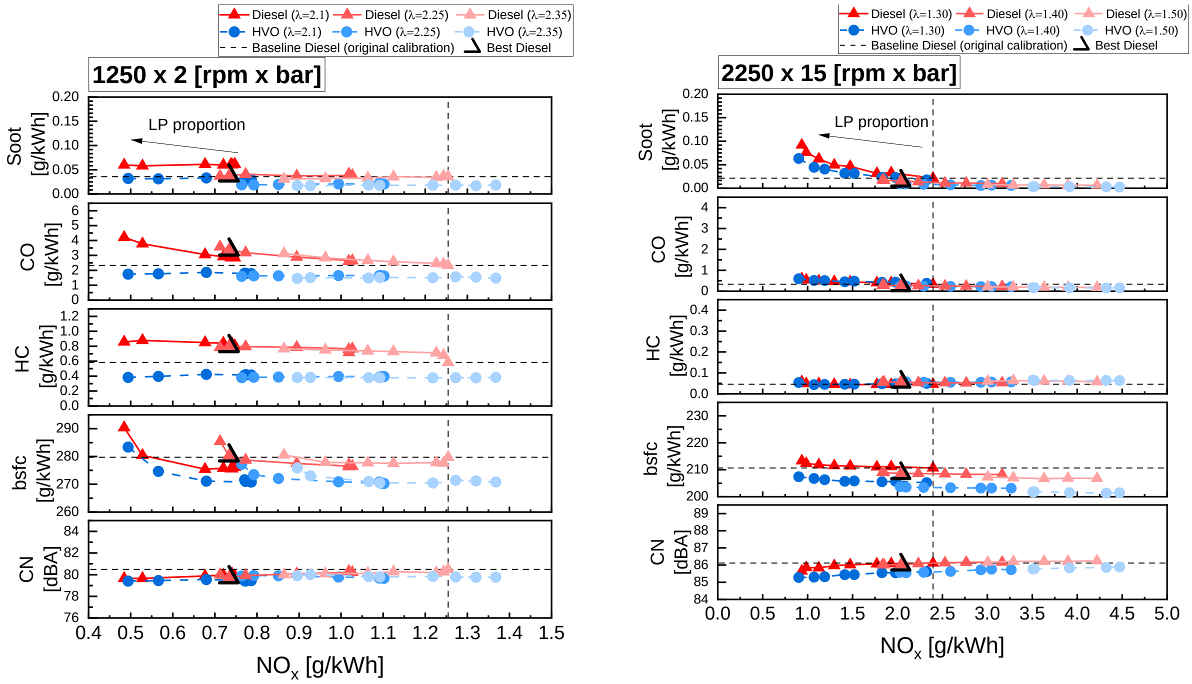
3.2. Sweeps of Calibration Parameters

Several single-parameter sweeps were performed at the same steady-state engine operating points investigated in the previous section, varying EGR valve positions and fuel injection calibration parameters. The effects of carefully changing these parameters on several trade-offs (vs. NO_x emissions, as previously carried out) are shown, with the goal of first providing an insight into the benefits on regulated pollutant emissions and engine performance given by replacing diesel with HVO, and then obtaining some indications on how to possibly recalibrate the tested engine to fully exploit the peculiar properties of HVO.

3.2.1. HVO vs. Conventional Diesel: Effect of EGR and EGR Split

The previously discussed baseline (diesel-oriented) calibration only includes HP EGR. In this part, the split between HP and LP EGR supplied by the short-route and the long-route paths, respectively, is investigated and referred to as "EGR split". In particular, various proportions of HP and LP EGR are investigated at several fixed λ values (i.e., the adimensional ratio between the air mass and the fuel mass, referred to as the stoichiometric value), to assess the benefits that could be obtained by adjusting the EGR rates and the EGR split when HVO is used.

Figure 3 shows brake specific soot, CO and HC, along with combustion noise and fuel consumption, on various y-axes stacked in function of brake specific NO_x emissions (with decreasing EGR values corresponding to increasing NOx values), for the same two engine operating points already discussed, i.e., 1250 × 2 (Figure 3a) and 2250 × 15 (Figure 3b). Tests using diesel fuel are represented by red solid lines with triangles, while tests using HVO are represented by blue dashed lines with circles. Darker shades denote lower λ, whereas lighter shades higher λ, for both fuels. As a reference, black dashed horizontal and vertical lines correspond to the values retrieved from the baseline diesel calibration (the same reported as red triangles in Figure 2a,b).



(**a**) (**b**)

Figure 3. Emissions, fuel consumption and combustion noise for diesel and HVO at 1250 × 2 (**a**) and 2250 × 15 (**b**). Results along EGR split sweeps at several fixed λ values.

Starting from the right end points of each trend (points obtained by allowing only HP EGR through the short-route path), LP EGR proportion, which indicates the contribution of LP EGR in the combined EGR loops, was progressively increased while the EGR through the short-route path was reduced all the way to the left end point of the trend, for which the HP EGR rate is zero (LP EGR only) and the minimum NO_x (for each λ value) is detected.

All the EGR trade-offs depicted in Figure 3 highlight that HVO has a better behavior compared to diesel, relative to all the examined pollutant emissions, as well as *bsfc* and CN. For each operating point, an "optimum" diesel calibration (labelled as "best Diesel" in the legend) has been selected through the use of a proper objective function (meant to deliver the best compromise in terms of exhaust pollutant emissions, *bsfc* and combustion noise) and highlighted with thick black edges. This "optimum" will be used in the next sub-sections as the reference calibration to explore all of the other parameter sweeps (VGT, p_{rail}, SOI_{Main}, DT_{PiI1} and q_{PiI1}).

Increasing the amount of EGR (hence decreasing λ values) is used primarily to reduce engine-out NO_x emissions. Nevertheless, NO_x can be also reduced at constant λ varying the EGR split in favor of LP EGR, as can be seen in Figure 3. This is due to the cooling effects on the LP share of recirculated exhaust gas given by their expansion across the turbine and by the intercooler (it is worthwhile noting that tests were carried out keeping the testbench intercooler efficiency around 85% by a PID controller that changes the coolant water flowrate in the intercooler accordingly). As a result, complete oxidation of the fuel may become more difficult to achieve, as evidenced by increasing CO trends with increasing LP EGR proportion, particularly at low load and with conventional diesel. Nevertheless, when the engine is run on HVO, this trend is much less noticeable, if not completely absent, once again denoting its superior ignition and combustion behavior even at lower temperatures. Moreover, HVO significantly drops HC-NO_x and CO-NO_x trade-off lines at 1250 × 2 by up to −60% when compared to diesel, at a NO_x level around 0.5 g/kWh, thus remaining well below the original baseline diesel calibration even at the highest EGR rate and LP EGR proportion. If the reduction in HC-NO_x and CO-NO_x trade-offs is especially beneficial at lower loads, the reduction in soot-NO_x trade-off (by up to −30%) brought about by HVO becomes more important at higher loads, as witnessed by Figure 3b.

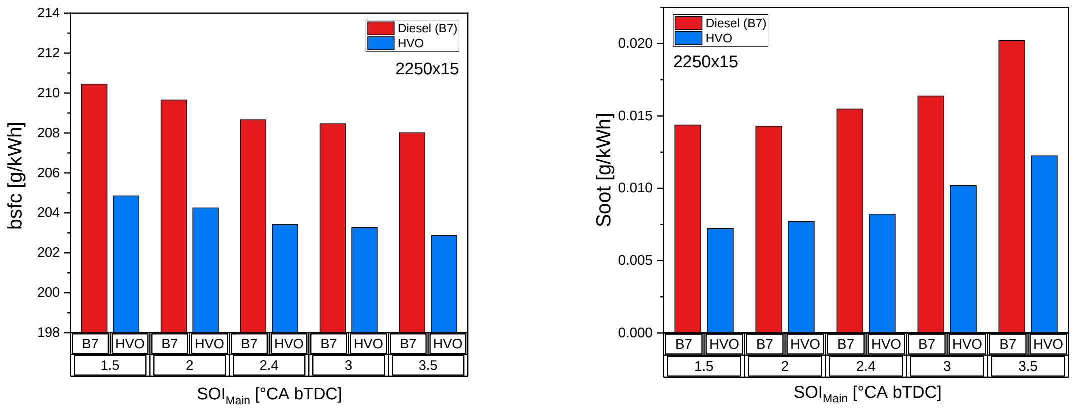
In general, as LP EGR proportion increases, so does the exhaust backpressure, owing to higher exhaust gas flowrate through the turbine (and smaller share of exhaust gases recirculated directly back into the intake manifold via the HP EGR path). The pressure differential between intake and exhaust manifolds rises as a result, as do pumping losses and *bsfc*, whose increasing trend is shown at 1250 × 2 in Figure 3a for both fuels, whereas *bsfc* trends are nearly flat at 2250 × 15 (cf. Figure 3b), possibly due to the effect of turbocharger efficiency, which appears only at higher loads while it is marginal at lower loads. Indeed, the higher the LP EGR proportion, the more exhaust enthalpy available at the turbine inlet, the higher the turbocharger efficiency, partially offsetting the negative effect of increased pumping losses.

Ultimately, the substantial reduction in engine-out soot, HC and CO levels achieved by HVO could give the possibility to run the engine with higher proportion of LP EGR, to limit NO_x emissions, especially at lower load.

3.2.2. HVO vs. Conventional Diesel: Effect of SOI_{Main}

Combustion phasing (and the associated HRR) has a direct impact on fuel consumption, and it is typically determined to achieve the lowest possible *bsfc* (for a given brake power), while adhering to proper constraints on engine-out pollutant emissions or engine performance parameters (for example, constrains on soot at higher loads). In this subsection, SOI_{Main} was varied (i.e., advanced or retarded) to investigate how adjusting combustion phasing could result in different outcomes for both examined fuels, taking into account their specific characteristics.

Figure 4a (referring to high load conditions, i.e., 2250 × 15) shows that, to reduce fuel consumption, the fuel injection pattern should be advanced. However, combustion phasing also plays a significant role on the whole in-cylinder combustion development, and thus has a significant impact on exhaust pollutant emissions and combustion noise, as well. For example, Figure 4b highlights how advancing SOI_{Main} at 2250 × 15 would, in turn, worsen engine-out soot emissions for both fuels, and similar trends were found also for other medium-to-high-load engine operating points (e.g., 1750 × 5 and 2000 × 9), which are not reported here. This behavior could be attributed to slower soot oxidation rates when advanced fuel injection patterns are employed (since they lead to lower exhaust temperatures during the expansion stroke), outweighing the increase of time available for the air-fuel mixture preparation, which could hinder the creation of fuel-rich local zones that directly affect the formation process of soot particles. In addition to soot increase, advancing the fuel injection pattern would generally increase NO_x emissions as well, due to higher in-cylinder peak temperatures, and combustion noise level, too. As a result, if the goal is to optimize fuel consumption, exhaust pollutant emissions and combustion noise, SOI_{Main} cannot be located for minimum *bsfc*, and a suitable compromise should be chosen. Nevertheless, due to the differences in specific fuel characteristics, this compromise may differ depending on whether HVO or conventional diesel is used. For example, taking into account the SOI_{Main} sweeps reported in Figure 4, soot reductions with HVO range between 38 and 50% when compared to conventional diesel, which is consistent with the findings discussed in Section 3.1, whereas *bsfc* reductions are around 2.5%. Therefore, the lower pollutant emissions provided by HVO could be exploited to reduce fuel consumption even further through more advanced combustion phasing, without exceeding soot levels of conventional diesel. Conversely, if the primary goal is to reduce NO_x emissions, more delayed combustion phasing could be employed, without incurring unacceptable penalties in *bsfc*.

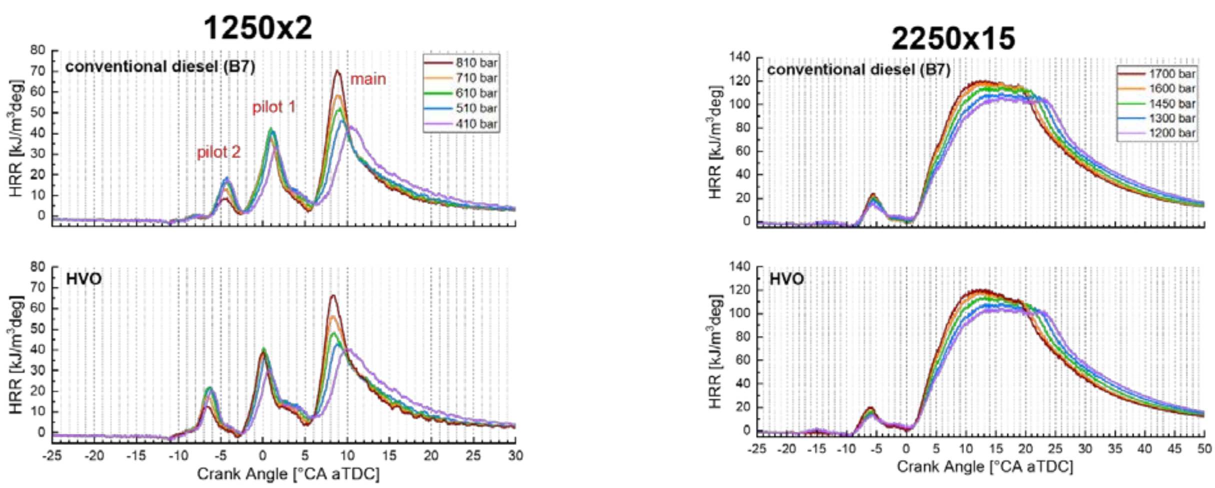


(**a**) (**b**)

Figure 4. Influence of SOI_{Main} on *bsfc* (**a**) and soot (**b**) for diesel and HVO at 2250 × 15.

3.2.3. HVO vs. Conventional Diesel: Effect of p_{rail}

Rail pressure is a critical calibration parameter that influences the formation of air−fuel mixtures, which has a direct impact on pollutant emissions and engine performance. Figure 5 depicts the response of both investigated fuels to changes in fuel injection pressure (p_{rail}) in terms of HRR, at 1250 × 2 (cf. Figure 5a) and 2250 × 15 (cf. Figure 5b). Green solid lines represent tests performed with the baseline p_{rail} values of the reference diesel-oriented calibration. For both fuels, orange and red lines indicate p_{rail} higher than baseline, while blue and purple lines denote p_{rail} lower than baseline.



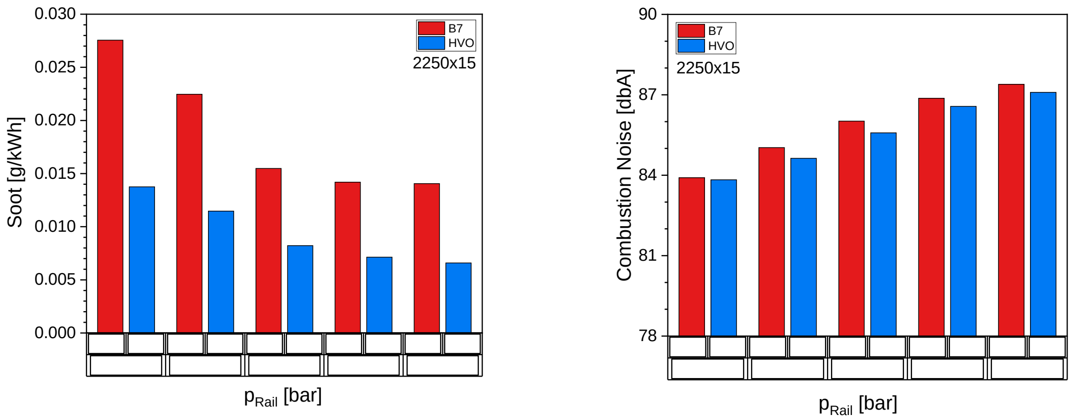
(**a**) (**b**)

Figure 5. Influence of p_{rail} on HRR for diesel and HVO at 1250 × 2 (**a**) and 2250 × 15 (**b**).

When p_{rail} is raised above baseline values at 1250 × 2 (i.e., p_{rail} > 610 bar), the main HRR rises and begins sooner. The increased fuel injection pressure causes the fuel spray of the main injection to reach faster the pilot-burning zone, resulting in a shorter ID of the main injection (ID_{Main}) and a faster, closer-to-TDC combustion process, due to hotter in-cylinder conditions. However, as p_{rail} values are raised from 610 to 810 bar, the earliest HRR of pilot 2 declines, implying that less fuel is burned during this phase, whereas no opposite behavior (i.e., an increase of pilot 2 HRR) occurs when p_{rail} is decreased from 610 to 410 bar. This may happen because injecting the fuel earlier and with faster spray penetration velocities (due to p_{rail} greater than 610 bar), combined with the low boost pressure values usually featured at low load, may cause wall-wetting phenomena, which do not seem to occur for rail pressures lower than 610 bar as well as for higher loads.

Indeed, at 2250 × 15, the highest pilot 2 HRR belongs to the highest p_{rail} case, which provides more energy and momentum to the fuel at the periphery of the spray to break up into smaller droplets, evaporate faster and ignite. On the other hand, no significant difference is detected when the behavior of the main HRR profile is observed at higher load. Since the start of the main combustion begins after the TDC in both cases, a higher p_{rail} causes the combustion to occur in a hotter and higher-pressure environment, resulting in steeper HRR profiles and greater HRR peaks. Ultimately, the effects of p_{rail} variations on HRR profiles are visible for both tested fuels and do not seem to be heavily affected by specific fuel properties.

Evaluating pollutant emissions and engine performance of both fuels confirms that, regardless of fuel injection pressure, HVO has a better behavior than conventional diesel. Figure 6 depicts, for example, how HVO roughly halves exhaust soot concentration along the whole p_{rail} sweep at 2250 × 15, while maintaining slightly lower combustion noise levels compared to conventional diesel. The decreasing soot and rising noise trends with increasing rail pressure values are rather straightforward to explain. Higher p_{rail} improves fuel atomization, expands the interface between spray particles and air and reduces evaporation time. Therefore, the air entrainment into the fuel spray and the mixture formation process are greatly enhanced and the fuel distribution is more uniform, thereby increasing in-cylinder pressure and temperature values, which ultimately favor soot oxidation process, resulting in up to 50% lower soot emissions for both tested fuels at 2250 × 15 (cf. Figure 6a). Conversely, as a result of the previously mentioned steeper HRR profiles provided by higher p_{rail}, combustion noise gradually increases, up to 3 dBA for both tested fuels at 2250 × 15 (cf. Figure 6a), establishing a clear trade-off with soot.



(**a**) (**b**)

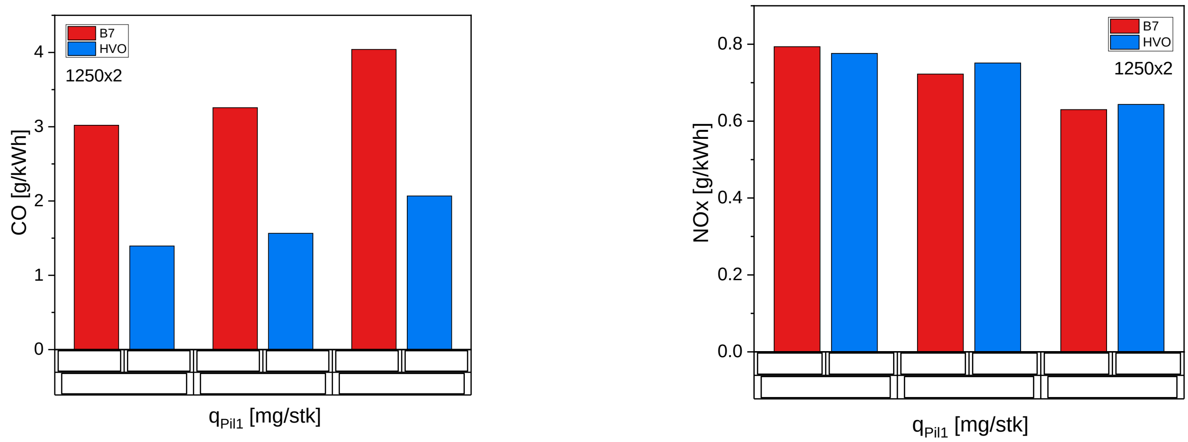
Figure 6. Influence of p_{rail} on soot (**a**) and combustion noise (**b**) for diesel and HVO at 2250 × 15.

According to what has been said, the lower soot emissions provided by HVO could be exploited to improve combustion noise at higher loads through lower rail pressure, without exceeding soot levels of conventional diesel. The same trend was observed at 1250 × 2, but it was not reported here because soot emissions and combustion noise levels are less important at lower loads.

3.2.4. HVO vs. Conventional Diesel: Effect of Pilot 1 Strategy (q_{Pil1} and DT_{Pil1})

Pilot injection strategy is critical in determining in-cylinder pressure and temperature conditions prior to the main injection event, thus influencing its ID and, consequently, its proportion of premixed and mixing-controlled combustion phases. Due to its really small injected quantities, pilot 2 (i.e., the most advanced pilot injection) calibration parameters were found to have less impact than those of pilot 1 in the tested engine operating points. For this reason, only pilot 1 injection parameters (i.e., fuel injection quantity, or q_{Pil1}, and dwell-time, or DT_{Pil1}) were investigated performing single-parameter sweeps, while keeping q_{Pil2} and DT_{Pil2} (respectively, fuel injection quantity and dwell-time of pilot 2) constant and equal to their baseline calibration values.

The impact of q_{Pil1} on engine-out CO and NO_x emissions at 1250 × 2 is shown in Figure 7. Comparing HVO with conventional diesel, it is possible to see how HVO significantly reduces incomplete combustion species emissions (similar findings for HC emissions were found as well), while the difference in NO_x is not stark. When it comes to the trends as a function of q_{Pil1}, the larger q_{Pil1}, the lower CO but the higher NO_x. Larger pilot 1 quantities generate more heat during combustion, which ultimately results in higher in-cylinder gas temperatures prior to the main injection. This results in higher peak temperatures during pilot 1 combustion and in a shorter ID_{Main}, which means that the main combustion develops further in the mixing-controlled phase, but with a lower peak temperature value across its diffusive flames. Because pilot injection quantities are comparable to main injection at low load (and the larger q_{Pil1}, the less the fuel in the main injection), the first effect apparently cause NO_x production during pilot 1 to outmatch the second effect, which causes a decrease in NO_x produced during the main combustion due to smaller amount of fuel burned in the premixed combustion phase [24,32]. On the other hand, since larger quantities of pilot 1 result in shorter ID_{Main}, combined with a lower fuel quantity in the main injection, the amount of fuel burned during the late rate-limited combustion phase is reduced [32], meaning that it is less likely that CO molecules are not properly oxidized to CO_2 (and in general, that incomplete combustion occurs, leading to a contemporary reduction in HC as well).

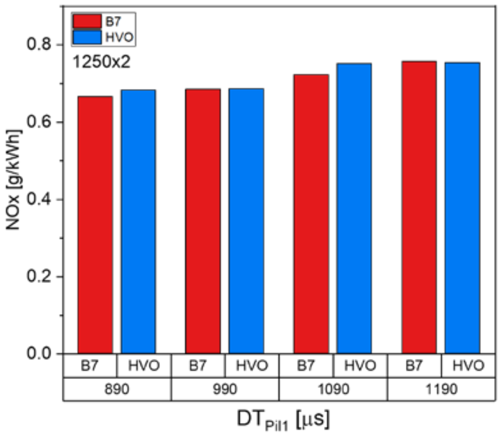


(**a**) (**b**)

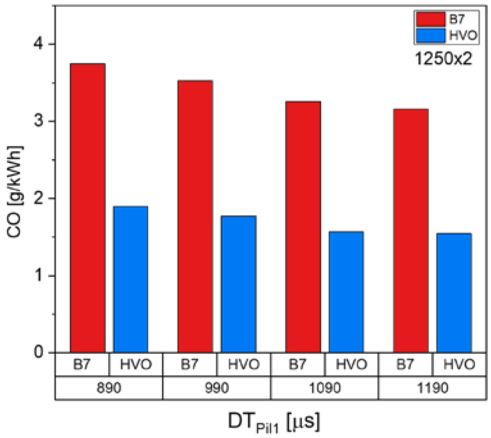
Figure 7. Influence of q_{Pil1} on CO (**a**) and NO_x (**b**) for diesel and HVO at 1250 × 2.

At 2250 × 15, the only significant patterns detected were a rather abrupt increase in CN when bigger q_{Pil1} values were adopted and a corresponding, albeit slight, increase in NO_x, indicating the need for relatively small q_{Pil1} at higher loads.

In order to control pollutant emissions, not only the quantity of pilot fuel, but also the dwell time between pilot and main injections needs to be accounted for. For example, the impact of DT_{Pil1} (i.e., the time interval between the end of pilot 1 and the start of the following main injection) on engine-out NO_x and CO emissions at 1250 × 2 is illustrated in Figure 8. By enlarging DT_{Pil1}, the pressure and temperature in the cylinder are reduced at the instant of pilot 1 injection (since SOI_{Main} is kept at a fixed value and a longer DT_{Pil1} means more advanced pilot 1), resulting in an increase in its ignition delay. This leads to a more diluted pilot 1 injection region and a decrease in the pressure and temperature rise generated by its combustion, diminishing the effect of pilot 1 injection on the ignition of the following main injection. As a consequence, a pilot injection further apart from the main shot (i.e., with the longest DT_{Pil1}) increases NO_x production, as it is evident from Figure 8.



(**a**)



(**b**)

Figure 8. Influence of DT_{Pil1} on NO_x (**a**) and CO (**b**) for diesel and HVO at 1250 × 2.

According to what has been said, due to its impact on in-cylinder combustion development, recalibration of pilot injection parameters (q_{Pil1} and DT_{Pil1}) at lower loads could either further reduce CO (and HC) emissions for HVO (if larger pilot quantities and/or shorter dwell times are implemented) or contribute to NO_x reduction.

4. Conclusions

In the current study, HVO (a paraffinic hydrogenated renewable fuel) was analyzed and compared to conventional petroleum-derived diesel in a Euro 6 compression ignition engine for light-duty commercial vehicles applications. The impact of both fuels on exhaust emissions and engine performance was investigated during steady-state operation, first without making any adjustments to the baseline (diesel-oriented) ECU calibration and then by investigating sweeps of EGR ratio, EGR split, rail pressure, main injection timing, pilot 1 injection quantity and dwell-time.

By running the engine on HVO as a "drop-in" fuel, without any ECU calibration parameter adjustments, significant reductions in soot (by up to 67%), HC and CO (by up to 40%) were highlighted at the selected engine operating points. However, there was no noticeable decrease in NO_x emissions; in fact, at low load, NO_x rose by nearly 10%.

By adjusting the ECU calibration along single-parameter sweeps, a preliminary evaluation of the possibility of achieving even greater improvements in engine performance and emissions with HVO (beyond those measured with the original diesel-oriented calibration) was carried out.

The reduction in engine-out soot, HC, and CO levels brought about by the intrinsic fuel characteristics of HVO could allow the engine to operate at higher EGR rates and, if the engine has a dual-loop EGR circuit, at a larger LP EGR proportion. This could result in a large decrease in NO_x emissions, particularly at lower loads, without incurring any penalties in incomplete combustion species.

At higher loads, if the primary objective is to reduce NO_x emissions, more delayed combustion phasing could be used when the engine is running on HVO, without incurring excessive penalty in *bsfc*. In contrast, more advanced combustion phasing can cut fuel consumption without exceeding soot levels of standard diesel. Furthermore, HVO's decreased sooting tendency could also be exploited to reduce combustion noise at higher loads, by reducing rail pressure.

Due to its impact on in-cylinder combustion development, recalibration of pilot injection parameters (injected quantity and dwell time) at lower loads could either further reduce CO and HC emissions from HVO (if larger pilot quantities and/or shorter dwell times are implemented) or further contribute to NO_x reduction.

Author Contributions: Conceptualization, S.d., A.M. (Alessandro Mancarella) and A.M. (Andrea Manelli); methodology, S.d., A.M. (Alessandro Mancarella) and A.M. (Andrea Manelli); validation, S.d., A.M. (Alessandro Mancarella) and A.M. (Andrea Manelli); investigation, A.M. (Alessandro Mancarella) and A.M. (Andrea Manelli); data curation, A.M. (Alessandro Mancarella) and A.M. (Andrea Manelli); writing—original draft preparation, A.M. (Alessandro Mancarella) and A.M. (Andrea Manelli); writing—review and editing, S.d., A.M. (Alessandro Mancarella) and A.M. (Andrea Manelli); supervision, S.d. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by ENI S.p.A. within the contract "Experimental and Numerical Characterization of Diesel Combustion with Standard and Innovative Fuels", number 2500037908.

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: Data is contained within the article.

Acknowledgments: The authors would like to acknowledge AVL for the utilization of CAMEO 4.4 and CONCERTO 5.6 within the University Partnership Program.

Conflicts of Interest: The authors declare no conflict of interest.

Abbreviations

°CAaTDC	Crank Angle degrees after TDC
°CAbTDC	Crank Angle degrees before TDC
ATS	After-Treatment System
bmep	brake mean effective pressure
bsfc	brake specific fuel consumption
CI	Compression Ignition
CN	Combustion Noise
CO	Carbon monoxide
CO_2	Carbon dioxide
DOC	Diesel Oxidation Catalyst
DPF	Diesel Particulate Filter
DT_{Pil1}	injection timing of the first pilot
DT_{Pil2}	injection timing of the second pilot
Δx	absolute change of the generic x variable
$\Delta\% x$	relative change of the generic x variable
ECU	Engine Control Unit
EGR	Exhaust Gas Recirculation
η_u	brake thermal efficiency
EU	European Union
FAME	Fatty Acid Methyl Esters
FPT	Fiat Powertrain Technologies
GHG	GreenHouse Gases
HC	unburned hydrocarbons
HP EGR	High Pressure EGR
HRR	Heat Release Rate
HVO	Hydrotreated Vegetable Oil
ICE	Internal Combustion Engine
ID	Ignition Delay
ID_{Main}	Ignition Delay of the main pulse
λ	relative air-to-fuel ratio
LP EGR	Low Pressure EGR
n	engine rotational speed
NO_x	Nitrogen Oxides
OEM	Original Equipment Manufacturer
PID	Proportional-Integrative-Derivative
PM	Particulate Matter
p_{rail}	rail pressure

q_{Pil1}	injected fuel mass quantity of the first pilot
q_{Pil2}	injected fuel mass quantity of the second pilot
SCR	Selective Catalytic Reduction
SOC	Start Of Combustion
SOI_{Main}	electric Start Of Injection of the main pulse
TDC	Top Dead Centre
vfc	volumetric fuel consumption
VGT	Variable Geometry Turbine
WLTC	Worldwide harmonized Light-duty Test Cycle

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