Surrogate fuel formulation to improve the dual-mode dual-fuel combustion operation at different operating conditions

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Abstract

Dual-mode dual-fuel combustion is a promising combustion concept to achieve the required emissions and CO_2 reductions imposed by the next standards. Nonetheless, the fuel formulation requirements are stricter than for the single-fuel combustion concepts as the combustion concept relies on the reactivity of two different fuels. This work investigates the effect of the low reactivity fuel sensitivity (S=RON-MON) and the octane number at different operating conditions representative of the different combustion regimes found during the dual-mode dual-fuel operation. For this purpose, experimental tests were performed using a PRF 95 with three different sensitivities (S0, S5 and S10) at operating conditions of 25% load/950 rpm, 50%/1800 rpm and 100%/2200 rpm. Moreover, air sweeps varying ±10% around a reference air mass were performed at 25%/1800 rpm and 50%/1800 rpm. Conventional diesel fuel was used as high reactivity fuel in all the cases. Moreover, commercial 95 RON gasoline was used as reference to compare the different TRFs. The engine settings were managed to adjust the rate of heat release to that found with 95 RON gasoline. To do this, a quality index imposing a maximum deviation of 5% point-to-point between the HRR curves from both fuels was defined. The results suggest that PRF 95 with S0 has the most similar behavior compared to conventional 95 RON gasoline whatever the engine load. As the engine load increases, the sensitivity effect is more noticeable and iso-HRR operation was only possible for S0. At low and medium load, the TRFs present similar engine-out emissions with equal fuel consumption. At full load, the NOx emissions are increased with respect to the reference 95 RON gasoline without fuel consumption benefits. The results from the air variation for the different octane numbers demonstrated that the greatest differences are obtained for low air mass (i.e, higher EGR). In addition, the decrease of the octane number limits the maximum air increase due to the pressure gradients, requiring modifications in the engine settings that increase the soot formation.

Introduction

According to the last reports, internal combustion engine propelled vehicles stand for the 90% of the total passenger cars [1]. A similar analysis on the distribution and transportation sector for medium and heavy-duty vehicles also indicates the dominance of the internal combustion engine (ICE) application [2]. Moreover, future predictions that take into account the different powertrains and energy sources that could be used in the future point out that the previous scenarios will be maintained for a long timeframe [3]. Nonetheless, the emission limits must be consistently decreased during the years to reduce the environmental impact of this technology. One of the main changes is the introduction of a legislation to limit the CO_2 emissions, targeting a reduction of 15% for 2025 [4]. Different from the other emissions that can be reduced using external devices as diesel oxidation catalyst (unburned hydrocarbons (HC) and carbon monoxide (CO)), diesel particulate filter (soot) and selective catalytic reduction (NOx) [5][6], the reduction mechanism of the CO_2 emissions is quite complex since it involves all the lifecycle of a fuel. Even in the case of the emissions that are already legislated, the associated cost to the aftertreatment system represents an important part of the final vehicle price. Moreover, the use of these devices imply an increase of the operational costs [7], as the urea consumption to reduce the NOx emissions, the additional fuel consumption during the DPF active regeneration and the increase of pumping losses due to the pressure losses in the monolith channels [8].

In this sense, efforts are being made to develop new combustion strategies able to reduce the main engine-out emissions while increasing the engine thermal efficiency, thus reducing the tailpipe CO_2 emissions. The low temperature combustion (LTC) techniques have been pointed as one of the most attractive combustion strategies as they offer a simultaneous decrease of the soot and NOx emissions with thermal efficiencies higher than those provided by the current diesel engines [9]. The LTC concepts are generally based on using a high degree of fuel premixing by means of early injections in combination with higher exhaust gas recirculation (EGR) dilution levels [10][11]. The combustion process is faster than in conventional diesel combustion (CDC), with low heat losses to the walls and a greater degree of volumetric combustion, which reduces the efficiency losses due to the volume expansion [12]. Reactivity controlled compression ignition (RCCI) is one of the most promising combustion concepts inside the LTC field [13][14]. The use of two fuels with contrasting reactivity injected and different injection systems allows to tailor the in-cylinder mixture reactivity on demand, which provides an extra degree of control over the combustion development. Nonetheless, the RCCI operation is limited due to excessive CO and HC emissions at low load and high-pressure gradients as the load is increased [15][16]. To extend the benefits of the RCCI combustion to greater regions of the engine map, different approaches are studied in the literature. Among them, the dual-mode dual-fuel combustion (DMDF) was found to be able to reduce the pressure gradients by switching from a fully premixed combustion to a dual-fuel diffusive as the engine load is increased [17]. This action penalizes the NOx and soot levels compared to RCCI, but the emissions levels are inferior than those presented by CDC [18].

The fuel properties play a dominant role on the DMDF combustion performance since this concept still relies on a great extend to the kinects to control the combustion process. The resultant mixture properties dictate the switching line from a fully premixed combustion to a dual-fuel diffusive one. Properties as the research octane number (RON) and motor octane number (MON) are generally used as a scale to represent the fuel reactivity. Investigations addressing the effect of the octane number have been performed for LTC concepts as the homogeneous charge compression ignition (HCCI). Generally, it is reported that higher octane fuels allow to extend the maximum upper limit of the HCCI combustion. By contrast, the use of low octane number fuels is beneficial to improve the fuel oxidation at low load conditions, which reduces the HC and CO concentration [19]. Similar investigations were performed in RCCI combustion by using a mixture of ethanol and gasoline (E85), i.e., increasing the octane number by means of ethanol addition. The addition of ethanol to the blend allowed to extend the higher operating limits to higher loads. Nonetheless, low load conditions were impaired increasing the HC and CO emissions [20][21]. Recent research demonstrates that the octane number variation is a key parameter to achieve efficiency improvements in DMDF combustion [22]. The fuel sensitivity (S), defined as RON-MON, serves as reference to characterize a fuel with respect to the different behaviors that can present at the same operating condition when the RON is maintained [23][24]. This parameter was deeply investigated in conventional spark ignited applications and its origins relies on the low temperature dissociation reactions of the paraffinic compounds that provokes an initial temperature increase enhancing the reactivity of the blend [25][26]. Since the advanced combustion concepts rely on using low reactivity fuels as gasoline, the fuel sensitivity should be also a dominant parameter on the combustion development. Previous studies demonstrated that the sensitivity effect on the DMDF combustion is deeply affected by the premixing degree as well as the load condition [27]. However, the literature addressing this subjected is still scarce. Moreover, the effect of the fuel parameters on advanced concepts as DMDF combustion are still not clear. In this sense, the aim of this research is to evaluate the effect of the fuel properties in the DMDF concept by means of experimental tests. Three different fuel sensitivities (S0, S5 and S10) were evaluated at 25%/950 rpm, 50%/1800 rpm and 100%/2200 rpm. In addition, three research octane number (100, 90 and 80) were assessed at 25%/1800 rpm and 50%/1800 rpm by sweeping the air mass in +/- 10%. Therefore, the impact of these properties in the combustion progression of the different combustion regimes can be identified. The evaluation was carried out in a commercial multi-cylinder medium-duty compression ignition engine platform modified to run on DMDF combustion.

Materials and methods

Engine characteristics

The experimental tests were performed in a six-cylinder 8L platform currently used in a commercial truck. Table 1 specifies the main characteristics of the engine. It should be highlighted that the engine was modified to enable the operation in dual-mode dual-fuel combustion. Among the different modifications, the reduction of the original compression ratio and the optimization of the piston geometry were the main responsible for the successful implementation of the DMDF combustion over all the engine map. The optimization of these parameters was done by means of computational simulations and experimental approaches in a previous work [28]. Moreover, auxiliary devices were added to enable the DMDF operation. First, individual port fuel injectors for each cylinder were installed at the intake manifold. In addition, a low pressure EGR system was added. This approach allows to regulate the exhaust flow in a manner that the energy in the turbine is enough to provide the required boost pressure for each operating condition. Moreover, the cooling of the low pressure EGR iallows to maintain the temperature levels near to those from ambient conditions.

Table 1. Engine characteristics.

Engine Type	4 stroke, 4 valves, direct
Number of cylinders [-]	6
Displaced volume [cm^3]	7700
Stroke [mm]	135
Bore [mm]	110
Piston bowl geometry [-]	Bathtub
Compression ratio [-]	12.75:1
Rated power [kW]	260 @ 2100 rpm
Rated torque [Nm]	1400 @ 1050-1600 rpm

Test cell description

The previously described engine was installed in a test cell facility presented in Figure 1, which is equipped with all the required devices to control the engine and obtain information about the combustion, performance and emission parameters. The engine control was based on a coupled actuation of the original electronic control unit (ECU) together with an additional Labview routine and a NI PXIe 1071 board. The ECU is responsible to control the variable geometry turbine (VGT) position, high pressure (HP) EGR and injection pressure. Labview routine controls the injection timing and duration of both low reactivity fuel (LRF) and high reactivity fuel (HRF) as well as the amount of low pressure (LP) EGR.

The engine load and speed were managed by means of an AVL active dynamometer. An AVL PUMA Open interface was used to manage the difference devices of the test cell such as the gas analyzer, fuel measurement system, smoke meter, etc. Moreover, this interface was connected to different acquisition boards allowing to obtain average values of pressure, temperature and mass flows at different locations of the setup. Each fuel (LRF and HRF) was measured independently by means of two AVL 733 S balances while air mass flow was obtained by an Elster RVG G100 sensor. K-type thermocouples were used at the different locations of interest to obtain the values of temperature and Kistler / 4045A pressure transducer were used to assess the pressure magnitude at these locations. High frequency signals were acquired in a different routine built in Labview inside the main code employed to control de engine. The in-cylinder pressure signals of each one of the six cylinders were measured using Kistler 6125C pressure sensors using an AVL 364 encoder with a resolution of 0.2 Crank angle degree (CAD). The resultant signal was acquired by the NI PXIe 1071 board and used to perform an online heat release analysis. This allows to obtain online information about the combustion process. In parallel, these signals were stored in packages of 200 cycles to be used as boundary conditions in the post-processing code. A five-gas Horiba MEXA-7100 DEGR analyzer was used to measure the engine-out gaseous emissions. This device contains an additional module that allows parallel measurement of CO_2 concentration at both intake and exhaust lines to calculate the EGR rate. Smoke emissions were measured by means of an AVL 415S smoke meter in filter smoke number (FSN) units. For each operating condition, three consecutive measurements of 1-liter volume each with paper-saving mode off were collected [29]. The accuracy of the main elements of the test cell is presented in Table 2.

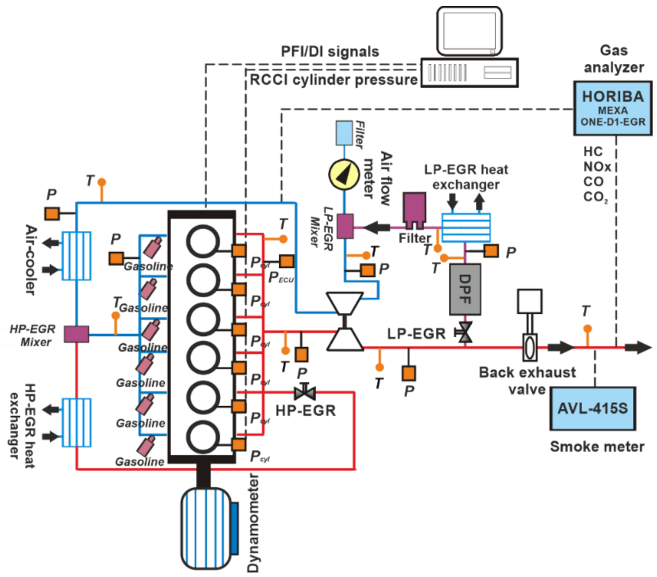


Figure 1. Experimental facility scheme.

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

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

Fuels and injection systems characteristics

In this work, commercial fuels as well as reference fuels were used. The HRF fuel was the same during all the study while the LRF was changed from commercial gasoline to primary reference fuels (PRF) as well as Toluene reference fuels (TRF). The PRFs are a blend of n-heptane and isooctane targeting a specific octane number while TRFs are ternary blends of isooctane, n-heptane and toluene. The main difference between then relies on the values of RON and MON. As previously discussed, the subtraction of the RON and MON is called fuel sensitivity. While the binary blends of isooctane and n-heptane presents S=0, the addition of toluene as a ternary component allows to increase the fuel sensitivity to a desired value. This is generally made employing mixing rules from multi component diagrams. In this work, the procedure proposed by [30] was used to obtain the desired sensitivities and octane numbers. The main characteristics of the fuels used in this research are shown in Table 3.

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

	EN 590 diesel	EN 228 gasoline	n-heptane	isooctane	Toluene
Density [kg/m^3] (T= 15 °C)	842	720	645	658	866
Viscosity [mm^2/s] (T= 40 °C)	2.929	0.545			
RON [-]	-	95.6	0	100	121
MON [-]	-	85.7	0	100	107
Cetane number [-]	51	-	-	-	
Lower heating value [MJ/kg]	42.50	42.4	44.57	44.43	40.59

Table 4 depicts the composition of the ternary fuel blends that were used to realize sensitive fuels. It should be remarked that the composition of the mixtures of binary components for RON variation are omitted, for brevity sakes, once the determination of isooctane and n-heptane volumetric fraction are straightforward.

Table 4. Volumetric content of the ternary blends that were used to realize RON 95 with different sensitivities.

	TRF 95		
Sensitivity	S=0	S=5	S=9
v/v Toluene	0.00	0.245	0.6
v/v Isooctane	0.95	0.637	0.212
v/v N heptane	0.05	0.117	0.188

The LRF and HRF fuels were provided to the injection systems by means of separated fuel lines. The main characteristics of each injection system are summarized in Table 5. Commercial diesel was injected directly into the cylinder using the stock common-rail fuel direct injection (DI) system, with a centrally located seven-holes solenoid injector. The LRF was injected at pressures around 5 bar using a low-pressure pump and six port fuel injectors (PFI) injectors located at the intake manifold.

Table 5. 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]	6.0

Testing methodology

This section presents the methodology followed to assess the sensitivity and the octane number influence on the combustion properties, performance and emissions.

Sensitivity evaluation

The sensitivity effect on the combustion process was evaluated by comparing the heat release rate (HRR) obtained for the given fuel with respect to a reference case operating with diesel-gasoline. Thus, for each operating condition, the engine settings were modified to obtain iso-HRR operation. To describe the similarity between the studied case and the reference, an iso-HRR quality index was defined. This parameter is defined as the sum of differences for each crank angle between the diesel-gasoline reference and the operating condition that is being measured, and normalized by the reference curve multiplied by 1.05. This means that the maximum deviation accepted to assume that the HRR are similar is 5% of the reference. The iso-HRR quality index lumps both the impacts of phasing and magnitude disparities in the combustion process, aiming to be conservative and assure that only HRR profiles with similar phasing and heat release profile will be considered iso heat releases. Figure 2 graphically illustrates the definition of the concept. This graph is built considering a variation of 1.05 for the reference HRR (black shaded graph) and the difference of a random measured condition minus the reference HRR (blue shaded graph). As it can be seen, the area of the blue plot is lower than the black one, indicating that the differences between the random operating condition are lesser than the one that should be found if one considers a 1.95 difference in the reference HRR. Therefore, the random HRR can be considered similar to the reference. The hypothesis of similarity in term of HRR is denied when the Area 2 is greater than the Area 1.

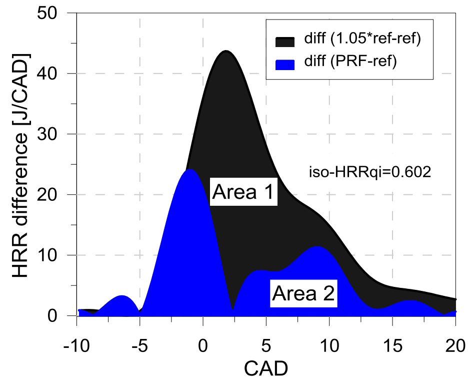


Figure 2. Graphical representation of the iso-HRR quality index.

The Equation 1 presents the mathematical definition of this factor considering the aforementioned:

$$iso - HRR_{qi} = \frac{\sum_{i=Startcomb}^{Endcomb} |HRR_{PRF}(i) - HRR_{ref}(i)|}{\sum_{i=Startcomb}^{Endcomb} |1.1 * HRR_{ref}(i) - HRR_{ref}(i)|} \quad (1)$$

If $iso - HRR_{qi} \leq 1$, good agreement between both HRR, consequently it can be stated that they are similar.

If $iso - HRR_{qi} > 1$, poor agreement between both HRR, consequently it can be stated that they are not similar.

Three different sensitivities (0, 5 and 10) were evaluated at a fixed RON value of 95. Each fuel blend was assessed in three operating conditions addressing the different combustion regimes of the DMDF concept. First, a low engine speed (950 rpm) and load (25%) was evaluated in a fully premixed condition with gasoline fractions (GF) lower than 50%. Second, a medium load (50%) medium speed (1800 rpm) condition with high GF values (>80%) was analyzed to assess the impact of the high levels of LRF on the combustion. Finally, the sensitivity variation impact was investigated at a full load condition and maximum engine speed (2200 rpm). This condition is characterized by low GF values, running in a dual-fuel diffusive combustion.

Octane number influence

Previous studies demonstrated that the octane number has a fundamental role on the reactivity of the in-cylinder mixture. Moreover, this parameter is generally related to the pressure gradients found at the combustion chamber which limits the extension towards the efficiency increase. This can be also observed in the DMDF concept, which relies on the different fuel reactivity to obtain the benefits of a low temperature combustion. Therefore, the effect of the octane number on the DMDF combustion concept was assessed by means of varying the octane number and the air sweep for different engine loads at a fixed engine speed. The air mass variations were achieved by modifying the EGR percentage to reach variations of +/-10 % of the air mass whenever possible. It should be remarked that the an iso-load comparison was aimed in each operation condition. This means that low variations of the fuel mass could be done to compensate any load variation. Nonetheless, these small modifications are far inferior that the variations of the air mass, which indicates that the equivalence ratio is not maintained during the air sweep. For each operating condition, the maximum air mass could be limited by the pressure gradients whilst the minimum air mass (and consequently higher EGR fraction) could be limited by engine-out CO concentrations higher than 3000 ppm. Three research octane numbers (100, 90 and 80) with sensitivity zero were evaluated a 25% and 50% of engine load at a fixed engine speed of 1800 rpm. Higher engine loads than 50% do not allow to modify the air mass in a flexible way due to the mechanical constraints associated to excessive pressure gradients. Generally, these operating conditions require the modification of two or more parameters to deal with the air mass, resulting in different boundary conditions for the comparison. The fixed engine speed was chosen to isolate the effect of increasing the flow turbulence intensity on the combustion process that can overshadow the octane number effect. The initial settings or each operating condition are a result of a calibration map obtained following the procedure described in [31], that aims to obtain the best values in terms of fuel consumption, soot and NOx emissions simultaneously.

Results and discussion

Sensitivity evaluation

First, an evaluation of the PRF 95 combustion compared to the commercial gasoline was done. For this, the three operating conditions were evaluated at iso-settings conditions. Figure 3 presents the different HRR for PRF 95 and commercial gasoline for each one of the operating conditions evaluated.

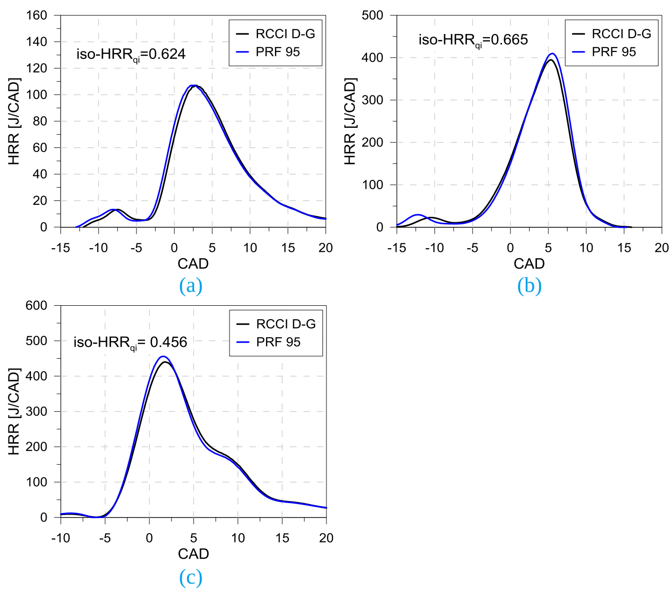


Figure 3. Heat release rates comparison between PRF 95, S=0 and commercial gasoline as low reactivity fuels at (a)950 rpm and 25%, (b)1800 rpm and 50% and (c)100% and 2200 rpm.

As it can be seen, the direct replacement of gasoline by PRF 95 resulted in similar combustion process, following both the trend and absolute values of heat released in each crank angle. This means that the dominant gasoline characteristics of the combustion process are maintained for PRF 95. Besides, equivalent fuel consumption is similar for both fuels as shown in Table 6. The use of an equivalent fuel consumption is required since the primary reference fuels present different heating value than gasoline as presented in Table 1.This equivalent fuel consumption was extended to the whole investigation once the LHV of the binary and ternary blends can differs from those of gasoline. This parameter is defined in Equation 2:

$$BSFC_{eq} = \frac{m_D + m_{PRF} \cdot \left(\frac{LHV_{PRF}}{LHV_{Gasoline}}\right)}{N_b} \quad (2)$$

Some deviations can be seen in the main engine-out emissions. The higher differences are noticed in the NOx emissions. Nonetheless, it should be noted that the concentration values of this component at the exhaust gases is lower than 50 ppm. Therefore, slight differences in the combustion process can affect the NOx formation. In this sense, even lower ppm differences can lead to noticeable differences in the specific emission. By contrast, emissions as CO and HC have lower sensibility to the concentration changes, presenting a maximum difference of 17%. Finally, soot emissions present similar values indicating that the formation mechanism of this component is maintained in the PRF combustion. Therefore, from now on the PRF 95 will be used as reference from the comparison with the additional sensitivities.

Table 6. Comparison between D-G and D-PRF 95 performance and emissions for the different operating conditions.

	25%@950 rpm		50%@1800 rpm		100%@2200 rpm	
[g/kWh]	D/G	PRF 95	D/G	PRF 95	D/G	PRF 95
$BSFC_{eq}$	227.6	225.14	209.6	206.14	216.8	214.85
NOx	0.37	0.28	0.35	0.24	2.22	2.02
HC	5.74	6.67	2.07	2.12	0.58	0.51
CO	16.77	15.3	2.57	2.11	3.76	3.91
Soot	0	0	0.00045	0.00042	0.147	0.156

Low load, low engine speed: 25% load and 950 rpm

First, the results of low load and low engine speed are presented. As it can be seen in Figure 4 low load condition allows to tailor the boundary condition to obtain similar HRR for the different sensitivities (iso-HRR quality indexes values around 1). The influence of the sensitivity on the combustion process can be also intuited by analyzing the engine settings modifications needed to obtain the iso-HRR, as shown in Table 7. From Table 6, it can be inferred that the major modification relies on the start of injection (SOI) of the main injection. It should be noted that the dwell time between the pilot and main injections are maintained constant for each operating condition. This means that the early SOIs for the main injection means an early SOIs also for the pilot. The EGR rate, GF and fuel mass were maintained nearly at the same level for all the operating conditions, indicating that the sensitivity modification has a low impact at low load conditions. As previously discussed, sensitive fuels are known to present a RON de-rated behavior at NTC conditions. Moreover, it can be concluded from the SoI modifications, that the pressure-temperature trajectory of this operating condition is placed on the NTC affected zone. This means that the reactivity in the cylinder should be tailored to avoid an early ignition than the reference case. This was accomplished by reducing the compositional stratification by means of increasing the mixing time of the blend avoiding richer zones (more reactive) in the combustion chamber.

Table 7. Boundary conditions for the different fuel blends for low load, low engine speed condition.

Inlet conditions	PRF 95-S0	TRF 95-S5	TRF 95-S10
Air mass [g/s]	37.81	37.85	36.47
Temperature [°C]	45.24	44.33	46.77
EGR [%]	42.48	42.65	43.52
Fuel mass [mg/cc]	32.7	33.78	33.41
GF [%]	49	50.9	49.9
SOI_{main} [CAD bTDC]	19	23	26

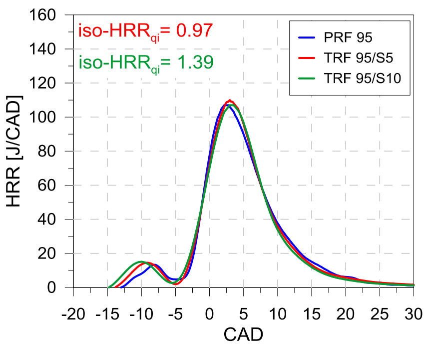


Figure 4. Heat release rate profiles for the different sensitivities evaluated for low load and low engine speed conditions.

Table 8 presents the impact of the sensitivity modification on the fuel consumption results as well as on the main engine-out emissions. It can be inferred that the major emissions are not impacted by the sensitivity modification as the values of soot, HC and CO are similar for each fuel blend. The NOx present a slightly decrease for the S=10 as lower HRR peak is obtained for this fuel. As the sensitivity is increased, it can be seen a proportional increase in the fuel consumption values. It should be remarked that the brake fuel consumption considers variations in the pumping losses that can be different for each fuel. As the HRR profiles and emissions are similar, it can be concluded that the equivalent brake specific fuel consumption (BSFC) variations are not related to the sensitivity modification of the fuel.

Table 8. Specific fuel consumption and engine out emissions for the fuel blends investigated at low load, low engine speed conditions.

	BSFC [g/kWh]	NOx [g/kWh]	CO [g/kWh]	HC [g/kWh]	Soot [g/kWh]
PRF 95 S0	225.14	0.28	15.3	6.67	0
TRF 95-S5	229.65	0.277	15.578	6.892	0
TRF 95-S10	233.45	0.208	15.542	6.585	0

Medium load, medium engine speed: 50% load and 1800 rpm

Figure 5 presents the heat release profiles for each one of the sensitivities evaluated at the operating condition of 50% engine load and 1800 rpm. It is interesting to note that the iso-HRR quality index increases compared to the previous condition. From the Table 8, it can be inferred that the modifications for S=5 compared to S=0 are minimal. The most significant change is performed for S=10, where the EGR concentrations are considerably increased. This indicates that the sensitivity increases results in a higher mixture reactivity. This behavior is presented in the literature only for narrow zones, where the operating condition is in the transition zone from a non-negative temperature coefficient (NTC) affected zone to an NTC affected one. Therefore, this could explain the results observed for the combustion process. It should be also stated that this operating condition is located near to the transition zone from the RCCI to the dual-fuel diffusive combustion. This present a hurdle in which regards the settings modification to realize the similar HR rate. Even small modifications on the early heat release rates (low temperature heat release) can significantly modify the combustion process, once the increment of temperature and pressure from this heat release phase can leads to pressure gradient limited operation. In this sense, the SoI modifications were constrained at this condition once the low temperature heat release is highly affected by the compositional and temperature stratification. The reactivity modification was then pursued by modifying the EGR concentration, once it is one of the most effective paths to reduce the reaction rates as it can be seen in Table 9.

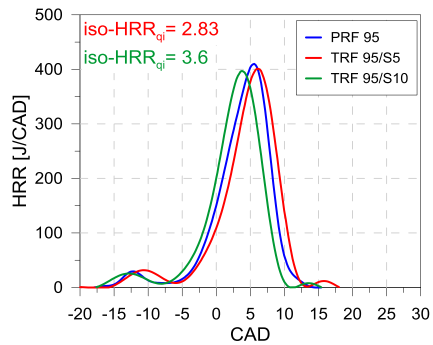


Figure 5. Heat release rate profiles for the different sensitivities evaluated for medium load and medium engine speed conditions.

Table 9. Boundary conditions for the different fuel blends for low load, low engine speed condition.

Inlet conditions	PRF 95-S0	TRF 95-S5	TRF 95-S10
Air mass [g/s]	140.9	144.1	129.63
Temperature [°C]	52.38	44.92	60.11
EGR [%]	40.35	40.13	48.2
Fuel mass [mg/cc]	80.2	80.43	80.6
GF [%]	83.38	84.3	84.3
SOImain [CAD bTDC]	49	49	49

The average results shown in Table 10 demonstrate that the fuel sensitivity has low impact on the main emissions. As it can be seen, the NOx values are similar for all the fuel blends, with values under 0.3 g/kWh. This results in an exhaust concentration lower than 60 ppm, which is achieved through a fast combustion process and high premixing degree. Soot values are virtually zero in all the cases, with one order of magnitude lower than the current EUVI normative. The HC and CO emissions also present a low sensitivity for the fuel blend variation. The most noticeable change is observed in the fuel consumption, which increases according to the sensitivity increment. This is mainly attributed to the early combustion process, which reduces the fuel-to-work conversion efficiency due to the increase of the energy released during the compression stroke.

Table 10. Boundary conditions for the different fuel blends for medium load, medium engine speed condition.

	BSFC [g/kWh]	NOx [g/kWh]	CO [g/kWh]	HC [g/kWh]	Soot [g/kWh]
PRF 95 S0	206.14	0.24	2.18	2.12	0.00045
TRF 95-S5	207.46	0.22	2.57	2.18	0.00091
TRF 95-S10	213.96	0.25	2.72	2.9	0.0017

Full load, maximum engine speed: 100% load and 2200 rpm

At full load operation, the sensitivity has a dominant role on the combustion development. Figure 6 presents the most similar combustion process that could be achieved after numerous combinations off settings. As it can be seen, S=5 still can be tailored to have a similar combustion process. Nonetheless, S=10 becomes a challenge to match the combustion development. The use of the same boundary conditions than those of the previous sensitivities provided a delayed combustion. This decrease of reactivity as the sensitivity is increased is generally verified at beyond RON conditions, that can be realized by high pressure and temperatures at the inlet valve close. The NTC region is avoided at these cases and the low temperature heat release starts to become less significant compared to the total energy released on the process. These conditions are fulfilled in the operating condition in discussion as it can be seen in Figure 6 and Table 11.

It was attempted to compensate the decrease of reactivity by modifying the injection timings without success. Early diesel injections and higher temperatures without success. Moreover, as the combustion progresses, there is a clear split of the premixed phase and diffusion phase, marked by the inflexion on the HRR profile. Therefore, the GF values were increased intending to fit the HRR by scaling the premixed phase. Nonetheless, at this condition the increase of the premixed energy is restricted by the pressure gradients. Finally, it was not possible to find a set of settings able to reproduce the reference HRR of PRF 95.

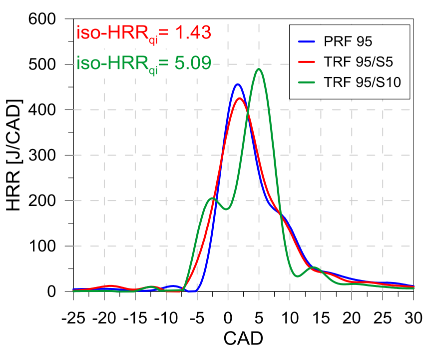


Figure 6. Heat release rate profiles for the different sensitivities evaluated for full load and maximum engine speed conditions.

Table 11. Boundary conditions for the different fuel blends for full load, maximum engine speed condition.

Inlet conditions	PRF 95-S0	TRF 95-S5	TRF 95-S10
Air mass [g/s]	263.13	274.85	260.89
Temperature [°C]	80.6	82.16	87.14
EGR [%]	18.86	19.43	19.53
Fuel mass [mg/cc]	129.66	130.78	127.41
GF [%]	38.4	38.8	49.5
SOImain [CAD bTDC]	12	14	15

Table 12 shows that, despite of the different HRR between the different sensitivities, the specific fuel consumption achieved are similar to the reference case. By contrast, the main engine-out emissions differ for each operating condition because of the settings used. The two first fuel blends (PRF 95 and TRF 95 S5) present more similarity as the gasoline fractions are similar. The NOx emissions increase with the sensitivity due to the higher combustion duration (Figure 6). On the other hand, CO emissions are reduced whilst HC are slightly increased maintaining an equilibrium. The main difference between the three cases can be observed in the soot emissions, which are more than 3 times lower than the reference case. This can be explained by the early injection timings used for S=5 that allows to improve the fuel mixing, reducing the number of rich zones that are one of the main soot formation sites. Finally, the settings used for S=10 are likely to decrease the soot emissions and increase the HC emissions as it can be seen in Table 11. Once higher GF is used, the premixed fraction is higher. This means an increase in the fuel that can enters the cylinder crevices and impinges in the walls. These mechanism favors the HC formation. By contrast, the reduction of the direct injected fuel allows the probability of having rich zones. In addition, the early fuel premixing from the LRF decreases the equivalence variations inside the combustion chamber. The reduction of these rich zones leads to a noticeable decrease of the soot formation.

Table 12. Specific fuel consumption and engine out emissions for the fuel blends investigated at full load, maximum engine speed conditions.

	BSFC [g/kWh]	NOx [g/kWh]	CO [g/kWh]	HC [g/kWh]	Soot [g/kWh]
PRF 95 S0	214.85	2.02	3.91	0.38	0.156
TRF 95-S5	213.88	2.589	2.6	0.526	0.047
TRF 95-S10	216.78	3.884	2.191	1.111	0.022

Summary and discussion

The detailed analysis of the sensitivity impact on the combustion development for different operating conditions of the DMDF concept allows to draw important conclusions. The introduction of aromatic compounds on the surrogate fuels have affected the combustion development, where the most sensitive fuels have prevented the operation at iso heat release combustion. This seems to be also dependent on the engine load in a low extent. Low engine load provides paths to realize similar combustion process even in the cases with higher sensitivities. By contrast, as the engine load is increased the shift of the pressure-temperature trajectories to beyond RON operation becomes a challenge to obtain similar combustion development.

Figure 7 shows the different iso-HRR quality indexes for the tested operating conditions in an iso-contour map as function of the engine load and the sensitivity. This highlights the importance of the fuel composition on the combustion development in a concept that relies on the fuel reactivity to control the different combustion phases. Finally, the variations observed in the results with respect to the sensitivity evidence that the fuel formulation must studied in detail to maximize the benefits of the advanced combustion concepts.

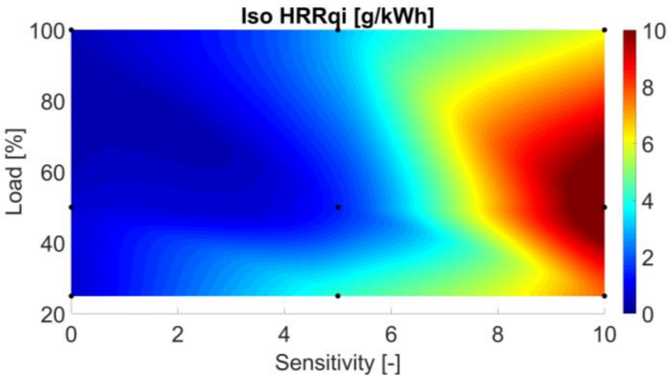
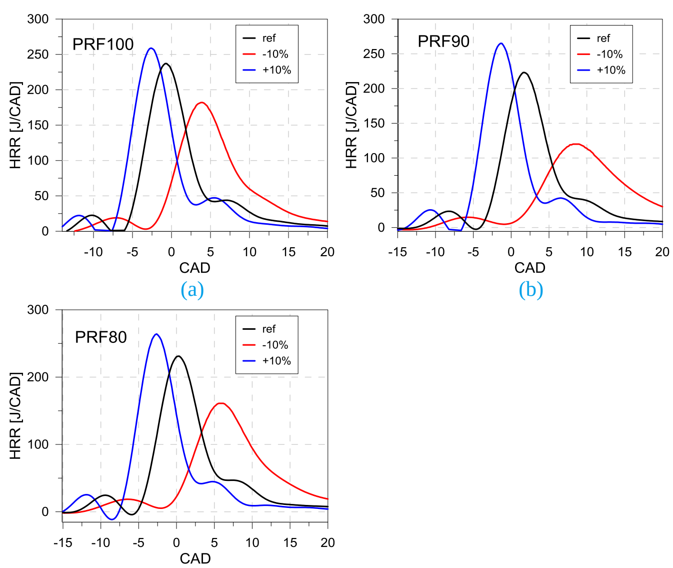


Figure 7. Iso-contour graph depicting the iso-HRR quality index variation according to the sensitivity and engine load.

Octane number influence

25% engine load

Figure 8 (a) to Figure 8 (c) present the HRR profiles for the different octane numbers and air variations evaluated. The shape and absolute values of the HRR are maintained for all the PRFs, mainly for the reference case and when the air mass is increased (↓EGR). Nonetheless, when the air mass is reduced compared to the reference case, higher differences can be noticed in the HRR, mainly for the PRF 90. This fuel blend presents the highest decrease of the maximum HRR value and a considerable increase in the HRR duration. This behavior should be related to the boundary conditions used for each one of the operating points. In this sense, the different engine settings and air management parameters are presented in Table 13.



(a) (b)

(c)

Figure 8. Heat release rate profiles illustrating the different air sweeps for (a) PRF 100, (b) PRF 90 and (c) PRF 80 for 25% of engine load at 1800 rpm.

As it can be seen, the air mass reduction did not achieve the target of 10% as the CO emissions surpassed 3000 ppm (imposed limit). For PRF 100, a higher inlet temperature was used to allow sweeping the air mass in a wider range (the same inlet temperature limited the increase of EGR to small percentages). Then, even in the case of an air mass reduction of 9.3%, the CO emissions were under the constraint. Nonetheless, as PRF 90 and PRF 80 presented higher reactivity, the temperature was set to a similar and lower value enabling the direct comparison of them. . Table 13 and Table 14, depict the operating settings for the operating conditions evaluated. It is possible to see that the EGR increase was limited for PRF 90 allowing air mass reductions of only 8.6% due to excessive CO emissions. The increase of the octane number allowed to achieve the desired reduction of 10% of air mass for the PRF 80 as a consequence of the higher reactivity of this fuel . This is also demonstrated in the bar charts of Figure 9, where this parameter is depicted together with the combustion phasing, pressure gradient and fuel consumption results.

Table 13. Engine settings and boundary conditions used for the different fuel blends at 25% of engine load.

Parameter→	P_int	T_int	Air mass	air mass change	EGR
Operating condition↓	[bar]	[ºC]	[g/s]	[%]	[%]
PRF 100 dw	1.5	56.0	83.1	-9.3	50.6
PRF 100 ref	1.5	55.8	91.7	0.0	44.6
PRF 100 up	1.5	56.3	100.9	10.0	39.5
PRF 90 dw	1.4	51.2	79.6	-8.6	48.4
PRF 90 ref	1.4	50.0	87.1	0.0	41.5
PRF 90 up	1.4	49.5	95.7	9.9	34.9
PRF 80 dw	1.4	51.6	78.3	-9.6	48.3
PRF 80 ref	1.4	51.9	86.7	0.0	41.6
PRF 80 up	1.4	52.1	95.1	9.7	34.8

Table 14. Engine settings and boundary conditions used for the different fuel blends at 25% of engine load.

Parameter→	SOIpilot	SOImain	PMI	GF
Operating condition↓	[CAD bTDC]	[CAD bTDC]	[bar]	[-]
PRF 100 dw	32.0	22.0	7.0	42.1
PRF 100 ref	32.0	22.0	7.0	41.9
PRF 100 up	32.0	22.0	7.1	42.4
PRF 90 dw	32.0	22.0	6.9	43.7
PRF 90 ref	32.0	22.0	6.9	41.7
PRF 90 up	32.0	22.0	7.0	42.0
PRF 80 dw	32.0	22.0	7.0	45.2
PRF 80 ref	32.0	22.0	7.0	44.7
PRF 80 up	32.0	22.0	7.0	44.1

As it can be seen in Figure 9 (b), the combustion duration does not present variations with respect to the RON for the reference and higher air mass cases. Only the reduction of air mass enlarges the combustion process in a different manner for each RON. The PRF 90 presented the highest combustion duration because of the lower intake temperature than that from PRF 100 and lower reactivity compared to PRF 80 even with lower air mass decrease. Moreover, the increase of the EGR concentration displaced the combustion phasing towards the expansion stroke. The combination of both modifications results in a higher brake specific fuel consumption for PRF 90 than for PRF 80 at this sweep condition. Considering both the reference and higher air mass cases, there is a slightly fuel consumption improvement towards the RON decrease as a consequence of improvements of combustion efficiency, as demonstrated by the CO and HC results in the following discussions. At this operating condition, the pressure gradient results are not critical and the variations in the octane number does not seem to impact this property.

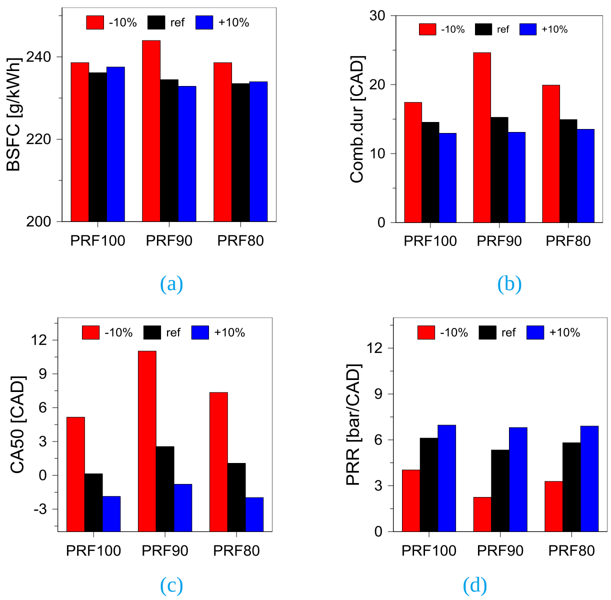
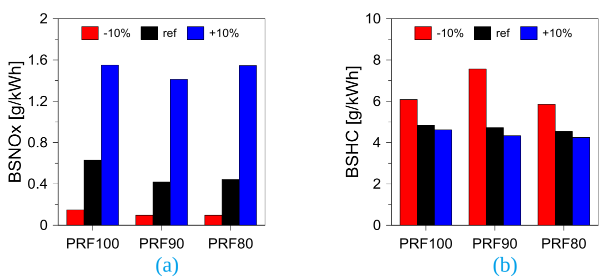


Figure 9. Brake specific fuel consumption (a), combustion duration (b), combustion phasing (c) and pressure rise rate (d) for the different octane number and air sweeps evaluated at 25% of engine load.

Despite of the different combustion results, the emissions showed a low sensibility to the octane number modification. As depicted in Figure 10 (a), NOx emissions are mainly dominated by the air mass modification whilst the variations of the octane number provided almost the same absolute results comparing similar cases. Even in the case where the inlet conditions are maintained (air mass and temperature) for PRF 90 and PRF 80, the differences of the NOx values are minimal. This can be also extended to other emissions as the HC and CO, presented in Figure 10 (c) and Figure 10 (d), respectively.



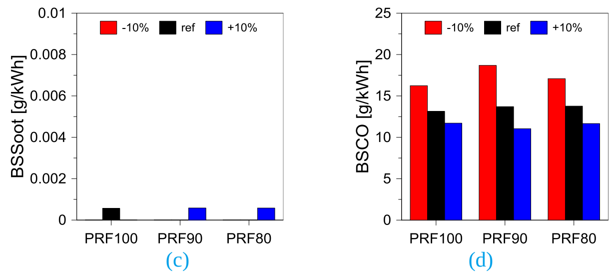


Figure 10. Brake specific emissions for the engine load of 25% (a) NOx, (b) Soot, (c) SHC, (d) CO for the different fuel blends and air sweeps evaluated.

Differences can be observed only when the air mass is decreased towards the limiting condition in which regards combustion stability. At these conditions, it can be noticed an increase in both HC and CO emissions for PRF90 compared to PRF 80 (fuels with similar air mass, pressure and temperature). Soot emissions are negligible independently on the fuel blend evaluated and the operating condition.

50% engine load

The operating condition of 50% engine load is characterized by higher GF levels and early injection settings. In these conditions, the LRF should play a dominant role on the combustion process. This can be easily verified in the results presented in Figure 11. Again, the different heat release profiles are depicted for each one of the octane number evaluated as well as the air sweeps. PRF 100 presents a narrow HRR rate with a higher peak than PRF 90 and 80. This is mainly based on the higher GF values for this fuel blend (Table 16). As the starting calibration point comes from the previous calibration maps, it was intended to maintain them for each fuel blend. Nonetheless, as the RON was decreased, the pressure gradients surpass the mechanical constraints. Therefore, the premixing level must be reduced to guarantee the engine safety. The same GF level was maintained for both PRF 90 and PRF 80. As it can be seen, the decrease of EGR from 90 to 80 resulted in faster combustion process, characterized by higher HRR peaks. From Table 15 and Table 16, it can be also verified that none of the fuel blends are able to achieve an air mass increase of 10%. In the case of PRF 80, the air mass could be increase just by 3.2% as higher air mass would leads to pressure gradients higher than the maximum allowed.

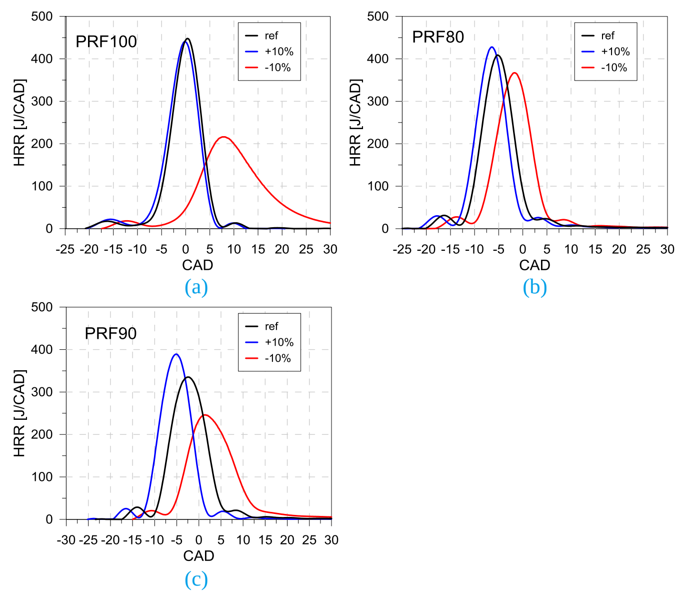


Figure 11. Heat release rate profiles illustrating the different air sweeps for (a) PRF 100, (b) PRF 90 and (c) PRF 80 for 50% of engine load at 1800 rpm.

Table 15. Engine settings and boundary conditions used for the different fuel blends at 50% of engine load.

Parameter→	P_int	T_int	Air mass	air mass change	EGR
Operating condition↓	[bar]	[ºC]	[g/s]	[%]	[%]
PRF 100 dw	2.2	63.7	114.2	-7.6	48.9
PRF 100 ref	2.1	68.2	123.5	0.0	46.4
PRF 100 up	2.2	61.8	133.3	7.9	41.3
PRF 90 dw	2.2	65.0	115.5	-9.7	48.1
PRF 90 ref	2.1	59.5	127.9	0.0	43.3
PRF 90 up	2.2	60.5	140.7	10.1	38.4
PRF 80 dw	2.2	69.7	115.0	-9.6	48.6
PRF 80 ref	2.2	66.1	127.2	0.0	44.4
PRF 80 up	2.3	70.4	131.3	3.2	42.4

Table 16. Engine settings and boundary conditions used for the different fuel blends at 50% of engine load.

Parameter→	SOIpilot	SOImain	PMI	GF
Operating condition↓	[CAD bTDC]	[CAD bTDC]	[bar]	[-]
PRF 100 dw	60.0	50.0	12.4	79.5
PRF 100 ref	60.0	50.0	12.5	79.9
PRF 100 up	60.0	50.0	12.4	80.1
PRF 90 dw	35.0	25.0	12.5	71.8
PRF 90 ref	35.0	25.0	12.5	72.0
PRF 90 up	35.0	25.0	12.4	71.8
PRF 80 dw	34.0	24.0	12.4	69.8
PRF 80 ref	34.0	24.0	12.4	69.7
PRF 80 up	34.0	24.0	12.5	70.0

Figure 12 illustrates the effect of the air variations for the different octane numbers in BSFC and combustion related parameters. PRF 100 presents the highest fuel consumption when the air mass is decreased. This is attributed to the higher combustion duration together with a delayed combustion process. Since PRF 100 has the low reactivity among the three fuel blends, the EGR effect is highlighted impairing the combustion process. Nonetheless, for the reference case and for higher air mass, the trend is inverted. In this case, the lower reactivity of PRF 100 allows to obtain a short and well phased combustion process compared to the other fuel blends. As the reactivity is increased, early CA50 are achieved, releasing a considerable amount of energy at the compression stroke. This negative work is directly translated to a fuel consumption increase. Moreover, it is interesting to note that with exception of the reference case, the pressure gradients increase as the octane number is reduced. For PRF 90 and PRF 80, the pressure gradients values are near of the limiting condition imposed by the engine manufacturer of 15 bar/CAD.

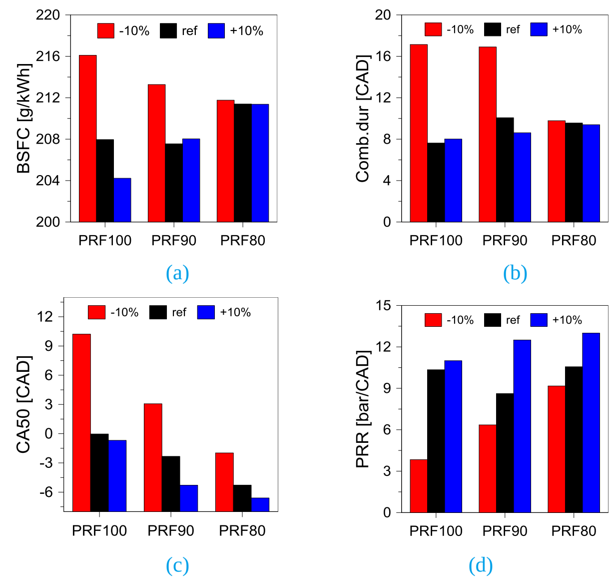
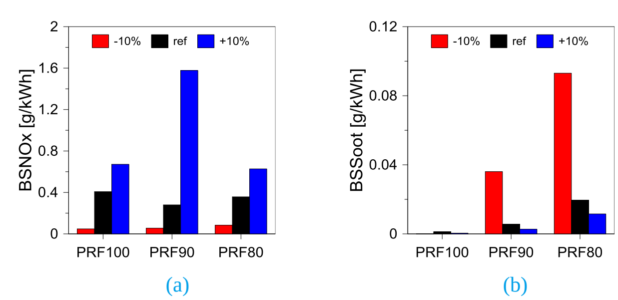
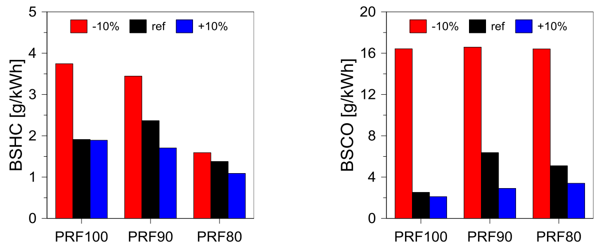


Figure 12. Brake specific fuel consumption (a), combustion duration (b), combustion phasing (c) and pressure rise rate(d) for the different octane number and air sweeps evaluated at 50% of engine load.

From Figure 13, it can be inferred that the octane number variations as well as the air variations have a noticeable impact on the main engine-out emissions. It should be noted that EUVI limits are constraints for the reference case. This means that the NOx and soot emissions should be lower than 0.4 g/kWh and 0.01 g/kWh respectively. This is accomplished for all the fuel blends, as depicted in Figure 13 (a) and Figure 13 (b). However, the air variations demonstrate that the EUVI normative can be only achieved for a small range of settings. If the air mass is increased, soot emissions are strongly reduced. By contrast, the lower dilution levels increase the in-cylinder temperatures, exceeding by far the NOx limits. PRF 90 demonstrated the highest increases, since it allowed the higher air mass modifications, achieving the target of +10%. Even small air increases as the +3.2% for PRF 80 are enough to surpass the normative values.

Air mass reductions (↑EGR) are an effective way to reduce the NOx emissions, allowing to obtain significant reductions compared to the reference case. Nonetheless, the lack of oxygen improves the soot formation. This is more perceptible for PRF 90 and PRF 80, which require lower GF fraction to maintain the pressure gradient under the mechanical limits. This means that more diesel is injected increasing the rich zones inside the combustion chamber, exceeding the normative soot emissions. Regarding the HC emissions, the increase of the fuel reactivity contributes to a better fuel oxidation, decreasing the final emissions of this component as the octane number is reduced. CO emissions do not demonstrated trends with respect to the ON. Both CO and HC increased by the oxygen reduction.





(c) (d)

Figure 13. Brake specific emissions for the engine load of 50% (a) NOx, (b) Soot, (c) SHC, (d) CO for the different fuel blends and air sweeps evaluated.

Conclusions

This paper investigated the impact of the fuel sensitivity and octane number on the combustion, performance and emission parameters of a dual-mode dual-fuel engine by means of experiments. First, the effect of the fuel sensitivity was assessed at different operating conditions for sensitivity values of 0, 5 and 10 and research octane number of 95. From this analysis, it could be concluded that the effect of the sensitivity is dependent on the operation condition, i.e., pressure and temperature values at which the combustion takes place. The major findings can be summarized as follows:

- For low load conditions, the sensitivity effect can be balanced by small modifications in the engine settings. This allowed to obtain iso-HRR independently on the sensitivity tested.
- At medium loads, S=10 increases the mixture reactivity. This is only found in a narrow range at the temperature-induction time diagrams. Therefore, such behavior cannot be considered a global conclusion.
- For high engine loads, the sensitivity decreases the fuel reactivity, indicating that the fuel mixture is operating at an NTC affected zone. Despite of the efforts in modifying the engine settings, iso-HRR operation could not be achieved.

Finally, the octane number effect was assessed by means of air mass variations. This allowed to evaluate the operating limits of each octane number at two fully premixed conditions:

- At 25% engine load, the combustion process seems to be affected in a low degree by the octane differences, mainly at the reference case and for higher air masses. This conclusion can be also extended for the emission results.
- As the engine load is increased, the octane number becomes a dominant parameter, limiting the maximum air increase as the octane number is reduced. Moreover, higher octane numbers demonstrated to be more affected by the EGR increase, resulting in delayed combustion process, higher fuel consumption and poor combustion efficiency.

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Definitions/Abbreviations

aTDC	After Top Dead Center
BSFC	Brake Specific Fuel Consumption
bTDC	Before Top Dead Center
CAD	Crank Angle Degree
CA50	Crank angle at 50% mass fraction burned
CDC	Conventional Diesel Combustion
CO	Carbon Monoxide
CO_2	Carbon Dionoxide
DI	Direct Injection
DMDF	Dual-Mode Dual-Fuel
DPF	Direct Particulate Filter
ECU	Engine Control Unit
EGR	Exhaust Gas Recirculation
GF	Gasoline Fraction
FSN	Filter Smoke Number
HC	Hydro Carbons

HCCI	Homogeneous Charge Compression Ignition
HP	High Pressure
HRF	High Reactivity Fuel
HRR	Heat Release Rate
ICE	Internal Combustion Engine
LHV	Lower Heating Value
LP	Low Pressure
LRF	Low Reactivity Fuel
LTC	Low Temperature Combustion
MCE	Multi Cylinder Engine
MON	Motor Octane Number
NOx	Nitrogen Oxides
NTC	Negative Temperature Coefficient
O_2	Molecular Oxygen
ON	Octane Number
PFI	Port Fuel Injection
PRF	Primary Reference Fuel
RCCI	Reactivity Controlled Compression Ignition
RoHR	Rate of Heat Release
RON	Research Octane Number
S	Sensitivity
SOI	Start of Injection
TDC	Top Dead Center
TRF	Toluene Reference Fuel
VGT	Variable Geometry Turbine