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Performance and Emissions of a Spark Ignition Engine Fueled with Water-in-Gasoline Emulsion Produced through Micro-Channels Emulsification

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Abstract: This paper presents an experimental study investigating the effects of water-in-gasoline emulsion (WiGE) on the performance and emissions of a turbocharged PFI spark-ignition engine. The emulsions were produced through a micro-channels emulsifier, potentially capable to work inline, without addition of surfactants. Measurements were performed at a 3000 rpm speed and net Indicated Mean Effective Pressure (IMEP) of 16 bar: the engine point representative of commercial ECU map was chosen as reference. In this condition, the engine, fueled with gasoline, runs overfueled (λ = 0.9) to preserve the integrity of the turbocharger from excessive temperature, and the spark timing corresponds to the knock limit. Starting from the reference point, two different water contents in emulsion were tested, 10% and 20% by volume, respectively. For each selected emulsion, at λ = 0.9, the spark timing was advanced from the reference point value to the new knock limit, controlling the IMEP at a constant level. Further, the cooling effect of water evaporation in WiGE allowed it to work at stoichiometric condition, with evident benefits on the fuel economy. Main outcomes highlight fuel consumption improvements of about 7% under stoichiometric mixture and optimized spark timing, while avoiding an excessive increase in turbine thermal stress. Emulsions induce a slight worsening in the HC emissions, arising from the relative impact on combustion development. On the other hand, at stoichiometric condition, HC and CO emissions drop with a corresponding increase in NO.

Keywords: water-in-gasoline emulsion; micro-channel emulsifier; experiments; spark-ignition engine; performance; fuel consumption; emissions

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1. Introduction

Water injection (WI) technology is seen as a promising solution to make turbocharged spark ignition (SI) engines operate with higher compression ratios, higher boost pressures and stoichiometric combustions at high loads. This allows a reduction in brake-specific fuel consumption (BSFC) and thus in CO_2 emission on the overall engine map [1,2].

In downsized turbocharged SI engines, at stoichiometric conditions and full load, it is necessary to control knock tendency and to fulfill the maximum allowable temperature at the turbine intake. In commercial SI engines, this is commonly performed by introducing an excess fuel in the combustion chamber and retarding the ignition timing, leading to lower thermodynamic efficiency. Water injection can replace the cooling effect of the air/fuel mixture enrichment with benefits in the BSFC. The introduction of water in the combustion chamber of an SI engine reduces the gas temperature level before the combustion takes place due to the water's high latent heat of vaporization that cools down the air/fuel mixture. Hoppe et al. [2] numerically investigated the relative weight of water vaporization and the sole dilution effect in the charge-cooling process. They compared the effect of liquid- and vapor-water injection on the air–fuel mixture temperature, at the spark timing. They found a reduction in air–fuel mixture temperature with liquid-water injection five times higher than in the case of vapor injection, indicating the evaporation as the main driver of charge cooling. Moreover, the evaporated water acts similar to an inert gas during the combustion process and it reduces the combustion temperature by increasing the global heat capacity [3].

Starting in the 1920s, research has been progressing on the use of water in internal combustion engines [4], and many researchers presented works on water injection in the intake manifold (port injection) or directly in the cylinder (direct injection) of spark-ignition engines [5].

In Reference [6], Zhuang et al. experimentally analyzed the influence of port water injection on the consumption and the thermal efficiency of a turbocharged direct injection engine. The tests were conducted at low speed and medium–high load, at stoichiometric condition. A reduction of about 3.5% in specific fuel consumption and an increase in thermal efficiency of about 1.5% were obtained when the maximum water/gasoline percentage of 50% was injected, thanks to the knock mitigation. The influence of advanced spark timing was found to be dominant compared to the effect of charge dilution on the combustion duration. Therefore, the combustion center was advanced of about 5 CAD. Further advantages can be obtained at high engine speed and full load. As aforementioned, in these conditions turbocharged engines run overfueled to keep the turbine temperature below the mechanical failure threshold. Advancing the knock-limited spark advance allow us to reduce the exhaust temperature, avoiding charge enrichment. Sun et al. investigated the influence of port water injection on the performance of turbocharged direct-injection spark-ignition engine at full-load (18 bar BMEP) conditions. The port fuel injection allowed us to reduce the fuel enrichment of about 13% with a thermal efficiency increase of about 4.5% [7].

Similar results were found by Tornatore et al., who experimentally tested the behavior of a small displacement turbocharged spark-ignition engine equipped with a port-water-injection system. They observed a reduction in fuel consumption between 6 and 12% at full load conditions, at varying engine speeds [1].

In Reference [8], Hunger et al. investigated the influence of direct water injection on knock mitigation and thermal efficiency. They found a relationship between the injection timing and charge cooling effect. When water injection is too advanced, the spray impinges against the cylinder walls, generating a liquid film. The vaporization process starts during the compression stroke; therefore, the water deposited against the liner evaporates extracting heat in part from the end gas and in part from cylinder walls with a drop in knock-suppression efficacy. Furthermore, a proper combustion chamber design is necessary to locate the nozzle, considering the limited available space left by inlet and exhaust four valves, the spark plug and the gasoline injector. On the other hand, port water injection does not allow us to fully take advantage of vaporization heat extraction from the end gas, and, at the same time, it has a limited influence on the volumetric efficiency, as the intake temperature is too low to induce water evaporation in the manifold.

An alternative way of introducing water in a SI engine is under the form of an emulsion with gasoline. This solution shows some advantages compared to the standard separate injections of water and gasoline. Indeed, the water-in-gasoline emulsion (WiGE) technique requires a single injector per cylinder, a lower number of injection control variables and, consequently, a simpler control of fuel injection. Referring to the WiGE very few papers are available in the technical literature discussing its production, employment in SI engines and relative influence on combustion, performance and pollutant emissions. On the other hand, the use of water-in-diesel emulsion (WiDE) in diesel engines has been found to be an economical technique for improved combustion and fuel economy; the presence of water in diesel engines also leads to a drop in NOx formation and in the rate of soot particles [9]. In the case of WiGE, old research studies [10,11] focused on the enhanced knock mitigation capabilities of emulsion fuel and its influence on performance of spark-ignition engines.

As a starting point of Reference [10], the authors stated that when WiGEs are correctly used and the engine is properly adjusted to give optimum performance, water addition lowers gasoline octane number requirement and reduces the thermal stress on the parts of the cylinder–piston group without any loss of maximum engine power or torque and without increasing the specific gasoline consumption. The study reported in Reference [11] also confirmed the improved gasoline octane number with the employment of water addition in the form of emulsion. In Reference [12], an analysis on the effects of supplementing gasoline with water on spark-ignition engine performance and emissions is reported. Experiments on a single-cylinder engine, engine cycle simulations, and vehicle tests were performed. This research showed that the concept of adding water to gasoline presents some negative aspects consisting in the increased hydrocarbon emissions and decreased vehicle drivability. The resulting main benefits of water–gasoline fuels were the higher octane ratings and the decreased nitric oxide emissions. A recent paper reports the outcomes of a basic research on direct injection of gasoline/water emulsion. It shows the influence of the amount of water in a WiGE on the spray evolution in a high-pressure chamber [13]. The main evidence of this study is represented by the opportunity to optimize the direct injection of WiGE to improve fuel consumption and emissions in DISI engines.

On the other hand, the scientific literature is very poor in the use of water–gasoline emulsions in modern turbocharged SI engines, and WIGE potential in fuel enrichment suppression has not yet been explored. Moreover, there is still a lot of progress to be made in the selection of the emulsifiers and in surfactant with the aim to guarantee the stability of the emulsion fuels for a regular on-board vehicle storage before its use [14]. Surfactants have an additional cost, and, according to the literature papers, they can lead to hazardous emissions during combustion, based on their chemical nature [15]; thus, the use of a stable emulsion with a very low level of surfactant is desired.

Starting from the above discussed state-of-the-art, this paper investigates the effects of water-in-gasoline emulsions on the performance and emissions of a turbocharged port fuel injection (PFI) spark-ignition engine. WiGE is produced through a prototype micro-channel emulsifier [16] that allows for the use of a very small amount of surfactant to create stable emulsions.

Further development of the prototype emulsifier will be aimed at using this device inline by the integration with the engine fuel injection system; this will allow for us to have a very flexible fueling system and to completely quit the use of surfactants, since it will not be necessary to face the issues associated with a long-term WiGE storage. In this way, using WiGEs will allow us to introduce water into the combustion chamber, without equipping the engine with a secondary injection system.

2. Experimental Setup and Procedures

The experiments for this work were carried out on a downsized and turbocharged SI engine equipped with port fuel injectors. The engine has 2 cylinders and 4 valves for each cylinder; its main characteristics are shown in Table 1.

Table 1. PFI twin-cylinder engine's main characteristics.

Characteristic	Dimension
Displacement (cm^3)	875.4
Compression ratio	10:1
Bore (mm)	80.5
Stroke (mm)	86.0
Connecting Rod (mm)	136.8
Rated Power (kW)	63.7 @ 5500 rpm
Max Torque (Nm)	146.7 @ 2000 rpm

The Mitsubishi turbocharger allows us to reach boost pressures up to 2.4 bar. The turbine volute presents a waste-gate valve that allows us to control the maximum admissible turbocharger speed and, thus, turbine inlet and compressor outlet pressures. The compressor is equipped with a dump-valve to prevent surge occurrence.

A sketch of the engine test bench is shown in Figure 1. An air-handling unit constantly supplies intake air to the engine compressor at a temperature of 293 K.

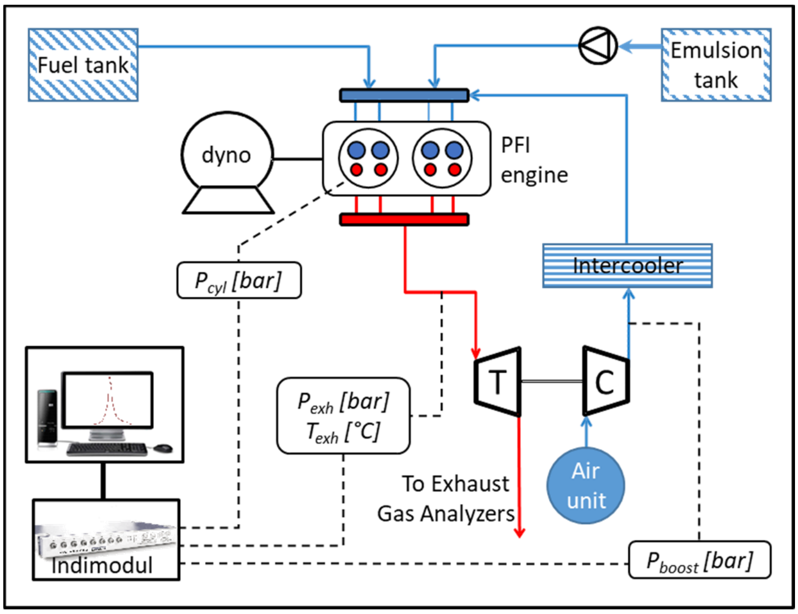


Figure 1. General layout of the experimental setup.

In-cylinder pressure signals are detected by using one piezo-quartz pressure transducer (accuracy of ±0.1%) for each cylinder. The boost pressure and the pressure upstream of the turbine are measured through piezo-resistive low-pressure indicating sensors installed at the compressor outlet and in the exhaust manifold. Exhaust flow temperature is monitored by using a thermocouple to check that the value does not exceed the allowable temperature limits for the turbine inlet.

The emulsion injection system is mainly composed of a commercial liquid pump to pressurize the fluid at 4 bar and a rail to accumulate a proper amount of emulsion upstream of the injectors.

The engine operation can be controlled by a commercial engine control unit (ECU) or alternatively by a prototype driver managed by a LabView software tool that is able to switch from the reference commercial ECU to the external mode [17]. Using the second option, it is possible to control the main engine parameters, such as spark advance (SA), start and duration of gasoline and emulsion injection. The fuel injection quantity is fine monitored to get the selected air/fuel (A/F) ratio according to the lambda sensor placed at the engine exhaust.

Crank-angle-degree resolved data are acquired by using an indicating system (AVL IndiModul) coupled to a shaft encoder. A combustion analysis software, AVL INDICOM, allows for signal processing and data saving. In-cylinder pressure data are collected with a high sampling resolution (0.1 CAD), from −90 to 90 CAD ATDC; outside of this angular interval, the sampling resolution is set at 1 CAD.

It is worth highlighting that the engine speed/load condition was actually controlled through the dynamometric brake, while a refined monitoring of IMEP level was performed through the indicating software. Engine IMEP was measured as the mean of single cylinders IMEPs. Therefore, controlling IMEP instead of BMEP allowed us to monitor the effects of cycle-resolved, cycle-to-cycle and cylinder-to-cylinder variations.

Regulated exhaust gaseous emissions and CO_2 were measured: hydrocarbon (THC), CO and CO_2 by means of a nondispersive infrared (NDIR) sensor, while an electrochemical sensor measured NOx.

During experiments, a reference engine operating condition was chosen among the points stored in the commercial ECU (full gasoline conditions). The selected point represents a knock-limited engine operation and is characterized by a speed of 3000 rpm, a high net IMEP of 16.0 bar, a rich relative A/F mixture (λ = 0.9) and the spark timing (ST) at −10 CAD ATDC. This condition was set as representative of the overfueling strategies adopted in downsized engines at high loads: knock occurrence is avoided by delaying the combustion phasing towards the expansion stroke with an increase in the temperature at the turbine inlet. For this reason, in order to preserve exhaust engine hardware, the ECU adopts fuel-enrichment strategies which reduce the exhaust gas temperature, with penalties over fuel consumptions. As an alternative knocking suppression technique to overfueling strategies, water was injected in the engine through emulsions with gasoline. Two different WiGEs were tested: WiGE 10 and WiGE 20. The first one consists of an emulsion containing 10% of water and 90% of gasoline by volume, while the second one contains 20% water and 80% gasoline.

As is well-known, an emulsion is a fine dispersion of two liquid phases. Details concerning the adopted emulsification technique can be found in Reference [18]. Briefly, two controlled fluxes of water and oil impact in a cross-section through properly dimensioned micro-channels. As a result of the impact, the water is atomized in microdroplets inside the oil matrix. Depending on the physical properties of the oil (viscosity, surface tension, etc.), the emerging emulsion is stable for a variable time, which is long enough to be used in inline feeding systems (some hours before water separation). In the present case, the emulsion is produced offline, and a small amount of nonionic surfactant (SPAN80 0.2%v) is added to preserve the emulsion stability.

First, the engine was run with gasoline in the reference engine operating condition. Then, at the same speed, gasoline was replaced with WiGE 10. The IMEP and the lambda values were kept the same as the reference condition by fine-tuning the injection duration (DOI) and plenum pressure, for a peer-to-peer comparison with reference gasoline case. Then, the spark timing was advanced until reaching the new knocking limited spark advance. Aim of the present work is to investigate the potential of WiGE injection as a knock mitigation strategy alternative to fuel enrichment. After the tests under a rich relative A/F ratio (λ = 0.9), WiGE injection duration was changed in order to reach a stoichiometric A/F mixture (λ = 1.0), while the IMEP was kept always the same of reference level. A spark timing sweep was performed under stoichiometric condition starting from the reference spark timing until reaching the new knock limit. This procedure was repeated for WiGE 20, and the overall test matrix is shown in the following Table 2. Gasoline and WiGE presented the same injection timings: −165 CAD ATDC.

During the experiments, the turbine inlet temperature (TIT) was always kept below 950 °C, and the maximum in-cylinder pressure was below 85 bar (±5 bar), as that is the maximum allowable peak pressure to preserve the engine from mechanical failure of components. To this aim, the maximum boost level was automatically controlled in the range 1.8–2.0 bar, acting on the waste-gate valve opening. The coolant temperature was set at 85 ± 1 °C, using a water heat exchanger.

Table 2. Test conditions.

Fuel	λ [±0.01]	ST [cad atdc]	DOI [cad]	P_{INT} [bar]
gasoline	0.90	−10	195	1.72
WiGE 10	0.90	−10	230	1.74
WiGE 10	0.90	−11	226	1.70
WiGE 10	0.90	−12	223	1.70
WiGE 10	0.90	−13	220	1.69
WiGE 10	1.00	−10	215	1.78

Table 2. *Cont.*

Fuel	λ [±0.01]	ST [cad atdc]	DOI [cad]	P_{INT} [bar]
WiGE 10	1.00	−11	213	1.77
WiGE 10	1.00	−12	210	1.75
WiGE 10	1.00	−13	208	1.73
WiGE 10	1.00	−14	206	1.76
WiGE 20	0.90	−10	255	1.70
WiGE 20	0.90	−12	255	1.70
WiGE 20	0.90	−14	255	1.70
WiGE 20	0.90	−16	255	1.70
WiGE 20	1.00	−10	250	1.84
WiGE 20	1.00	−12	243	1.81
WiGE 20	1.00	−14	235	1.78
WiGE 20	1.00	−16	230	1.74
WiGE20	1.00	−19	225	1.72

3. Results

The effect of WiGE on the engine performance was investigated through the in-cylinder pressure analysis. As aforementioned and further discussed in the following, with reference to the TIT results, the gasoline case at stoichiometric A/F mixture, and SA = −10 CAD ATDC is missing because it is not feasible with the maximum TIT target. Figure 2 shows a comparison between the gasoline and WIGE in-cylinder pressure traces at the reference engine operating condition (λ = 0.9, SA = −10 CAD ATDC). As aforementioned, this condition is representative of commercial ECU map and the spark timing under rich air/fuel mixture corresponds to the gasoline knock limit. When switching to WiGE, the injection duration and the plenum pressure were adjusted to keep the A/F ratio and IMEP load constant. The first effect of WiGE is the charge cooling, due to the evaporation of water droplets; moreover, the water in the combustion chamber acts as an inert during the combustion process; this causes a slowdown of the rate of energy release with a reduction in the pressure peak proportional to the water content.

Figure 3 shows the relationship between water content and combustion duration and phasing, at different spark timings and relative A/F ratios and WiGEs, considering a representative engine cylinder (Cyl #2). For each spark timing sweep, a clear indication of knock-limit (KL) point is depicted in Figure 3a. This indication of KL points extends to the other figures proposed below. In agreement with the pressure traces shown in Figure 2, at SA = −10 CAD ATDC and λ = 0.9, the use of WiGE prolongs the combustion duration (Figure 3a) and delays the combustion phasing (Figure 3b). On the other hand, proportionally to the water content, the cooling and dilution effects of WiGE on the in-cylinder charge mitigate the knock tendency and allow us to advance the spark timing up to −13 and −16 CAD ATDC (for WiGE 10 and WiGE 20, respectively); consequently, also the combustion center is advanced with respect to gasoline reference case. At stoichiometric condition, the knock-limited spark timing can be further advanced with the result of a better combustion phasing (MFB50 = 12.4 CAD ATDC @ SA = −19 for WiGE20).

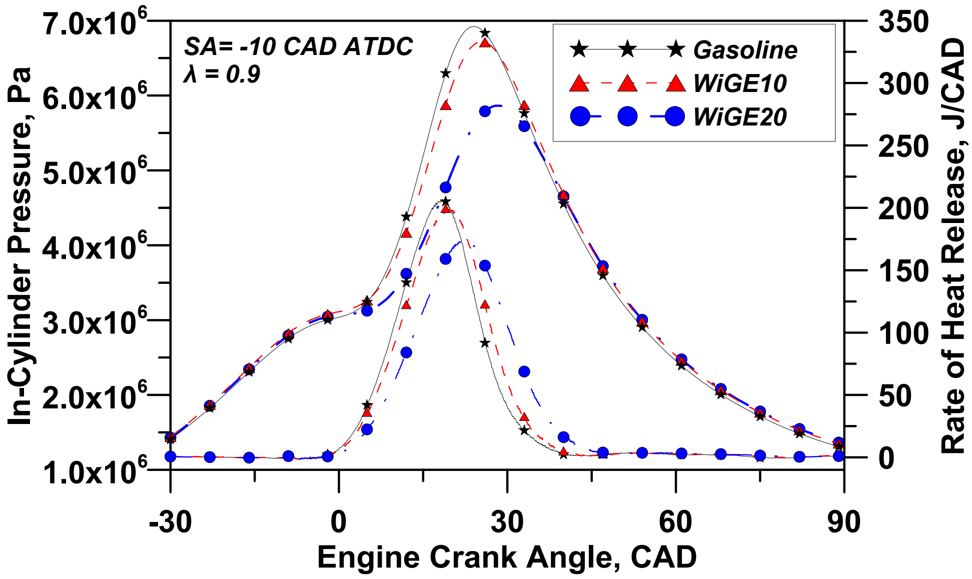
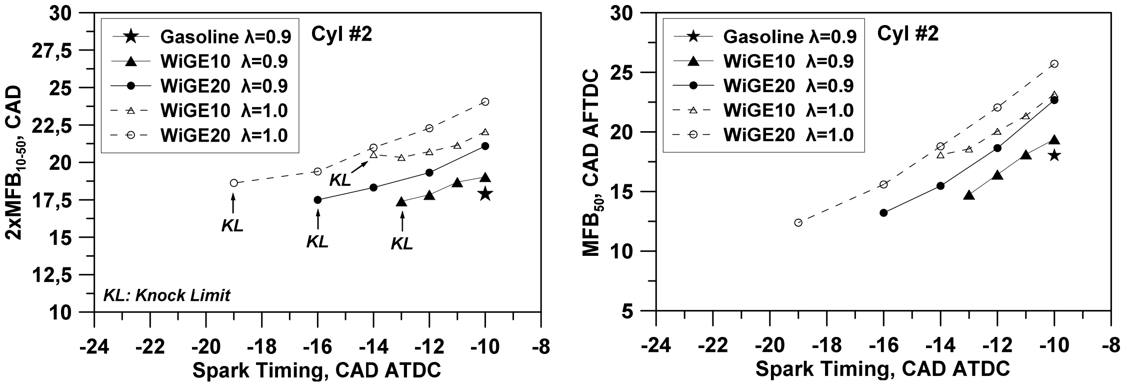


Figure 2. In-cylinder pressure trace and rate of heat release at SA = −10 CAD ATDC, λ = 0.9.

Figure 4 shows the trend of TIT against spark timing and relative air-to-fuel ratio. As shown, at a reference spark timing of −10 CAD ATDC, the turbine inlet temperature is almost the same for gasoline and WiGE in rich mixture condition. Of course, the rich A/F mixture at the reference point was selected in the manufacturer calibration to avoid, with a certain safety margin, the knock onset and the excessive thermal stress to the turbine. On the other hand, in the case of the engine mounted on the test bench, a lower heat transfer can be realized for the turbine compared to the case of an engine on the real vehicle. Based on this consideration, even if the allowable maximum TIT is 950 °C, a target of 819 °C, which is representative of the gasoline reference condition, was set for WiGE. In light of the above discussion, the criterion followed for the engine tests is the identification of knock-limited operations with WiGE and stoichiometric mixture realizing a TIT level equal or lower to the one achieved at the reference ECU condition and a lower ISFC. By increasing the spark advance, the earlier combustion phasing allowed by WiGE results in a significant reduction in the turbine inlet temperature, because the temperature of the in-cylinder gases is lower at the opening of the exhaust valves if the combustion takes place early in the engine cycle. The use of WiGE10 does not allow us to reach the TIT target at stoichiometric condition (TIT = 827 °C at knock limit; SA = −14 CAD ATDC), while WiGE20 resulted in a lower TIT than gasoline reference case at the two most advanced spark timing: 818 °C at SA = −16 CAD ATDC, and 804 °C at SA = −19 CAD ATDC.

The Indicated Specific Fuel Consumption (ISFC) for all the investigated WiGE fuels, spark timings and relative air-to-fuel ratio is shown in Figure 5. It is worth pointing out that fuel consumption is estimated by considering only the gasoline content of WiGEs, as the water is inert.

As expected, introducing water in combustion chamber prolongs the combustion duration with a worsening in efficiency. Hence, at fixed spark timing (−10 CAD ATDC) and at the same air/fuel ratio (λ = 0.9), fuel consumption associated to WiGE is higher than the one measured in the reference condition. When leaning WIGE/air mixture the fuel efficiency is improved. On the other hand, this improvement is not large enough to eliminate the gap with gasoline reference case. Therefore, at fixed spark timing (−10 CAD ATDC), the WiGE fuel consumption at stoichiometric condition is still higher than the one measured for gasoline at rich condition (λ = 0.9).



(a) (b)

Figure 3. Combustion duration (2xMFB10-50) (**a**) and combustion center (MFB50) (**b**) against spark timing and relative air-to-fuel ratio for WiGE 10 and WiGE 20.

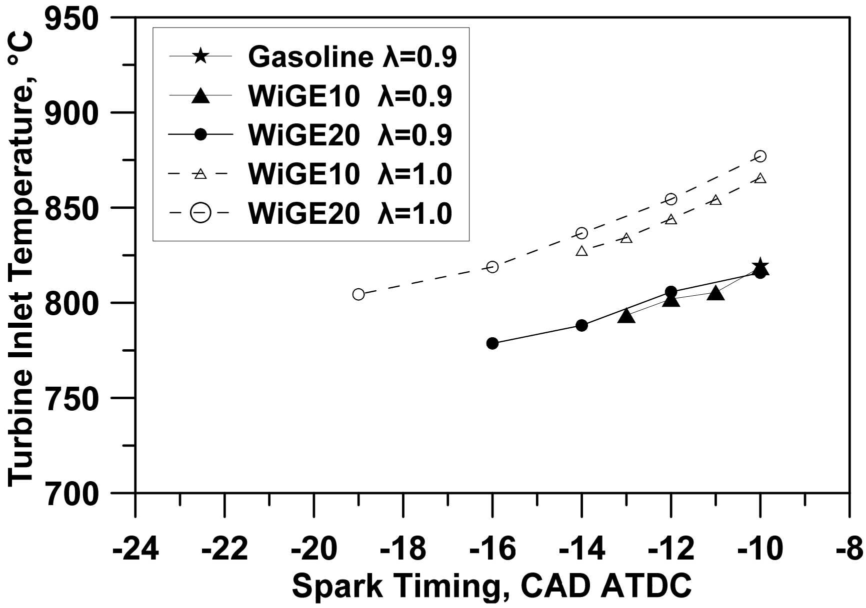


Figure 4. Turbine inlet temperature against spark timing and relative air-to-fuel ratio.

Thus, to achieve the same IMEP as the reference case, it is necessary to inject a higher amount of gasoline. Higher consumption is measured for WiGE20 compared to WiGE10. This behavior is probably ascribed to the water-induced lengthening of combustion process.

A decreasing trend of ISFC with advancing the spark timing is observed for both air-to-fuel ratios and both emulsions. At $\lambda = 0.9$ and reference spark timing (SA = −10 CAD ATDC), WiGE employment induces a worsening in the ISFC values compared to gasoline reference condition, due to combustion duration extension induced by water. On the other hand, the improved knock resistance related to water presence allows us to optimize the combustion phasing. The discussed water effects are mutually conflicting for the engine efficiency, and ultimately the ISFC level of KL points (for WiGE 10 and 20) under the rich A/F mixture never reaches the one attained by the ECU-reference rich gasoline case. At engine operation under stoichiometric A/F mixture, the combined effect of WiGE knock mitigation and the leaner mixture allows for an ISFC reduction of about the 3.7% and 7.1% compared to ECU-reference gasoline point at rich mixture condition, for WiGE 10 and WiGE 20, respectively.

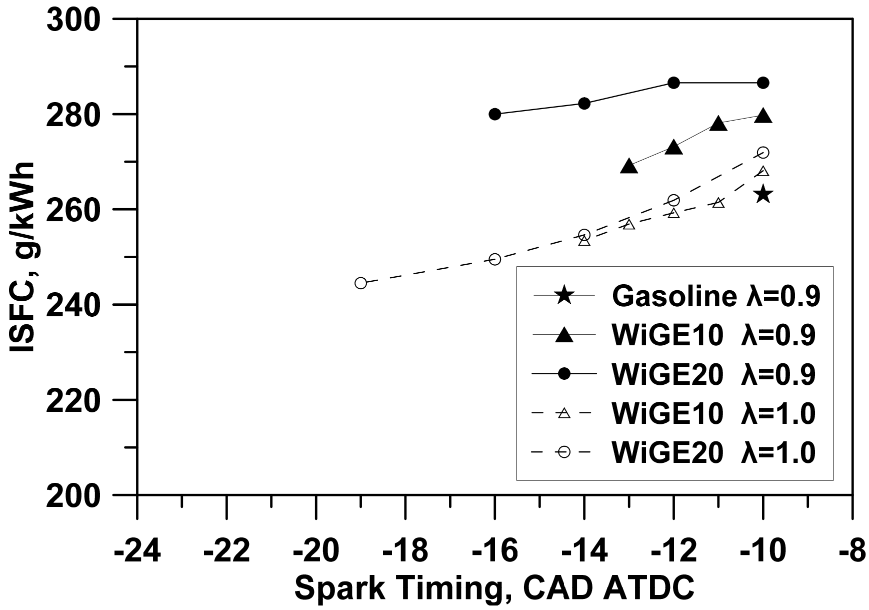


Figure 5. Indicated Specific Fuel Consumption against spark timing and relative air-to-fuel ratio.

Figure 6 shows the correlation between the CO and CO_2 exhaust emissions and emulsion fueling. For the rich A/F mixture condition, coherently with fuel consumption results, an increase in CO_2 is measured for emulsions (both with WiGE 10 and 20) if compared with the ECU reference gasoline case, while the CO is almost similar. In stoichiometric A/F mixture condition, CO_2 emissions for emulsions are higher than the ones measured under rich condition, due to a more complete combustion process.

Figure 6a also shows that the main CO variations have to be attributed to the modification of the A/F mixture quality. As expected, a strong reduction in CO of one order of magnitude is measured when switching from the rich to stoichiometric A/F mixture.

Figure 7 shows the variation of HC and NO emissions against the spark advance at the selected relative air-to-fuel ratios and for all the investigated fuels. At the rich mixture condition, the WiGE combustion produces larger HC, proportional to the water content, almost independent of the spark timing. Similarly, at stoichiometric condition, the increase in water content in the emulsion induces slightly higher unburned HC emissions. On the other hand, for a fixed water-in-gasoline emulsion, a reduction in the unburned HC is observed passing from the rich to stoichiometric A/F mixture.

Referring to NO emissions, at rich condition, the switch from full gasoline to WiGE involves a reduction in NO. This reduction is more marked for WiGE 20 than WiGE 10, and it is due to the combined effect of charge cooling and thermal dilution.

It is the case to highlight that, even with WiGE 20, the water quantity contained in the in-cylinder charge is low and the corresponding vapour conversion (evaporation process) is constrained by thermodynamic conditions, available time and full humidity conditions. However, the high latent heat of vaporization for water has a relevant role on the reduction of in-cylinder mixture temperature. As reported in Reference [19], at the spark timing, a reduction in charge temperature of 13 °C can be achieved at 3500 rpm and 17 bar BMEP when injecting 17%w of water in the intake manifold.

In stoichiometric conditions, the increase in oxygen content and the higher combustion temperature due to the lack in any diluent (excess fuel or water) promote NO formation, with a maximum NO level at KLSA for both WiGE 10 and WiGE 20, due to the increase of in-cylinder peak temperature by advancing the spark timing.

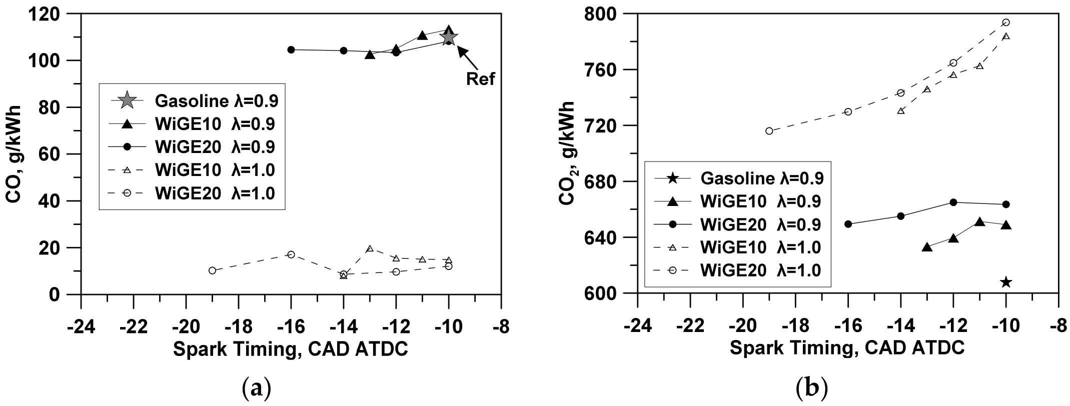


Figure 6. CO (**a**) and CO_2 (**b**) emissions against spark timing and relative air-to-fuel ratio.

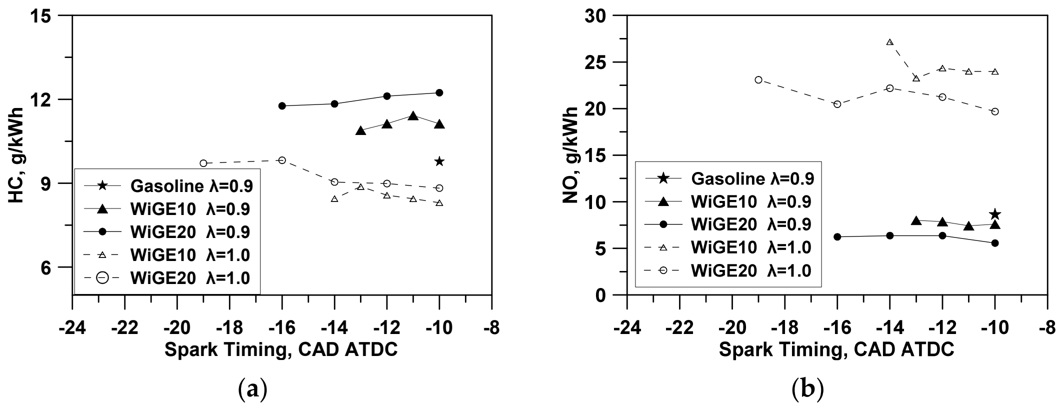


Figure 7. HC (**a**) and NO (**b**) emissions against spark timing and relative air-to-fuel ratio.

4. Conclusions

In this work, an experimental study was carried out to quantify the influence of WiGEs on combustion, performance and exhaust emissions at a high-load knock-limited operation of a small turbocharged PFI spark-ignition engine. The emulsions are produced through a micro-channels emulsifier, potentially capable to work inline, without the addition of surfactants. Two different water contents in emulsion were tested, 10% and 20% by volume, respectively. As in the present work, the emulsion is produced offline, and a small amount of nonionic surfactant (SPAN80 0.2%v) is added to preserve emulsion stability. Engine tests are performed with full gasoline and emulsions injected in the intake runners and considering a reference engine point at a speed of 3000 rpm and 16 bar IMEP. For the selected operating condition, the standard ECU calibration is applied in gasoline mode, running the engine under rich air/fuel mixture (λ = 0.9) and a spark advance (SA = -10 CAD AFTDC) at knock limit. Starting from the above reference point, a spark timing sweep is realized for each WiGE up to the new knock-limited condition, keeping the IMEP constant. Further, the cooling and dilution effects of water evaporation in WiGE has allowed us to work at stoichiometric condition. Engine overall performance, in-cylinder pressure traces and pollutant emissions are measured in each tested condition.

The analysis of experimental in-cylinder pressure cycles and burn-rate profiles show that the water presence in the combustion chamber produces a cooling and dilution effect of charge, inducing an increasing slowdown of combustion velocity and a lowering of pressure peak with the water content. On the other hand, the cooling and dilution effects of WiGE allow us to mitigate the knock occurrence and, consequently, to advance the spark timing, reaching an optimized combustion phasing.

A decreasing TIT trend with spark timing is observed, and the measured TIT level under stoichiometric mixture at the most advanced spark timing reaches a quite similar value to the one attained in the reference gasoline condition.

Relevant ISFC benefits are realized with WiGE 10 (3.7%) and WiGE 20 (7.1%) in stoichiometric mixture and optimized combustions.

Concerning the exhaust emissions, a comparison with the reference gasoline mode highlights a slight increase in HC emissions with a corresponding reduction in NO when using WiGEs. When switching at stoichiometric A/F ratio, the more complete combustion results in a certain reduction in HC, while major penalties for NO are found. One order of magnitude reduction in CO levels is obtained.

Summarizing, water-in-gasoline emulsions demonstrated to be a technique to improve fuel consumption at medium/high loads of turbocharged spark-ignition engines, enabling the stoichiometric combustions, while preserving the turbine blades from severe thermal stresses.

Author Contributions: Conceptualization, methodology and formal analysis, L.M., C.T., L.T., P.M. and J.B.; experimental setup, investigation, data acquisition and data curation, L.M., C.T. and L.T.; writing—original draft preparation, L.M., C.T., L.T., P.M. and J.B.; writing—review and editing, visualization and supervision, L.M., C.T., L.T., P.M. and J.B. All authors have read and agreed to the published version of the manuscript.

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

Abbreviations

A/F	air-to-fuel
ATDC	After Top Dead Center
BSFC	brake-specific fuel consumption
CAD	crank angle degree
DISI	direct-injection spark ignition
DOI	duration of injection
ECU	engine control unit
IMEP	indicated mean effective pressure
ISFC	Indicated Specific Fuel Consumption
KL	knock limit
KLSA	knock-limited spark advance
MFB10	10% of mass fraction burned
MFB50	50% of mass fraction burned
MFB10-50	combustion core duration
NDIR	nondispersive infrared
PFI	port fuel injection
ROHR	rate of heat release
SA	spark advance
SI	spark ignition
ST	spark timing
TDC	Top Dead Center
THC	Total Hydrocarbon Content

TIT	turbine inlet temperature
WI	water injection
WiDE	water-in-diesel emulsion
WiGE	water-in-gasoline emulsion

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