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A numerical investigation on the potentials of water injection to increase knock resistance and reduce fuel consumption in highly downsized GDI engines

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Abstract

3D CFD analyses are used to analyse the effects of port-injection of water in a high performance turbocharged GDI engine. Particularly, water injection is adopted to replace mixture enrichment while preserving, if not improving, indicated mean effective pressure and knock resistance. A full-load / maximum power engine operation of a currently made turbocharged GDI engine is investigated comparing the actual adopted fuel-only rich mixture to stoichiometric-to-lean mixtures, for which water is added in the intake port under constant charge cooling in the combustion chamber. In order to find the optimum fuel/water balance, preliminary analyses are carried out using a chemical reactor to evaluate the effects of charge dilution and mixture modification on both autoignition delays and laminar flame speeds. Thanks to the lower chemical reactivity of the diluted end gases, the water-injected engine allows the spark advance (SA) to be increased; as a consequence, engine power target is met, or even crossed, with a simultaneous relevant reduction of fuel consumption.

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Keywords: tabulated chemistry; water injection; fuel consumption; knock

1. Introduction

Engine downsizing is a standardized practice for the standard passenger car market, while it is a more recent approach on the high performance engine side, due to the potential loss of appeal that could derive from the displacement reduction for the specific application. Nevertheless, a trend can now be seen in this market segment,

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where a new generation of highly downsized engine with specific power outputs around or above 150 HP/litre is emerging. The increased in-cylinder pressure in SI engines has always been limited by the arising of abnormal combustion phenomena, which are considered nowadays one of the most severe performance limiters in SI engines [1-4]. To prevent, or at least limit, the risk of knock onset, charge cooling by means of the adoption of fuel-rich mixtures is usually applied. Despite the implied reduction of engine efficiency (or related increase of specific fuel consumption), mixture enrichment is preferable to the reduction of either the SA or the boost level, since these last are responsible for the amount of indicated work. The use of very rich mixtures is also effective to limit the exhaust gas temperature at turbine inlet.

An alternative option to avoid fuel-rich mixtures could be the use of water injection to achieve the same charge cooling effect with reduced gasoline amounts. While frequently dismissed as simply a remedy for poor intercooling, the benefits of water injection stretch beyond simply cooling engine intake air, and can lead to more power than intercooling alone [5, 6]. The application of water cooling is not a novelty in internal combustion engines, and the first studies on its potential on knock inhibition can be traced back to the early 1930's studies of Sir Harry Ricardo. Much later, a limited number of road vehicles with large-displacement engines from manufacturers such as Chrysler have included water injection. Saab offered water injection for the Saab 99 Turbo. With the introduction of the intercooler the interest in water injection disappeared.

Water injection may improve not just power output and emissions, but also fuel economy, if carefully included in the design and optimization process of the engine. To confirm this statement, the proposed activity aims at using water injection in the intake port to reach the same knock limited condition as a rich-mixture operated engine , while injecting a stoichiometric amount of fuel (thus reducing fuel consumption) and trying to maintain (or even increasing) its brake performance. Since the presence, although modest, of an inert such as water simultaneously affects both the laminar flame speed and the end gas autoignition tendency, an optimized injection strategy was found with the help of 3D-CFD simulations for both water and gasoline in order to maximize the ignition delay of the end gases while limiting the reduction of the laminar flame speed due to the increase of EGR.

2. Methodology

As stated earlier, water injection potential for knock suppression is investigated through the analysis of a currently made turbocharged GDI engine for high performance car applications. In the engine, fuel-rich mixtures are adopted at full-load operations to avoid knock inception and to limit the turbine inlet temperature, this resulting in high levels of brake specific fuel consumption. Since the engine is operated at the edge of knock, as documented in previous publications [7, 8], it provides a useful benchmark to investigate the effects of water injection on the performance limit. Investigations are carried out by means of combined 0-D detailed chemistry and 3D-CFD full engine cycle simulations, in order to analyse a wide range of design parameters and to avoid the limitations and the costs of the experimental practice. Detailed autoignition and laminar flame speed 0-D calculations are initially performed to find the trade-off between the variation of the ignition delays and the variation of the laminar flame speed (both, in turn, originated by modifications in the fresh gas thermal state and composition), with reference to the fuel-only case. Based on the indications from the 0-D chemical simulations, 3-D analyses are conducted to test different injection strategies (for both water and fuel), water injection locations, injected water quantities and equivalence ratios. Results are evaluated focusing on the engine performance and knock tendency. In particular, in order to accurately take into account the effects of water injection on the combustion process development and on the knock inception, a specific procedure implemented by the research group is used based on a tabulated-chemistry approach for auto-ignition, as described in [9, 10]. Particularly, a purposely coded look-up table based knock model is here applied for knock tendency predictions, being it an optimal trade-off between the need to accurately represent the chemical kinetics of the unburnt charge and CPU costs. The knock model used in the current work, successfully applied by the research group [9, 10], is based on the mass fraction of an intermediate fictitious species for autoignition called Y_{IG}, as defined by Lafossas et al. [11]:

$$\frac{dY_{IG}}{dt} = Y_{TF} \cdot \frac{\sqrt{\tau^2 + 4(1-\tau)\frac{Y_{IG}}{Y_{TF}}}}{\tau} \tag{1}$$

A numerical criterion is then adopted to discern the occurrence of knock, based upon the equality between the mass fraction of Y_{IG} and the mass fraction of fuel tracer Y_{TF}, this latter being defined as the “non-reacting” fuel. As the knock model is based on two transported scalars (Y_{IG} and Y_{TF}), it is then straightforward to define a third scalar variable, here called Knock Tolerance (*KT*), as the difference between the two species (2):

$$KT(\vec{x},t) = Y_{TF}(\vec{x},t) - Y_{IG}(\vec{x},t) \tag{2}$$

Focusing on a RON98-E0 surrogate model, the dependence of the mixture reactivity on equivalence ratio and temperature is obtained by its autoignition delay map reported in Fig. 1, as computed from a constant pressure reactor. The end-gas thermo-physical range is highlighted in Fig. 1, and it is comprised between 700K and 900K, from slightly lean to quite rich mixtures (Φ from 0.9 to 1.4, as a first estimate). This visual assessment is useful to gain a first-glance sensitivity of the gasoline model to temperature and mixture variations, both of which can be locally relevant in the investigated GDI engine. As expected, a strong dependence on temperature is verified for all the equivalence ratios. However, for mixtures close to stoichiometry and slightly lean a non-negligible reactivity dependence emerges also on local equivalence ratios. Hence, a detailed description of fuel mixing in the periphery is fundamental in order to retrieve an accurate estimation of the ignition delay.

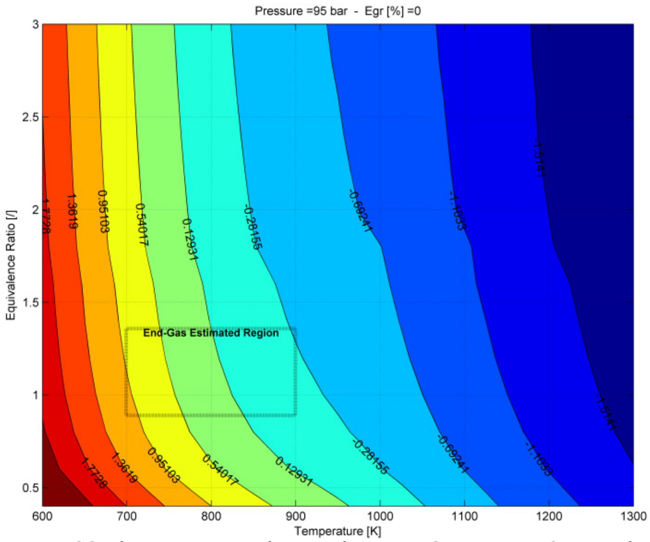


Fig. 1. Log10 iso-lines of autoignition delay for RON98-E0 as a function of mixture and temperature. The region for end-gas physical status is highlighted.

3. The engine

The studied engine is a 8 cylinder V-shaped direct injection spark ignition (DISI) turbocharged engine, whose characteristics are reported in the Table 1 below.

Table 1. Engine parameters

Displacement	≈3800 cm3
Bore/Stroke Ratio	> 1
Max Power	> 500 kW @ 7000rpm

The investigated operating point is the 7000rpm WOT full load/peak power one.

4. Simulation Models

4.1 0-D simulations

The first part of this study is developed in a 0-D framework, to identify a trade-off region between the increase in autoignition delay time and slow down of burning velocity due to EGR addition. While the former is a desired result, the latter is a drawback brought in by this technology. Focusing on the burning velocity side, the sensitivity of this parameter to EGR addition is evaluated in a 0-D approach based on the reasonable assumption that the EGR rate is not influencing the in-cylinder turbulence level, since water is injected in the intake port by a low-pressure injection system. This allows to consider the turbulent burn rate as dependent on the laminar burning velocity, and whatever change in burning rate is introduced by EGR, this can be evaluated from the Metghalchi and Keck correlation for laminar flame speed [12] which is adopted in the 3D-CFD code. The reference combustion duration is the one given by the reference 3D-CFD simulation, without water addition. Regarding the increase in autoignition delay times, these are directly evaluated from the chemical kinetics software for the various EGR rates, through a 0-D constant pressure reactor; their calculation is essentially 'laminar' and well suited to the 0-D evaluation here proposed.

4.2 In-cylinder simulations

Combustion analyses are developed in the framework of Star-CD 4.20, licensed by CD-adapco. The combustion model adopted for the analyses is the ECFM-3Z, whose well-known suitability for premixed as well as diffusive and autoigniting combustion regimes is a fundamental requirement for the present study. In particular, knock is taken into account thanks to the previously described hybrid user-coded/standard formulation. The fuel is injected by means of a 7-hole injector placed between the intake valves. The lagrangian phase is modelled by means of a user coded routine [13] for primary breakup, the Reitz model for secondary break-up and the Bai-Gosman approach for droplet-wall interaction, for both fuel and water. In order to achieve a converged solution, several preliminary RANS cycles are performed based on periodic boundary conditions from a 1-D model of the whole engine provided by the engine manufacturer. Once a cyclic-converged cycle is obtained, this represents the baseline for the subsequent analyses.

5. 0-D Results

The reference condition for the 0-D analyses is the experimental one, where a rich mixture (Φ=1.21, fuel only) meets the edge-of-knock condition. The reduction in fuel is compensated by the addition of water to meet the same charge cooling. The evaluated conditions are reported in Table 2. For each status, the balance between combustion extension and reduction in autoignition tendency is reported in Fig. 2.

Table 2. Physical conditions evaluated

Condition Name	Equivalence Ratio	Injected Water [mg]
Reference	1.21	-
A	1.1	4.93
B	1.0	9.24
C	0.9	13.55

As visible in Fig. 2, starting from the reference condition there is a large advantage in operating a simultaneous mixture leaning and water addition. This is verified in both A and B conditions (respectively Φ=1.1 and Φ=1.0), where the increase in AI delay times is larger than the extension in combustion duration. For these conditions, a reduction in knock tendency is expected together with a reduction in fuel consumption. The latter is particularly true for B condition (Φ=1.0), which will be carefully evaluated in the second part of the study. Mixture leaning starts to become detrimental for condition C, where the negative effect (longer combustion) prevails over the reduction in end-gas reactivity. This preliminary 0-D evaluation identifies the stoichiometric condition with 9.2 mg of injected water as the best trade-off.

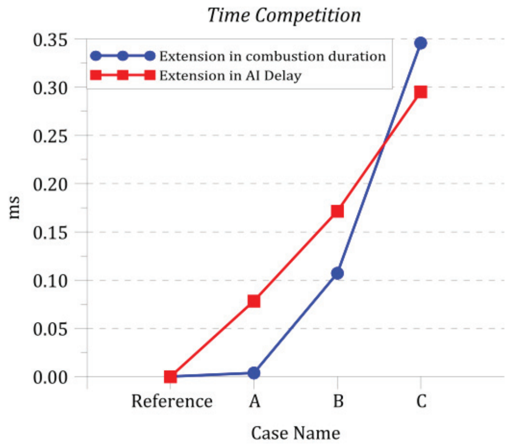
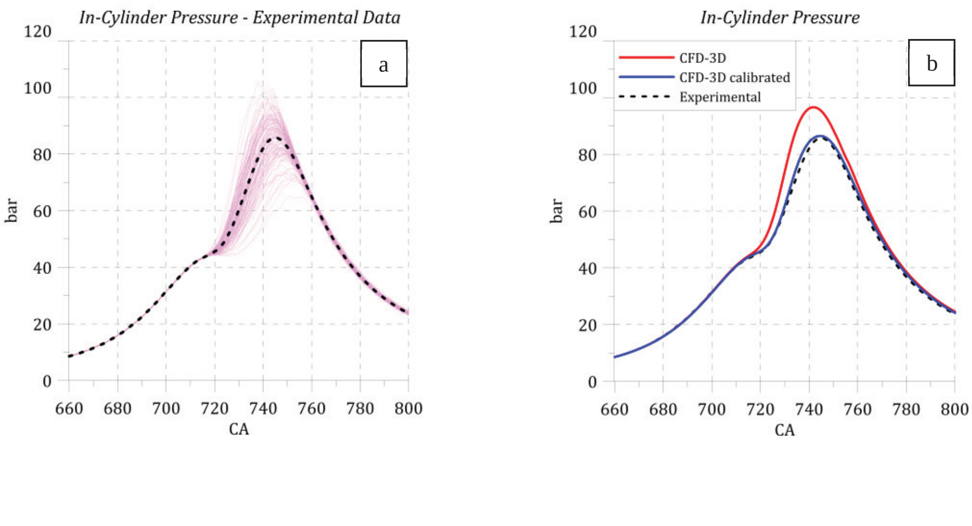


Fig. 2. Combined effect of combustion duration extension and reduction in AI times for various EGR rates and Equivalence Ratios.

6. 3-D Results

6.1 Model Calibration with Experimental Data

The CFD model is preliminarily calibrated to match the experimentally measured in-cylinder average pressure traces. SA is set close to the experimental one (to account for the arc delay),while fuel injected mass, start of injection angle (SOI) and valve lift are perfectly matching the actual operation. Three subsequent engine cycles are simulated in order to reach a converged solution. Experimental curves of in-cylinder pressure are presented together with their average in Fig. 3 (a) below. The average one is then compared to two CFD results, the first-cycle / non-calibrated calculation and the third-cycle calibrated one respectively. As visible, the non-calibrated cycle shows much higher pressure levels in the combustion chamber, while the calibrated engine cycle shows a much more consistent pressure trace. This is reflected in the auto-ignition heat release traces visible in Fig. 3 (c), where the non-calibrated cycle undergoes knock around 750°CA, while the peak value of the calibrated cycle is lower than that due to the spark, and its integral value (i.e. the heat released by autoignition) is negligible compared to the total heat released by combustion. This proves that the investigated engine operation can be considered at the edge of knock.



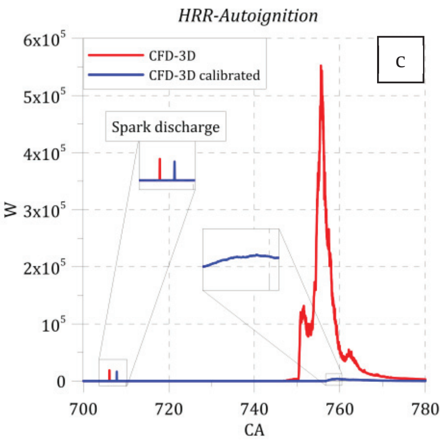


Fig. 3. (a) Ensemble average (black line) of in-cylinder pressure traces; (b) Comparison between CFD pressure trace (model not calibrated), CFD pressure trace (model calibrated) and experimental data; (c) Heat Release Rate for Autoignition

6.2 Water injection optimization

Due to the presence of many mutually-interacting phenomena, a preliminary optimization is carried out for the water injection process. A PFI single-hole water injector with a diameter of 0.15 mm is added to the CFD model. The injection pressure is set to approx. 5 bar. The injector position is investigated by comparing two mounting locations: close to the intake port junction (one water injector per cylinder, hereafter referred to as "I") and close to the intake valve (two water injectors per cylinder, hereafter referred to as "II"). Both locations are on the top side of the port, to allow easier mounting, as visible in Fig. 4.

Water injection phasing is also analysed. Several SOI values are chosen, ranging from 410° CA (so that water injection falls well-within the main intake process) to 250°CA (almost 100°CA before IVO). Each simulation is repeated for the two injector locations, in order to create a matrix of CFD cases whose aim is that of simultaneously maximizing the trapped water amount and the evaporation of the water droplets in the combustion chamber, to lower the charge temperature before the start of combustion.

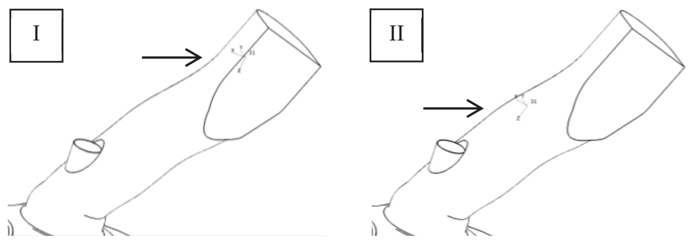


Fig. 4. Positions (I and II) of the injector.

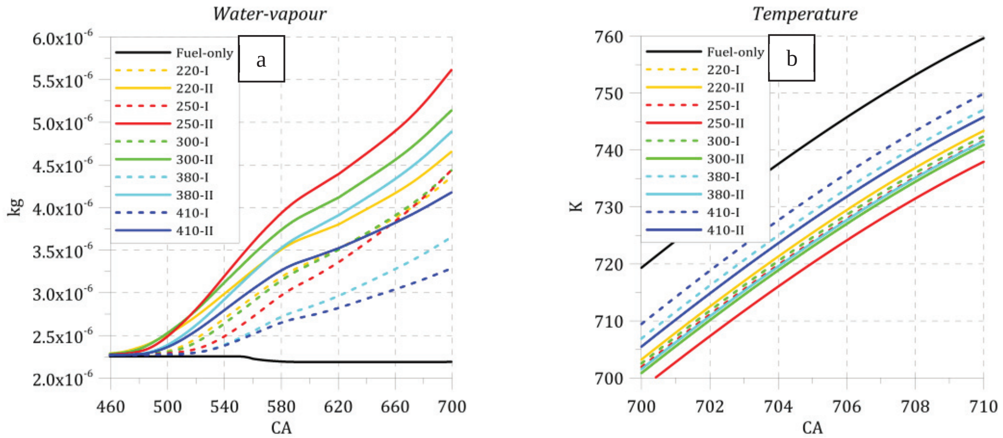


Fig. 5. (a) Evaporated water in the cylinder; (b) In-cylinder temperature.

Fig. 5 (a) above demonstrates that the amount of trapped water increases for the “II” position. Concerning phasing, 250°CA seems to be the best solution, since trapped water decreases for more advanced and more delayed SOI values. Case named 250-II provides therefore the greatest amount of vaporized water in the chamber, thus allowing the charge to reach the lowest temperature at the end of the compression stroke.
Thanks to the reduction of the injected fuel request, fuel injection timing is also investigated by comparing two phasing: the first keeping the same SOI, while the second keeping the same EOI as the original case. λ distributions at the spark time show that the equal EOI case is characterized by a better fuel stratification, resulting in leaner end gases (to increase knock resistance) and slightly richer λ near the spark plug (to promote flame kernel development). It also results in reduced fuel backflow in the intake port.

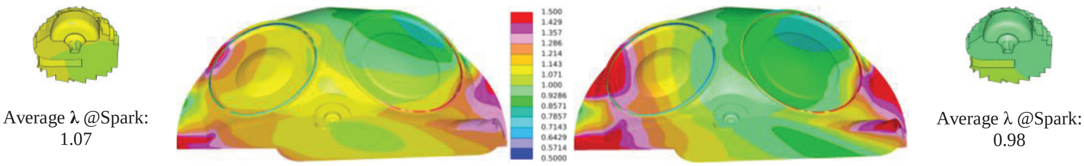


Fig. 6. λ @ SA: same SOI (left) and same EOI (right).

6.3 Combustion and knock tendency

The optimized water/fuel combination is then deeply investigated to evaluate the effect of water injection on knock. The original fuel-only case is used as a reference. As visible from Fig. 7 below, the charge temperature in the combustion chamber at the spark time is lower for the water injection case. Combustion is slightly slowed down by the combination of lower temperatures and higher residual gas concentrations. This results in a decrease of the in-cylinder pressure peak, visible in Fig. 8 (a), and in a simultaneous drop of the end-gas reactivity, since for the water case there is no signal of any autoignition heat release, Fig. 8(b). A visual confirmation is reported in Fig. 9, where the flame front distribution at 740°CA is reported.

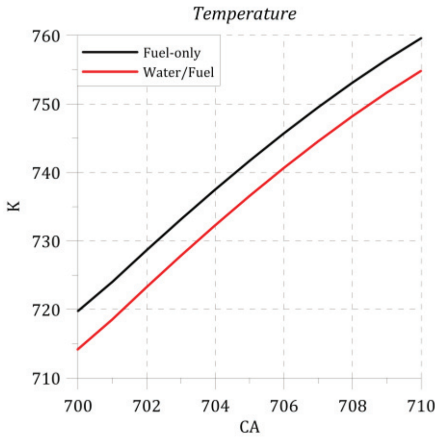


Fig. 7. in-cylinder temperature.

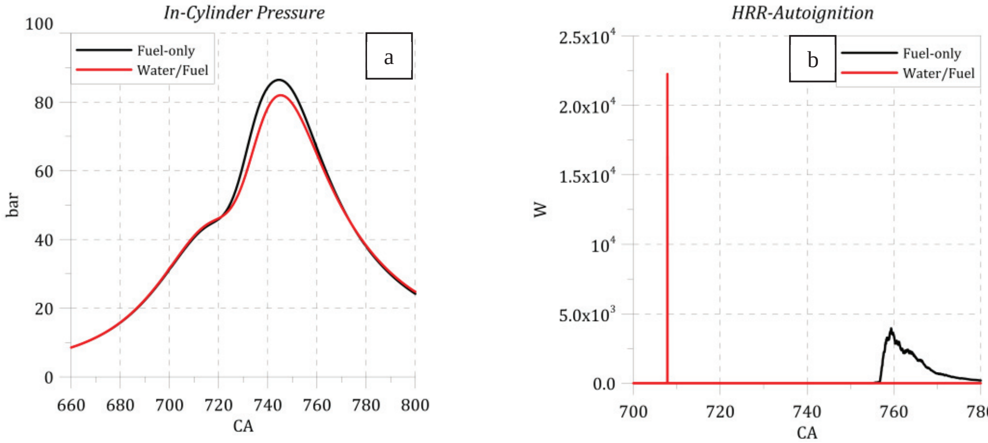


Fig. 8. (a) In-cylinder pressure traces; (b) Autoignition Heat Release Rate.


Fig. 9. flame front @ 740°CA: fuel only (left) and water/fuel (right).

The reduced end-gas reactivity is confirmed by the analysis of the knock tolerance function in the periphery of the combustion chamber, visible in Fig. 10 below. A 2 mm thick cell layer bounding the combustion chamber is reported: as visible, the knock tolerance function defined earlier exhibits higher values in the water/fuel case than in the fuel only one.

The decrease of IMEP of about 1.8% can be compensated thanks to the increased safety margin to knock onset. In fact, for the water/fuel case it is possible to increase the SA in order to recover the in-cylinder pressure trace and so engine performance. Advanced spark timings are tested, ranging from 2°CA to 7°CA, in order to find the trade-off between IMEP and knock tendency. At the end of the simulations, SAs greater than 4° CA are excluded because knocking becomes critical, as Fig. 11(a) shows. The case with 3° CA SA can be considered knock limited, see Fig. 11(b), and almost equivalent to the original fuel-only case, since for both cases autoignition heat release rate is comparable to that of the spark discharge. Still, the advanced spark allows the combustion to produce a higher pressure peak, see Fig. 11(c) and an increase of about 2% in the indicated power, despite the reduction of almost 17% in the overall injected fuel.

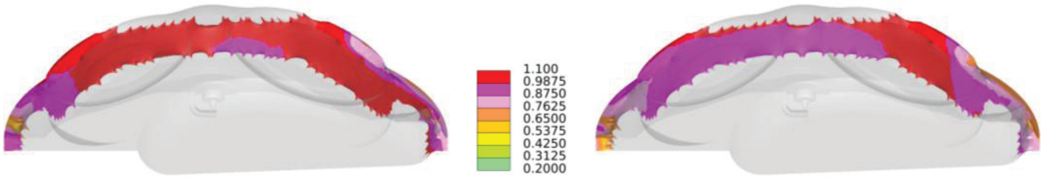


Fig. 10. Knock tolerance at 736° CA: on the left the case with water injection, on the right the case with fuel alone.

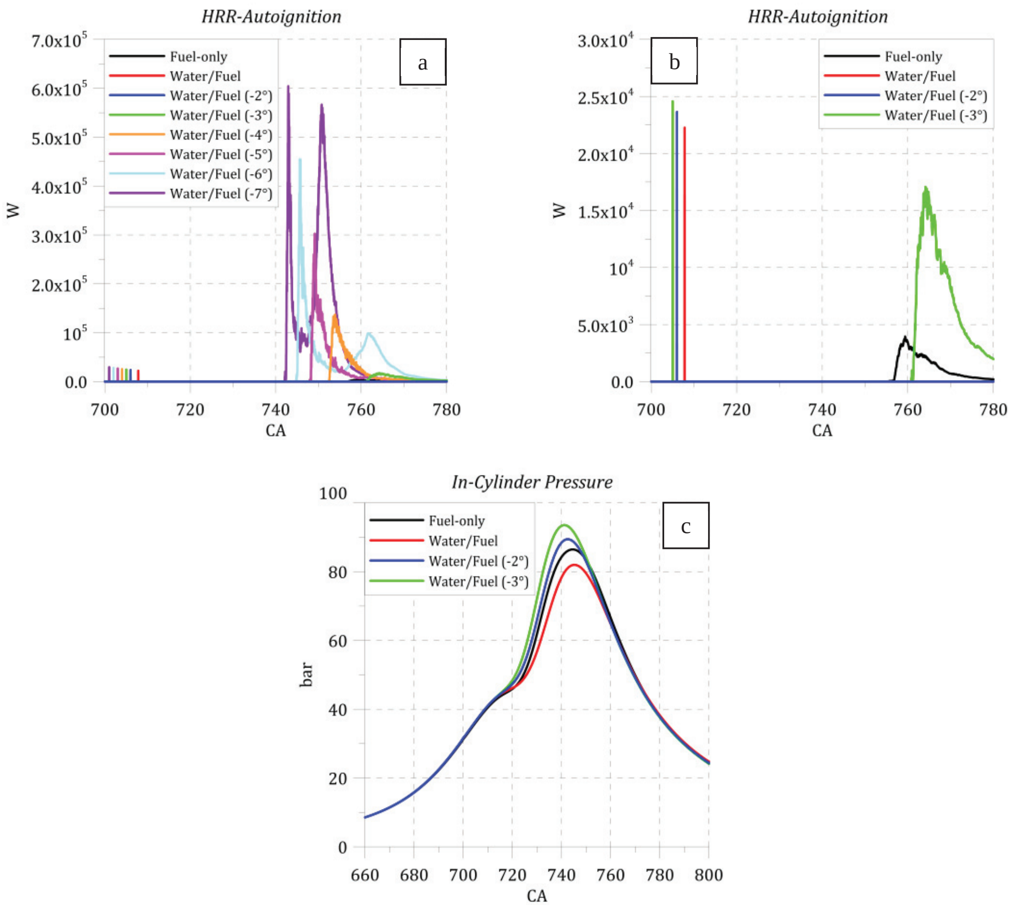


Fig. 11. (a), (b) Heat Release Rate for Autoignition; (c) Comparison between in-cylinder pressure traces.

7. Conclusions

Port water injection to limit fuel-enrichment in a high performance turbocharged GDI engine is investigated in the paper. Despite the increase in engine complexity caused by the introduction of the water injection system, benefits are found in terms of combustion efficiency increase and related reduction in brake specific fuel consumption. In particular, the water/fuel operation is optimized thanks to 3D-CFD analyses and a case is found which is characterized by a knock tendency comparable to the original fuel-only engine, an increase of indicated power of about 2% and a reduction of the injected fuel around 17%, thus leading to an overall reduction of brake specific fuel consumption of nearly 20%.

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