

Experimental and Numerical Study of the Water Injection to Improve the Fuel Economy of a Small Size Turbocharged SI Engine

Vincenzo De Bellis
University of Naples - Federico II

Fabio Bozza and Luigi Teodosio
University of Naples

Gerardo Valentino
Istituto Motori, CNR

ABSTRACT

In this work, a promising technique, consisting of a liquid Water Injection (WI) at the intake ports, is investigated to overcome over-fueling and delayed combustions typical of downsized boosted engines, operating at high loads. In a first stage, experimental tests are carried out in a spark-ignition twin-cylinder turbocharged engine at a fixed rotational speed and medium-high loads. In particular, a spark timing and a water-to-fuel ratio sweep are both specified, to analyze the WI capability in increasing the knock-limited spark advance.

In a second stage, the considered engine is schematized in a 1D framework. The model, developed in the GT-Power™ environment, includes user defined procedures for the description of combustion and knock phenomena. Computed results are compared with collected data for all the considered operating conditions, in terms of average performance parameters, in-cylinder pressure cycles, burn rate profiles, and knock propensity, as well.

Finally, the validated model is applied to investigate the full potential of water injection in reducing the knock tendency and improving the fuel economy in a wide load range. The numerical results highlight that WI technique involves significant Brake Specific Fuel Consumption (BSFC) advantages, especially at the medium-high loads. These benefits are limited by the maximum allowable levels for the in-cylinder pressure, while additional advantages are obtained in terms of reduced turbine inlet temperature, turbocharger speed, and boost pressure.

The developed numerical procedure, based on validated combustion and knock sub-models, is able to take into account the complex interactions among different parameters, which affect the engine behavior. It is hence believed to realistically forecast the WI-related BSFC advantages and constraints, induced by thermo-mechanical stresses. Simultaneously, it underlines the need of a partial engine redesign to fully exploit WI potential.

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INTRODUCTION

The current regulations oblige the car manufacturers to explore innovative, sometimes very complex, solutions to limit pollutants and CO₂ emissions of internal combustion engines [1]. On the other hand, the customers require improving performance and, in general, "fun-to-drive". In the case of Spark Ignition (SI) engines, a well-assessed technique to attain the above objectives is the so-called downsizing [2] coupled to turbocharging. The latter, in fact, allows decreasing the Brake Specific Fuel Consumption (BSFC) at part loads, thanks to a reduced throttling of the intake system. On the contrary, the boost level required to reach the prescribed high-load performance promotes the onset of knock or even pre-ignition

phenomena [3, 4, 5]. Both have to be avoided to preserve the engine safety. Knock occurrence, especially at low speeds, is commonly prevented by moving the combustion phasing late during the expansion stroke [3]. This choice, while penalizing the thermodynamic efficiency [4], involves an increased exhaust gas Temperature at the Turbine Inlet (TIT). This may cause thermal and structural problems for the turbine wheel and, for this reason, an enrichment of the Air-to-Fuel (A/F) ratio is usually applied at high speeds, with further BSFC penalizations [5]. Knock mitigation hence represents one of major prerequisite to attain a better fuel economy at high loads.

The above need will be even more stringent in the near future, since more severe driving cycles are going to be imposed to manufacturers for vehicle homologation, such as WLTC. The latter, in fact, involve a greater impact of medium-high load engine operation on the total CO₂ emissions. This further constraint determines the demand of a reduced fuel consumption also for the above operating conditions.

As well known, a lower knock propensity may be obtained if enhanced turbulence levels are realized inside the cylinder, since a faster burning process takes place [6,7]. A lower intake air temperature and/or cylinder wall temperature also play a positive role in knock avoidance [6,7]. Additional advantages can be gained through the employment of alternative fuels with higher octane number [7, 8, 9, 10, 11, 12, 13] or standard fuels with anti-knock additives [14]. The adoption of an external cooled Exhaust Gas Recirculation (EGR) circuit for knock avoidance has been also analyzed in a number of papers [15, 16, 17, 18, 19, 20, 21]. The lower gas temperature inside the combustion chamber, due to introduction of a large amount of inert, is the main driving parameter promoting the higher knock resistance. This technique, however, may induce a higher cyclic variability and a lower power output. Simultaneously, fluid-dynamic and thermal inertia of the EGR circuit pose control problems during fast transient maneuvers.

More recently, a greater attention has been devoted to water injection techniques, either within the cylinder or at the intake ports [22, 23, 24]. In the latter case, it can be applied to an existing engine with minor architecture modifications, and employing components of well-assessed technology, such as low-pressure injectors. The water evaporation heat, subtracted to the intake or trapped charge, causes a temperature drop during the compression stroke [22], hence substantially reducing the knock tendency. The above effect is enhanced by the increased heat capacity of the charge, due to the higher content of inert in the combustion chamber.

In the current literature, several papers have shown, from an experimental point of view, the potential of the water injection (WI) in terms of knock mitigation at high loads. Soyelmez et al. [23] studied its effects on the performance of a naturally aspirated 4-cylinder LPG SI engine at a constant throttle valve setting. In particular, five different Water-to-Fuel (W/F) ratios, from 0 up to 0.5, at various engine speeds, ranging from 1000 up to 4500 rpm, were considered. The spark timing was always set at the maximum brake torque (MBT) value. The results pointed out that engine torque, power and brake thermal efficiency improved as the W/F ratio increased, attaining maximum benefits at the highest investigated W/F ratio. The results also showed that the BSFC and the exhaust gas temperature decreased as the W/F ratio increased. In [24], Busuttill and Farruggia investigated the water injection effects with reference to a single cylinder SI engine for various spark timings, engine speeds and pressure levels in the intake manifold. The proposed results showed maximum torque improvement of 16 %, with a fuel-to-water ratio of 4.65 by volume. In [22], a single cylinder research engine equipped with two direct water-injectors was tested. A potential for efficiency increase of 3.3% – 3.8% was found in the region of the minimum BSFC, through a combination of water

injection, Miller cycle and cooled external EGR. More significant efficiency improvements, around 16%, were obtained at low speed and high load, mainly related to a combustion phasing optimization.

Concerning the study of water injection from a numerical point of view, very few papers are indeed available in the current literature. In [25], a high performance turbocharged engine was analyzed by a 3D approach, where the water was port injected to replace mixture enrichment, while preserving the indicated mean effective pressure and the knock safety margin. The results highlighted that WI allowed the spark advance (SA) to be increased, while maintaining the load target with a simultaneous relevant reduction in the BSFC. Boretti [26] analyzed by a 1D model the effects of WI for a turbocharged SI engine fueled with pure ethanol. Numerical findings showed that the port injection of liquid water involved the possibility to apply higher compression ratios, boost levels, and spark advances closer to the MBT. Both combustion and knock phenomena were modelled by simplified approaches (Wiebe-like function and Douaud and Eyzat model [27], respectively), which do not directly take into account the influence of operating parameters, such as the A/F ratio and the inert content.

Based on the above literature overview, 3D models proved to be very useful in better understanding and in predicting with higher accuracy the effects of water on the in-cylinder phenomena (inert and fuel spatial distribution, flame front distortion, etc.) [28,29]. On the other hand, in order to foresee the actual advantages of the discussed techniques, which depend on many different control parameters (valve strategy, boost pressure, water amount, etc.), a 1D model of the whole engine appears to be the most suitable numerical tool. In this case, however, more refined combustion and knock models with respect to those employed in the previously recalled literature are mandatory. The water introduction actually exerts thermal and chemical effects on combustion and auto-ignition phenomena, since they modify in-cylinder temperature and composition, as well. In addition, especially in a turbocharged engine, the WI substantially changes the boost pressure, airflow rate, TIT and hence relevant adjustments of spark advance, A/F ratio and waste-gate opening are expected in a prefixed torque-speed operating point. The way the above effects and control parameters modify combustion development, maximum in-cylinder pressure and temperature, and knock occurrence has to be hence directly included in the phenomenological combustion and knock models. Only in this way, a preliminary engine calibration with an enhanced knock resistance can be numerically identified, together with realistically expected BSFC improvements.

The main topic of the present work is the experimental and 1D numerical investigation of the water injection as a path to improve the fuel economy of a modern downsized engine at medium-high loads under knock limited operations. The study regards a PFI downsized twin-cylinder turbocharged VVA engine. The paper is organized as follows: preliminarily, the experimental setup and the considered operating conditions are illustrated. Then, the 1D engine model is described and validated against the measured data. Finally, a numerical calibration under knock limited operation, both with and without water injection, is discussed, highlighting the BSFC advantages and limitations of the WI technique.

EXPERIMENTAL SETUP AND TESTS

Main characteristics of the engine under investigation are reported in Table 1. Engine operating parameters, such as torque, fuel rate and air-to-fuel ratio are acquired at the test bench. In addition, each cylinder is equipped with a piezo-quartz pressure transducer (accuracy of $\pm 0.1\%$) to detect in-cylinder pressure signal. Furthermore, the instantaneous pressure traces at compressor outlet, intake plenum and turbine inlet are measured through piezo-resistive low pressure indicating sensors. Instantaneous in-cylinder pressure signals are acquired for 270 consecutive cycles and an automatic post-processing tool computes the related ensemble averages. In particular, the average pressure cycles are post-processed by a thermodynamic model to derive the burn rate profiles. More details on engine experimental setup and procedures may be found in [30].

Table 1. Engine main characteristics

Twin-cylinder PFI Engine	
Displaced volume, cm ³	875.4
Layout	4 valve/cyl
Bore, mm	80.5
Stroke, mm	86
Connecting rod length, mm	136.85
Compression ratio	10:1
Maximum brake power, kW	63.7@5500 rpm
Maximum brake torque, Nm	146.7@2000 rpm

The investigation is carried out at medium-high load engine condition (15-20 bar IMEP) and at a speed of 3500 rpm, for various spark timings and an assigned level of the relative air-to-fuel ratio, $\lambda = (A/F)/(A/F)_{st}$. The latter corresponds to the standard ECU-stored value of 0.87. The throttle valve is wide open (WOT), while the turbocharger setting is automatically set by the ECU in each condition. A water-to-fuel ratio sweep is realized in the range 0-0.3, with step 0.1 at each selected SA. A prototype driver, managed by a software tool developed within LabViewTM environment, directly controls spark advance and timing / duration of fuel and water injections. On the other hand, the standard ECU, based on manufacturer-developed logics, controls valve strategies and turbocharger waste-gate setting.

Nineteen operating points are collected on overall. Figure 1 depicts, under the form of a bubble chart, the investigated test matrix in terms of spark timing and W/F ratio. In the above figure, the circle diameter is proportional to the load level. Knock occurrence is monitored by a knock sensor installed on the commercial engine, which automatically cuts out the power output and avoid engine operation in the shaded region of Figure 1. The latter indicates, as expected, that the Knock Limited Spark Advance (KLSA) increases with water amount.

Additional parameters are also monitored to preserve the safety of the turbocharger and of the whole engine. In particular, the boost pressure is controlled to provide a proper operation for the injectors. The maximum in-cylinder pressure is constrained to limit mechanical stresses. The mechanical and thermal safety of the turbine moreover obliges to control the turbocharger speed and the turbine inlet temperature. Summarizing, the following constraints are applied:

- Max. turbocharger speed: 260'000 rpm
- Max. boost pressure: 2.6 bar

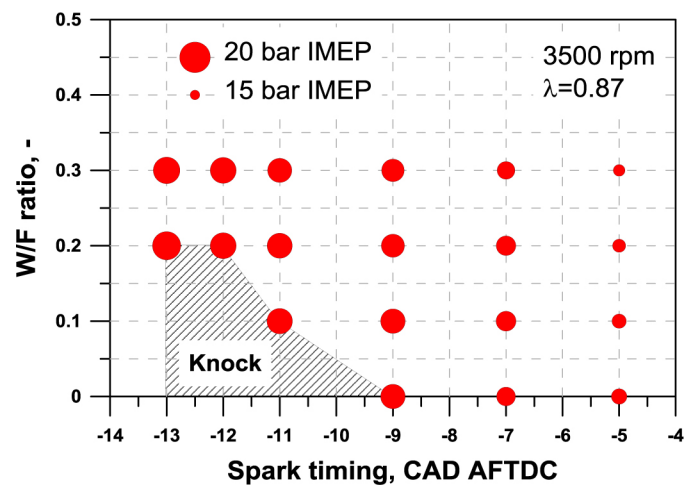


Figure 1. Tested operating conditions

MODEL DESCRIPTION AND VALIDATION

The tested engine is schematized in detail in the GT-Power environment. The experimentally actuated A/F, spark advance and valve strategy are imposed as input data in the simulations. According to the experimental campaign, the throttle is fixed at WOT, while the waste-gate setting is adjusted by a PID controller to match the measured boost level.

The water is introduced in the engine model by injectors located at the end of each intake runner, just upstream the standard fuel injectors. A discontinuous injection of liquid water at a temperature of 25 °C is specified. It is assumed that 20% of total mass of water vaporizes immediately upon the injection. This value is similar to the one advised for a typical fuel injector in the GT-Power user manual. A parametric analysis, not reported here for sake of brevity, has shown that substantial variations in the above percentage do not significantly affect the predicted results. Water condensation and evaporation are disabled within the intake pipes. In-cylinder water evaporation rate is indeed described by a semi-empirical correlation, formally resembling the fuel evaporation rate implemented in GT-Power. A certain reduction (about 20%) is however applied, to take into account the lower water vapor pressure. With this assumption, water evaporation begins during the compression stroke, and completes before the spark event. Although currently no data are available to verify the reliability of this assumption, it is important to stress that, in the experimental campaign, problems such as oil dilution, misfire, or partial combustion were not detected. This consideration suggests that the injected water completely evaporates in the real engine. Of course, this issue will be more deeply addressed in the future development of this activity, by means of a dedicated water evaporation sub-model.

- Max. TIT: 950 °C
- Max. in-cylinder peak pressure: 80 bar

The combustion process is modeled through a two-zone “fractal approach” [30], where the burn rate is computed as:

$$\frac{dm_b}{dt} = \rho_u A_T S_L \quad (1)$$

ρ_u being the unburned gas density, A_T the area of the turbulent flame front and S_L the laminar flame speed. The latter is evaluated according to the thermodynamic state, equivalence ratio and molar residual fraction in the unburned mixture [32], as reported in eq. (2).

$$S_L = S_{L0} \left(\frac{T}{T_{ref}} \right)^\alpha \left(\frac{p}{p_{ref}} \right)^\beta \exp(-4.5 c_f \tilde{x}_r) \quad (2)$$

Based on the fractal geometry theory, the turbulent flame area, A_T , increases with respect to the laminar one, A_L , as described by eq. (3):

$$\frac{A_T}{A_L} = \left(\frac{L_{max}}{L_{min}} \right)^{D_3-2} \quad (3)$$

L_{max} and L_{min} being the length scale of the maximum and minimum flame wrinkling, respectively, and D_3 the fractal dimension. For A_L estimation, the flame front is assumed as a smooth spherically-shaped surface and its area is derived by an automatic procedure implemented in a CAD software, processing the actual 3D geometry of the combustion chamber. The estimation of the above quantities, namely L_{max} , L_{min} and D_3 , is based on a turbulence sub-model, extensively reported in [33]. The above paper also illustrates the procedure for turbulence model tuning, aiming to fit the mass-averaged 3D-CFD-derived mean flow velocity and turbulent intensity inside the cylinder. Constants identified in [33] are also applied in the present analyses.

In the combustion model, the effect of the water presence on the burning rate is mainly demanded to the laminar flame speed variation, eq. (2), since it depends on the overall molar inert fraction, $\tilde{x}_r$. Of course, with this formulation, the influence of water is assumed to act in the same way as any other inert specie. This aspect will be refined in the future development of this work, basing on a numerical evaluation of the laminar flame speed, directly taking into account chemical effects induced by water. At the moment, indeed, flame speed is simply adjusted by a tuning constant, c_f , according to experimental observations. In particular, c_f is kept unchanged at 1.7, to get the best agreement with the measured burning rates, both for dry and water injection operation. Two additional tuning constants are introduced in the combustion model, affecting the transition between laminar and turbulent flame propagation and the maximum wrinkling scale [34]. They are selected in order to fit the in-cylinder pressure cycles, and then kept unchanged for all the considered operating conditions.

Concerning the knock modeling, a kinetic scheme for the oxidation of a Toluene Reference Fuel (TRF) is solved at each time step in the unburned gas zone. Actual gasoline is hence treated as a three-

component fuel, namely iso-octane, n-eptane and toluene, whose volume fractions are defined to get the same RON and MON of the commercial unleaded gasoline utilized in the experimental campaign [35]. The adopted kinetic scheme was developed by Liu et al. [36] and includes 5 elements, 56 species and 168 reactions. The mechanism handles both low and high temperature reactions and is built to well reproduce the auto-ignition delay and the temporal evolution of most important species. Amount and composition of inert gases in the combustion chamber hence affect the TRF kinetics and hopefully capture both the thermal and chemical effects exerted by the presence of water vapor in the unburned mixture, better than a simple correlation [27,37]. Knock event is recognized as a sudden jump in the end-gas temperature, due to auto-ignition occurrence. The knock index, Δp_{knock} , is computed as the pressure increase occurring in an isochoric combustion of the unburned fraction at the knock event, $x_{u,knock}$ according to [38,39]:

$$\Delta p_{knock} = \frac{R}{c_v} \frac{x_{u,knock} m_f LHV}{V_{knock}} \quad (7)$$

R and c_v being the average gas constant and the constant volume specific heat of the in-cylinder gases, m_f and LHV the mass and the lower heating value of the fuel, and V_{knock} the in-cylinder volume at the knock event.

Results shown in Figure 2–Figure 5 are obtained after the tuning of the combustion model. With reference to the cylinder #1, Figure 2–Figure 4 show the numerical/experimental comparison in terms of IMEP, maximum in-cylinder pressure, and ISFC, respectively. Similar results are obtained for cylinder #2, but, for sake of brevity, they are not shown here. It can be noted that IMEP and in-cylinder peak pressure are satisfactorily predicted (average error of 2.13 and 2.26%, respectively), denoting the good accuracy in both engine geometrical schematization and combustion description. Moreover, the ISFC shows a reduced average error (3.46% - Figure 4), with most of the points included in the error band $\pm 5\%$. On the contrary, the TIT prediction (Figure 5) is affected by a systematic overestimation of about 50°C. The above disagreement is probably due to an inadequate modelling of the heat transfer inside the cylinder and for the exhaust pipes, starting from the exhaust valves up to the turbine inlet. In addition, the measured values may be also affected by a certain inaccuracy, related to the thermocouple heat exchange. Although a more refined tuning of heat transfer multipliers in the above components would be possible, this is beyond the aim of the proposed work. TIT errors however are expected to not significantly affect the judgement of numerical results presented in the next paragraph, since the same heat transfer modeling is applied in each case, preserving the consistency of the comparison among different operating conditions.

The model reliability is further demonstrated by the agreement of experimental/numerical pumping loops of Figure 6. The latter refers to the operating point without water injection, and spark timing of -7 CAD AFTDC. Pressure fluctuations are well captured by the model, both in terms of shape and pulse amplitude. Some inaccuracies occur in the reproduction of the exhaust pulse ahead the turbine inlet,