

Extension and Validation of a 1D Model Applied to the Analysis of a Water Injected Turbocharged Spark Ignited Engine at High Loads and over a WLTP Driving Cycle

Fabio Bozza
Univeristy of Naples

Vincenzo De Bellis
Federico Ii

Pietro Giannattasio
University of Udine

Luigi Teodosio
Univeristy of Naples

Luca Marchitto
Istituto Motori CNR

ABSTRACT

The technique of liquid Water Injection (WI) at the intake port of downsized boosted SI engines is a promising solution to improve the knock resistance at high loads. In this work, an existing 1D engine model has been extended to improve its ability to simulate the effects of the water injection on the flame propagation speed and knock onset. The new features of the 1D model include an improved treatment of the heat subtracted by the water evaporation, a newly developed correlation for the laminar flame speed, explicitly considering the amount of water in the unburned mixture, and a more detailed kinetic mechanism to predict the auto-ignition characteristics of fuel/air/water mixture. The extended 1D model is validated against experimental data collected at different engine speeds and loads, including knock-limited operation, for a twin-cylinder turbocharged SI engine. The model predictions are compared with the experimental data, in terms of in-cylinder pressure cycle, burn rate profile and knock propensity. The numerical model correctly reproduces the experimental findings of fuel consumption, turbine inlet temperature and in-cylinder peak pressure. The combustion process, both with and without water addition, is predicted quite well, except for some inaccuracies in the early stage of combustion. Both experimental and numerical data confirm that the WI technology is able to improve significantly the fuel consumption of the tested engine under high-load knock-limited operation. Main drivers of the above advantages are a reduced over-fueling and a better combustion phasing. In a second stage, the validated model is used to build-up a complete engine operating map aimed at investigating the potential of WI technique to improve the fuel economy of a segment A vehicle. Engine maps with and without WI are introduced in a vehicle model to estimate the grams of CO₂ per kilometer over a WLTP driving cycle. A reduced impact of WI is observed, since a knock-free operation occurs along most of the WLTP cycle. Nevertheless, some limited benefits can be still appreciated.

CITATION: Bozza, F., De Bellis, V., Giannattasio, P., Teodosio, L. et al., "Extension and Validation of a 1D Model Applied to the Analysis of a Water Injected Turbocharged Spark Ignited Engine at High Loads and over a WLTP Driving Cycle," *SAE Int. J. Engines* 10(4):2017, doi:10.4271/2017-24-0014.

INTRODUCTION

Nowadays, car manufacturers are continually committed to find proper technical solutions to meet the CO₂ emission target. For instance, a target of 95 grams of CO₂ per kilometer, evaluated as the mean of the total fleet of vehicles sold, has been scheduled to be gradually achieved by 2020 in the European Union [1,2]. Various techniques have been proposed to meet this target, such as a

completely flexible Variable Valve Actuation (VVA) system [3, 4, 5], Variable Compression Ratio (VCR) [6-7], cylinder deactivation [8], cooled Exhaust Gas Recirculation (EGR) [9, 10, 11, 12, 13, 14, 15] and Water Injection (WI) [16, 17, 18, 19, 20, 21, 22, 23]. All the above approaches showed a certain improvement in terms of Brake Specific Fuel Consumption (BSFC) and, hence, CO₂ emissions. However, the extent of these benefits strictly varies according to the

engine operating condition: for instance, VVA and cylinder deactivation mainly impact at part load, in terms of pumping work reduction, while VCR and WI mainly act at high loads, in terms of knock mitigation.

The related benefits on CO₂ emissions are quantified on type approval laboratory tests. The latter follow standard procedures put into force by local authorities. In the European Union, the New European Driving Cycle (NEDC) is currently used as the reference driving cycle, while the FTP-75 is considered in the USA. The World-wide harmonized Light vehicles Test Procedure (WLTP) and Cycle (WLTC) are under assessment, with the aim to state a global procedure and driving cycle to be followed world-wide. Along the homologation driving cycles, the engine more frequently operates at reduced load levels, and speeds usually below 3000 rpm. Hence, the effectiveness of the above listed technical solutions has to be verified with special reference to the cited operating conditions.

Each technical approach has pros and cons, in terms of architecture complexity, control and CO₂ reduction effectiveness. Among the other solutions, the injection of liquid water at the intake port or in the cylinder appears a promising technique to improve the engine fuel economy, especially at high load operation. This approach can be applied with minor modifications of the engine architecture, and, in the case of ported injection, components of well-assessed technology can be used, such as low-pressure injectors.

Because of heat subtracted by evaporation, the introduction of liquid water within the cylinders has a cooling effect on the fresh charge, increasing its density during the intake phase, and hence enhancing the volumetric efficiency. In addition, the injected water results in a greater mass of working gas during the expansion phase, and hence in an increased expansion work. On the other hand, the water directly affects the combustion process, generally causing a reduced burn rate. The evaporation-related cooling promotes a temperature drop during the compression stroke [16] that substantially contributes to mitigate the knock tendency. In addition, the temperature reduction is enhanced by the increased heat capacity of the charge, due to the larger content of inert in the cylinder.

The effectiveness of WI technique has been verified experimentally and numerically in some works available in the current literature [16, 17, 18, 19, 20, 21, 22, 23]. For instance, in [17] the improvements in the engine performance, in terms of brake torque (up to 2.4 %) and efficiency, have been proved experimentally for a naturally aspirated 4-cylinder LPG SI engine at various speeds and for a prefixed setting of the pedal position. The spark timing is always set at the Maximum Brake Torque (MBT). Torque enhancements up to 16% were measured in [18], with reference to a prototype single cylinder engine. In [16], efficiency gains of 3.3% - 3.8% were found in the region of the minimum BSFC. More significant advantages, around 16%, were obtained at low speed and high load, mainly promoted by a combustion phasing optimization allowed by the water-induced knock mitigation. Water injection effects on pollutant emissions were also addressed in [16, 19, 20, 21], showing that WI contributes to reduces NO_x, while penalizes CO and HC. Increased knock resistance and improved engine performance with WI were also numerically showed in [22] and [23], by 1D and 3D simulations, respectively. In

both analyses, the presence of the water in the charge allowed a re-phasing of the spark timing, in order to compensate the combustion speed slowdown and/or to move towards MBT operation thanks to the knock mitigation.

Although the effectiveness of the water injection has been proved both experimentally and numerically at high load, where engine performance drastically improves, its usefulness has still not quantified in terms of CO₂ emissions along a vehicle driving cycle. This information is indeed relevant at industrial level, in order to estimate the real potential of the WI technique in contributing to meet actual and future CO₂ emission targets. The passage from NEDC to more load-demanding driving cycles, such as WLTP or Artemis, imposes in fact a more frequent medium-high load operation of the engine, where knock may occur, and where WI is expected to provide higher benefits.

The usual quasi-steady approach to perform a driving cycle simulation requires the prediction of the whole engine operating plane. In absence of experimental data, a proper calibration of several control parameters, such as Spark Advance (SA), throttle opening, Waste-Gate (WG) opening, and Air-to-Fuel (A/F) ratio, is mandatory all over the operating plane. A proper strategy to provide a realistic numerical calibration of the engine with or without WI represents an onerous task to be accomplished. Different targets have in fact to be considered, such as maximum power output at full load and minimum fuel consumption at part-load. Moreover, many constraints must be met to guarantee engine and sub-components operation below prefixed levels of thermal and mechanical stresses.

Based on the above considerations, in the present work, a refined 1D model is employed to explore the potential CO₂ reduction of the WI technique over a WLTP cycle, with reference to a Ported Fuel Injection (PFI) downsized twin-cylinder turbocharged engine powering a small city-car of segment A. WI impact on WLPT emission of other noxious species is indeed not addressed in this work. Despite, in fact, engine-out emission varies, regulated species concentration downstream a stoichiometric operated catalytic converter is expected to still comply with the international legislations.

In a first stage, dry and water injected tests are performed to characterize the WI impact on the combustion and knock processes. These data are used to assess and validate a 1D engine model, provided with refined sub-models of combustion and knock. Then, an automatic procedure for engine calibration is set-up to calculate the engine performance over the whole operating plane, with and without water injection. Finally, the computed fuel maps are used in a vehicle simulation to quantify the potential CO₂ emission reduction of a WI operated engine.

EXPERIMENTAL SETUP AND TESTS

The main features of the engine under investigation are reported in Table 1, while a sketch of the experimental test bench is shown in Figure 1. Starting from the commercial engine, two low-pressure injectors (one for each cylinder) have been installed at each intake port to allow the water injection. 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 the in-cylinder pressure signal.

Furthermore, the instantaneous pressure traces at compressor outlet and intake plenum are measured through piezo-resistive low pressure indicating sensors. Instantaneous in-cylinder pressure signals are acquired for several consecutive cycles and an automatic post-processing tool computes the related ensemble averages. In particular, the average pressure cycles are post-processed by an inverse model to derive the burn rate profiles. More details on engine experimental setup and procedures can be found in [24].

Table 1. Engine main features

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

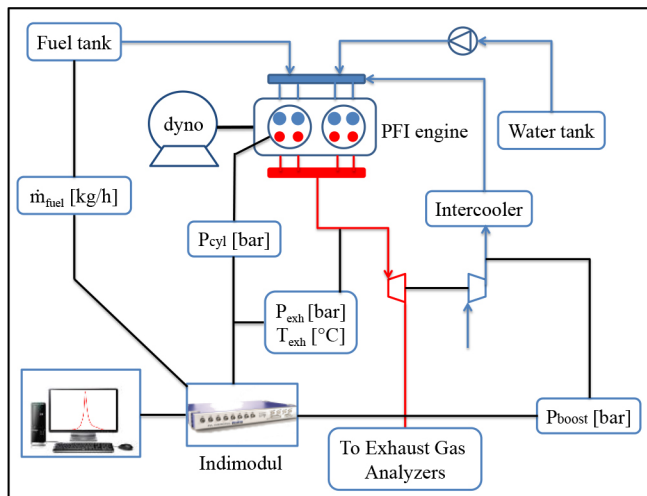


Figure 1. Sketch of the engine test bench

Preliminary measurements are performed over the whole engine operating plane in dry operation. Engine speed is varied from 1500 up to 5500 rpm, with a load sweep in the range 1.0 - 19.5 bar BMEP, summing up 149 operating points. The standard manufacturer calibration of each control parameter, namely the spark advance, the A/F ratio, and the WG opening, is employed.

During measurements, many parameters are monitored by the standard ECU to preserve the safety of the turbocharger and of the whole engine. In particular, the boost pressure is automatically controlled acting on the waste-gate valve to provide a proper operation for the injectors and to avoid compressor surge at low engine speed. The spark timing is controlled in closed-loop by a knock sensor to avoid abnormal combustions, and to limit maximum in-cylinder pressure. Moreover, the mechanical and thermal safety of the turbine, obliges controlling the Turbine Inlet Temperature (TIT). To this aim, injection duration is regulated to ensure a fuel

enrichment of mixture at high speed and load. The maximum delivery rate of the fuel injectors also defines a minimum A/F value.

Summarizing, the following constraints are indirectly derived from the above measurements:

- Max. TIT: 930 °C
- Max. in-cylinder peak pressure: 85 bar
- Max. boost pressure: 2.4 bar
- A/F ratio range: 10.5 - 14.5

Additional investigations, with and without WI, are carried out at high loads (WOT condition, 16.4-18.0 bar Indicated Mean Effective Pressure - IMEP) and speeds ranging from 2500 up to 4500 rpm. A prototype driver, managed by a software tool developed within LabView™ environment, is in this case employed to directly control spark advance and timing / duration of fuel and water injections. The waste-gate setting is indeed still controlled by the standard ECU. In order to set a reference condition (dry operation), the ECU-stored values of the relative air-to-fuel ratio, $\lambda = (A/F)/(A/F)_{st}$, are initially applied for each engine speed. Then, the SA is externally varied, up to knock onset. In this case, the ECU, which no longer controls the SA, activates a recovery mode that automatically cuts out the fuel injection. For each speed, starting from the base knock-limited point, WI is activated, and λ is firstly leaned up to the stoichiometric level. Then, the SA is increased once more, until knocking combustions are detected again.

Under WI operation, a Water-to-Fuel (W/F) mass fraction of 0.2 is assigned. This choice arises from the experimental observation that higher W/F ratios result in an unacceptable oil dilution in the engine crankcase, caused by a poor water atomization related to the employed injection pressure (4 bar). Such a reduced water amount also results in a limited increase of the combustion variability with respect to dry operation. The coefficient of variation of IMEP maintains below 4% for all the considered operating conditions, both with and without water injection. Hence, despite a certain cyclic variability rise, this issue does not represent a critical point for the considered engine operation.

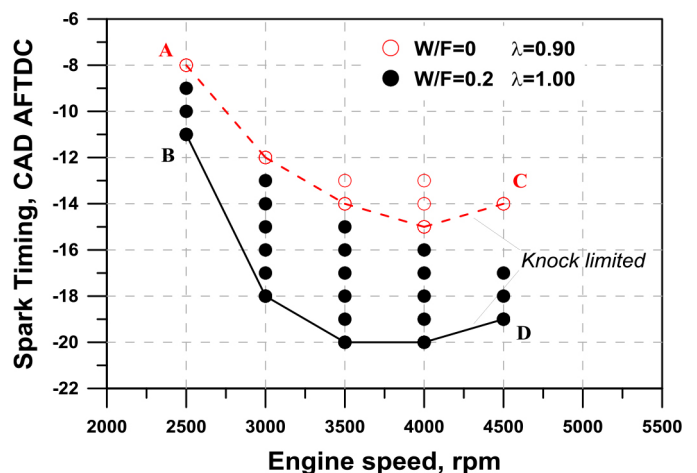


Figure 2. Tested operating conditions

Thirty-one operating points are collected. Figure 2 depicts the investigated test matrix in terms of engine speed and spark timing. Empty circles represent points in the reference calibration, where a rich A/F is considered ($\lambda=0.90\pm0.01$), while filled circles correspond to operation with WI and stoichiometric A/F ratio ($\lambda=1.00\pm0.01$). The Figure also shows the spark timing at knock borderline, namely the Knock Limited Spark Advance (KLSA), for each speed, connected by dashed and continuous lines, for dry and WI operation, respectively. However, the KLSA along these lines also depends on the boost level, which is slightly changing in each point. The water introduction allows advancing the spark timing from 3 CAD at 2500 rpm up to 6 CAD at 3500 rpm. At 3000 rpm, two additional operating conditions, not reported in the test matrix, are collected without WI, at $\lambda=1.0$, and SA of -13 and -14 CAD AFTDC, by setting the maximum boost level at knock borderline.

ENGINE MODEL DESCRIPTION

The tested engine is schematized in detail in the GT-Power environment. The experimentally actuated A/F ratio is imposed as input data in the simulations. According to the experimental evidence, a fuel metering imbalance between the two cylinders is also reproduced in the model, resulting in a greater amount of fuel injected, and a richer A/F ratio, in cylinder #2. The throttle valve is fixed at WOT, while the waste-gate setting is adjusted by a PID controller to match the measured boost level. The SA is automatically modified by another controller targeting the experimental angle of 50% Mass Fraction Burned (MFB₅₀).

Consistently with the experimental layout, the water is introduced in the engine model by injectors located at the end of each intake runner, just upstream of the standard fuel injectors. A discontinuous injection of liquid water at a temperature of 25 °C is specified. It is assumed that 30% of the total mass of water vaporizes immediately upon the injection. This value is equal to the one advised for a typical fuel injector in the GT-Power user manual. A parametric analysis, not reported here for brevity, showed that substantial variations in the above percentage do not significantly affect the simulation results. Due to the low temperature and high pressure, water evaporation is disabled in the intake pipes downstream of the injectors. Therefore, water evaporates only inside the cylinders, at a rate provided by a semi-empirical correlation. With this assumption, water evaporation begins during the intake stroke, and completes before the spark event. The water evaporation heat is assumed to be subtracted from the cylinder/piston walls and the working fluid to the same extent (50%). Although no data are currently available to verify the reliability of the previous assumptions, they appear reasonable. In the future development of the present work, they will be more deeply addressed by means of a dedicated water evaporation sub-model.

A refined model of the heat transfer inside the cylinder and exhaust pipes is also introduced, applying a wall temperature solver based on a finite element approach. Concerning the in-cylinder heat transfer, a Hohenberg-like correlation is used, while convective, radiative and

conductive heat transfer modes are considered for the exhaust pipes. The combustion process is modeled through a two-zone “fractal approach” [25], 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. Based on the fractal geometry theory, the ratio of the turbulent flame area, A_T , and the laminar one, A_L , varies according to the following equation:

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

where $L_{\max}$ and $L_{\min}$ are the length scales of the maximum and minimum flame wrinkling, respectively, and D_3 is the fractal dimension. For the estimation of A_L , the flame front is assumed as a smooth spherically-shaped surface and its area is computed by an automatic procedure implemented in a CAD software capable of processing the actual 3D geometry of the combustion chamber. The estimation of $L_{\max}$, $L_{\min}$ and D_3 is based on a turbulence sub-model extensively reported in [26].

The laminar flame speed, S_L , in eq. (1) is evaluated by a new general correlation, developed in the present work, that explicitly considers the water content in the unburnt mixture. This correlation is the result of the fitting process of laminar flame speed outcomes from a chemical kinetic solver (CHEMKIN) applied to the computation of 1D laminar flames. The actual gasoline is modelled by a Toluene Reference Fuel (TRF), composed of iso-octane, n-eptane and toluene. The volume fractions of the three components of the surrogate fuel are selected so as to match the RON and MON of the actual gasoline [27]. Different oxidation mechanisms for gasoline surrogates are available in the current literature, ranging from simplified [28] to semi-detailed [29] and detailed kinetics [30]. They are characterized by an increased number of species and reactions, and hence by growing computing times. Since the development of a quite general correlation requires a large number of flame speed data (more than 3000 evaluations), a compromise between accuracy and computational effort is mandatory. For this reason, it was decided to use the semi-detailed mechanism developed by Liu et al. [29], which handles both low and high temperature kinetics, and includes 5 elements, 56 species and 168 reactions. The computation of S_L was performed for several values of pressure, p , temperature, T , equivalence ratio, $\phi=1/\lambda$, RON, MON, EGR mass fraction, x_p , and water mass fraction, x_w , in the following ranges:

- Pressure, p : 1 - 10 bar
- Temperature, T : 323 - 473 K
- Equivalence ratio, ϕ : 0.7 - 1.6