

Investigation of Water Injection Effects on Combustion Characteristics of a GDI TC Engine

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ABSTRACT

This paper presents simulation and experimental results of the effects of intake water injection on the main combustion parameters of a turbo-charged, direct injection spark ignition engine. Water injection is more and more considered as a viable technology to further increase specific output power of modern spark ignition engines, enabling extreme downsizing concepts and the associated efficiency increase benefits. The paper initially presents the main results of a one-dimensional simulation analysis carried out to highlight the key parameters (injection position, water-to-fuel ratio and water temperature) and their effects on combustion (in-cylinder and exhaust temperature reduction and knock tendency suppression). The main results of such study have then been used to design and conduct preliminary experimental tests on a prototype direct-injection, turbocharged spark ignition engine, modified to incorporate a new multi-point water injection system in the intake runners. The experiments allowed to validate the model results, demonstrating the effectiveness of the proposed technology, and to further investigate on the mechanisms that allow controlling thermal load and knocking tendency by varying the water-to-fuel ratio.

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INTRODUCTION

High load conditions for SI engines, especially in downsized and boosted engines, are heavily limited in terms of achievable efficiency levels by two main issues: excessive exhaust temperatures and heavy knocking operation. Exhaust temperatures cannot exceed maximum tolerable inlet temperature of the turbine, determined by speed and used materials, and the common solution is to operate with rich mixture (exploiting heat subtraction due to fuel evaporation) waiving to after-treatment possibilities and heavily increasing fuel consumption.

The other main limit is imposed by knocking combustion, which occurs at high load conditions, due to high in-cylinder temperature and pressure levels. This forces to adopt a sub-optimum Spark Advance (SA) angle, thus reducing maximum temperature and knock intensity, while correspondingly increasing exhaust temperatures, with respect to maximum efficiency conditions.

Such strategies are no longer compatible with new homologation regulations and corresponding real driving cycles and consumption and emissions upper bonds, so much that a pre-treatment of the fresh mixture seems to be a promising solution.

Among several other possible solutions, water injection is a well-known technology, and many examples of its implementation can be found [1, 2, 3], to actively control knock tendency, while maximizing combustion efficiency. The principle consists in reducing charge and combustion temperature by means of the high latent heat of vaporization of injected liquid water. In this way, benefits on exhaust temperature and knock intensity reduction can be obtained.

In the next years, water injection will probably become a standard equipment for many production engines [4, 5].

This work investigates port water injection possibilities, since this layout results in no modification or complication of the cylinder head and can be easily installed on a pre-existing engine. The simplicity of this layout may compensate extra benefits and operation flexibility obtainable with direct water injection.

The investigation presented in this paper was carried out by 1-D modeling analysis in a first stage, then experimental tests were conducted in order to verify and validate the model. Even if experimental tests were limited to port water injection, possibilities given by a direct injector have also been investigated with the 1-D model.

WATER INJECTION AND KNOCK MODEL

To evaluate water injection effects on combustion, a 1-D model of the prototype engine used for the test (Table 1) has been developed in Gt-Power [6] environment. In this first stage, a single-cylinder model has been set, to reduce computational efforts.

Table 1. Engine specifications.

Displaced volume	1389.9 cc (4 cylinder)
Stroke	75.6 mm
Bore	76.5 mm
Connecting Rod	144 mm
Compression ratio	10:1
Number of Valves	4
Exhaust Valve Open	580° BTDC @ 0.1 mm lift
Exhaust Valve Close	356° BTDC @ 0.1 mm lift
Inlet Valve Open	358° BTDC @ 0.1 mm lift
Inlet Valve Close	132° BTDC @ 0.1 mm lift

Model Layout

The developed engine model is based on a 1-D representation, and only one cylinder, in this first stage, is considered. Moreover, components upstream the intake and downstream the exhaust manifolds have been neglected. Model parameters identification has been carried out through experimental data, without water injection, in several operating conditions, with standard, or production, engine calibration. Combustion simulation is governed by a predictive model, which should be sensitive to in-cylinder variations of the charge physical properties due to heat subtraction operated by water evaporation. Three-zones combustion model equations are default ones implemented in GT Power cylinder template [6].

Primary aim of modelling activity is to simulate a port water injection system, which is also the experimental configuration available on the test bench. To consider water film formation and evaporation dynamics in the intake ports, *PipePort* and *FlowSplitPort* templates (circled in green in Figure 1) have been used, which are designed to model port water injection [6]. Water injection within intake runners implies that a fraction of liquid mass vaporizes within ducts, and remaining mass enters the combustion chamber in liquid form. But model intake ducts thermal conditions cause evaporation of entire injected water mass, so configuration with a port water injection doesn't allow considering liquid fraction that evaporates within cylinder. In the simulations, ducts wall temperature depends on working conditions (manifold pressure and engine speed). The water introduced with port injection (circled in blue in Figure 1) forms a fluid film against intake ducts walls, and it entirely evaporates within runners. In particular, the fluid film evaporates during intake stroke, when liquid it achieves critical pressure and temperature conditions.

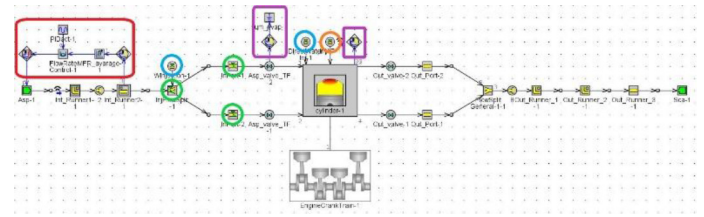


Figure 1. Gt-Power model layout. Intake, cylinder and exhaust sub-models are shown from left to right. Sensors that measure water mass toward valves and within cylinder are marked in violet.

To control the amount of liquid water entering the cylinder, a fictitious direct injector was added in the model (blue circle upon cylinder in Figure 1). This injector is set in order to vaporize injected water instantaneously during injection and in this way evaporation rate for the liquid mass fraction that reaches the combustion chamber can be controlled (i.e., imposed) by the direct injection rate. Direct injection starts with IVO because it is assumed that water which is still in liquid form in the intake ducts starts entering the cylinder as soon as the intake valves open. A PID controller calibrated with Ziegler-Nichols method (marked in red in Figure 1) has been introduced to ensure the same air mass flow rate within intake runners, managing manifold pressure. Figure 1 shows a schematic of the developed 1-D model.

Water Injection Modelling Parameters

The total amount of injected water is described by r , which is the ratio between injected water quantity and stoichiometric fuel mass. This convenient normalization is proposed in [1].

The split between port-injected and (fictitious) directly-injected water mass is described, and user-imposed, by F_i and F_d , which represent the intake-injected and directly-injected water mass fractions, respectively. By definition, the sum of these two parameters is 1. The *Start of Injection* (SOI) for the port injector has practically no effects due to the model intake dynamic characteristics. In fact, as mentioned above, evaporation of entire fluid wall film takes place during intake stroke. This is one of the main limits of this model, which cannot represent tridimensional interactions between evaporating water spray and pressure waves in the intake manifolds. In-cylinder injection start is imposed at IVO (Intake Valve Opening), since it should model port-injected liquid water entering the combustion chamber during the intake stroke. This hypothesis is not necessarily verified: depending on the port-injection timing, injected water may not enter the cylinder during the first part of the intake stroke. But model simulations reveal that this is the most unfavorable condition, and also the most probable one when port water injection is implemented.

Injection (and therefore evaporation) duration is imposed too and described by the parameter ToE: *Time of Evaporation*, which will be considered below in terms of equivalent crank angle interval (for now, evaporation rate, or equivalently direct injector mass flow, is supposed to be constant).

The split between in-cylinder and in-runner water evaporation (i.e. F_d and F_i) has to be identified and calibrated, as well as the parameter ToE , depending on injection phasing and engine operating conditions. Indeed, this modelling approach is relatively simple but it is not predictive.

A first sensitivity analysis to these parameters variations has been conducted using the above described model. In this way, parameters that effectively represent degrees of freedom of the model can be identified to fit experimental data. Variation of parameters F_d , F_i and r affects intake air mass flow rate, so during these simulations PID controller has been set on. Other parameters, which do not significantly affect the model outputs, due to the grey/black-box nature of this part of the model, or because of expected very weak physical correlations, will be kept constant.

Location of the water evaporation (where it takes place) has a severe impact on the useful effect of water injection or, equivalently, on water consumption. As shown in Figure 2, by varying the parameter F_i from 0 to 1 (corresponding to F_d variations from 1 to 0), the same water mass ($r = 0.5$, in this case) progressively produces minor impact on in-cylinder temperature. Figure 2 shows that temperature reduction achievable with direct injection is the highest. The engine operating point related to Figure 2 is defined by engine speed equal to 3000 rpm and intake manifold pressure equal to 1.5 bar, with a constant SA of 13° CA Before Top Dead Center (BTDC), but the results are qualitatively valid also for different operating conditions.

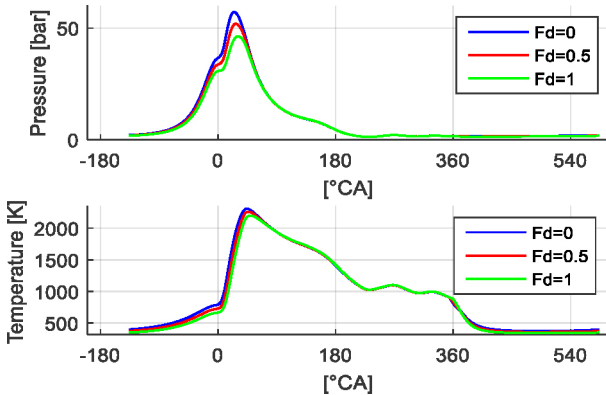


Figure 2. Effect of F_d variation for fixed water quantity. Top graph represents such effects on the in-cylinder pressure profile, while in the bottom graph the corresponding effect on the in-cylinder temperature is shown.

This means that direct injection is undoubtedly the best solution. These evidences confirm results reported in [1] and suggest, supposing a port injection solution, an injector installation as close as possible to the intake valves, while a single-point configuration is not recommended. Figure 3 clearly shows that also the angular position of the in-cylinder evaporation (i.e. by varying in-cylinder injection duration, since the SOI is bonded to IVO event) affects temperature reduction, for fixed r and F_i/F_d values.

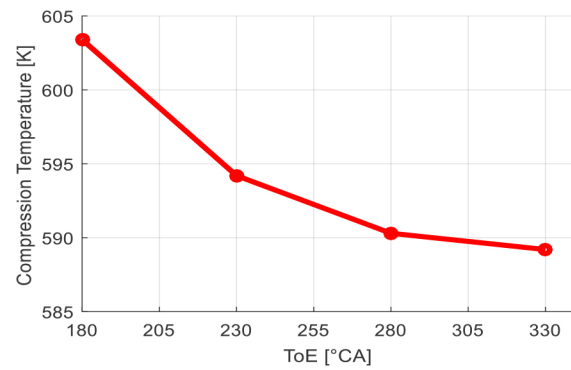


Figure 3. Simulated in-cylinder temperature during the compression stroke (25°CA BTDC) depending on ToE parameter. As in-cylinder evaporation duration (ToE) is longer, the charge temperature is lower.

Figure 3 is related to the same operating conditions of Figure 2, and the effect of ToE parameter is shown: the later (or longer) evaporation in the compression stroke produces higher temperature reduction. Also in this case, such considerations can be generalized, as verified by applying the very same model to different engine operating conditions.

Injected liquid water temperature does not significantly affect model response, as heat subtraction is given basically by its latent heat of evaporation. This means that, supposing an on-board application, water tank position under the bonnet is not particularly constrained. As mentioned before, the model is not sensitive to SOI variations for the port injector, contrary to how the real process is expected to behave.

Similarly, water consumption dependence on these parameters can be analysed, for fixed/given useful temperature reduction effect. In Figure 4, the increment of water consumption is evaluated as a function of F_d (i.e. the fraction of water that evaporates within the cylinder), compared to a direct injection ($F_d = 1$). This analysis is done with fixed ToE , in this case equal to 330°CA, while r is determined to achieve the target effect, which is the same in-cylinder maximum temperature. The engine operating point is the same as the one shown in previous figures, and in the subsequent Figure 5.

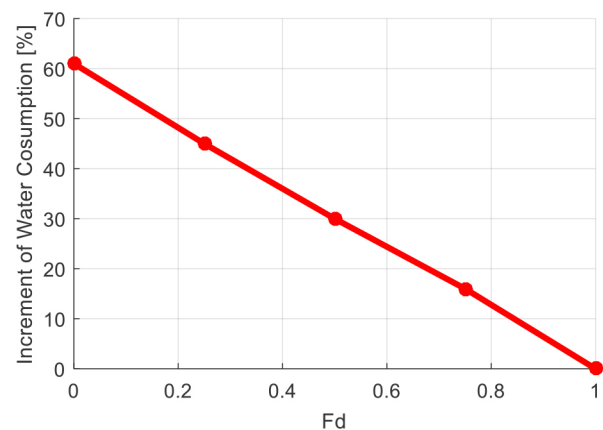


Figure 4. Percentage increase of water consumption, for fixed maximum in-cylinder temperature, depending on the evaporation split between intake ducts (F_i) and combustion chamber (F_d).