

Evaluation of the Potential of Water Injection for Gasoline Engines

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ABSTRACT

Gasoline engine powertrain development for 2025 and beyond is focusing on finding cost optimal solutions by balancing electrification and combustion engine efficiency measures. Besides Miller cycle application, cooled exhaust gas recirculation and variable compression ratio, the injection of water has recently gained increased attention as a promising technology for significant CO₂ reduction. This paper gives deep insight into the fuel consumption reduction potential of direct water injection. Single cylinder investigations were performed in order to investigate the influence of water injection in the entire engine map. In addition, different engine configurations were tested to evaluate the influence of the altering compression ratios and Miller timings on the fuel consumption reduction potential with water injection. Based on the single cylinder investigations, drive cycle simulations covering a low, intermediate, and high load profiles were performed to evaluate the fuel consumption reduction potential as well as the corresponding water consumption under varying load demands for the different engine configurations. Furthermore, these simulations were used to optimize the water injection rate for the altering boundary conditions regarding the trade-off between the fuel and water consumption by the comparison of different criteria. Finally, the most promising criterion regarding a maximal fuel consumption reduction at a given water consumption in all of the investigated driving cycles is used to calculate the trade-off curves between fuel and water consumption as well as possible ranges with for a suitable water tank size.

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INTRODUCTION

Today and for the upcoming years, a continued usage of fossil fuels for worldwide passenger car operation can be expected. However, more stringent regulations regarding CO₂ and pollutant emissions will be enacted by policy makers. At the same time, customers requesting further increased engine power. Especially in the case of transverse engine installation configurations, an increase of the specific power level up to 200 kW/L and beyond is forecasted [1]. Water injection has gained increasing interest again in recent years, as it features efficiency increase, emission reduction by reducing enrichment demand, as well as performance enhancement. A first small series production vehicle has been brought to the market by BMW with the M4 GTS. It uses water injection into the intake manifold for performance enhancement [2, 3].

The effect of water injection was found to enable significant fuel consumption reduction especially at high loads. A detailed description on the different system configurations for water injection in gasoline engines can be found in [4, 5, 6, 7, 8]. Pauer et al. [6] investigated the influence of the positioning of the port water injection. It was found that a closer position to the inlet valve reduces water consumption to less than 50 % compared to an upstream position. In addition, it was found that water injection enabled fuel efficiency increases up to 20 %

when the engine is operated at wide-open-throttle. At the same time power increase by more than 4 % was achieved compared to operation without water injection. Worm et al. [9] investigated the potential of port water injection when combined with a reduced knock resistance of the fuel. Compared to the baseline (91 AKI fuel) efficiency increases of more than 30 % were achieved when operated with 87 AKI fuel and port water injection. At the same time full load engine output was improved by more than 5 %. Similar results for the efficiency increase were found by De Bellis et al. [10] in a combined experimental und numerical study. The increased potential of water injection when injected directly into the combustion chamber was investigated by Durst et al. [4, 5, 8]. Water manifold injection was compared against a separate direct injection with a rather low pressure of 50 bar and a direct injection of water fuel mixture via the fuel injector with 350 bar injection pressure. Advantages compared to the manifold injection where found especially for the high pressure mixture injection. When also high injection pressure is applied to the direct separate injection similar efficiency gains are expected. More information on water fuel mixture injection in gasoline engines can be found in [7, 11, 12]. High pressure separate direct water injection was investigated by Thewes et al. [13] where water injection was combined with a high compression ratio and Miller cycle to exploit the knock reduction potential also at lower engine loads. CO₂ emission reductions of 5 % were possible in the WLTP when compared to an engine concept that already features

variable compression ratio. Further potential was gained when EGR was applied in addition. More detailed experimental analysis on port as well as direct water injection in state-of-the-art gasoline engines can be found in [4, 5, 6, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20, 21].

To investigate the effect of the water injection on expected future high efficiency engine concepts and to account for the strong knock reduction potential of the water injection rather high compression ratios and also Miller cycle are considered in this study. The aim of this contribution is to give a detailed insight into the potential of direct water injection for different engine configurations and boundary conditions. Thereto experimental investigations as well as driving cycle simulations have been conducted. Based on these findings, an optimization regarding the trade-off between fuel and water consumption has been performed.

METHODOLOGY

Engine Specification

The engine used for this part of the investigations uses the same basis as already described in previous publications on water injection [16, 17, 18, 22]. In order to achieve highest reasonable compression ratios (CRs), the stroke of the engine was extended to a stroke/bore-ratio of 1.21 resulting in a cylinder displacement of 400 cm³. More technical data can be derived from [Table 1](#).

Table 1. Technical data of the single cylinder research engine (experimental)

Displacement	cm ³	400
Bore	mm	75
Stroke	mm	90.5
Stroke / bore	1	1.21
Compression ratio	1	13.5 / 14.7
Valves per cylinder		4
Fuel injection pressure	bar	200
Water injection pressure	bar	200
Maximum peak firing pressure	bar	170
Intake valve angle	°	17.5
Exhaust valve angle	°	17.5
Intake event length (1 mm)		
Base	° CA	186
Miller	° CA	230
Exhaust event length (1 mm)	° CA	186

The central DI injector and the spark plug are placed in a central cross position in the combustion chamber roof which means that the spark plug is installed between the exhaust valves, while the fuel injector is installed between the intake valves. The side water injector is installed below the intake port at an installation angle of 19.5°. The central fuel injector features a 6 hole spray configuration which was found to be optimal in terms of mixture homogenization, oil dilution behavior and pre-ignition prevention. The side injector features a 5 hole spray pattern. Fuel, more specific conventional RON95 E10 pump fuel, was used. Two volume flow controlled 200 bar high

pressure fuel pumps were used to pressurize the fuel as well as the water. On the intake side, either a base camshaft with also 186° CA event length or a Miller camshaft with 230° CA event length (late intake valve closing - LIVC) was applied. The CR varies between 13.5 and 14.7. The cylinder pressure is measured by two Kistler 6045B pressure transducers. For the determination of the knock limit the maximum amplitudes (peak-to-peak) of the oscillations due to knock are evaluated. A value of 1.5 times the rpm divided by 1000 is considered as the knock limit. In all load points the ignition timing was set to an optimal center of combustion (50 % of mass fraction burned - MFB50) of 7-8° CA ATDC or to knock limited spark advance (KLSA) in case knock restrictions occur.

Cycle Simulation

The cycle simulations were conducted with a FEV vehicle model with the GT-SUITE software of Gamma Technologies. The vehicle model was adapted to the additional water injection. To account for the high fuel consumption reduction potential of water injection at high engine loads, even with increased CRs, a high specific power engine configuration in combination with an E-segment vehicle was chosen. This increases the share of intermediate to high load points in the driving cycle. Furthermore, the rather high market price of an E-segment vehicle reduces the relative cost for the water injection system.

[Table 2](#) lists the most relevant properties of the vehicle used for the subsequent simulations. The displacement of the 1600 cm³ four cylinder engine corresponds to the cylinder displacement of the single cylinder research engine used for the experimental investigations.

Table 2. Technical data of the vehicle (simulation)

	Unit	Value
Displacement	cm ³	1600
Cylinder number	-	4
Max. engine power (6000 rpm)	kW	180
Max. engine torque (1600 – 5400 rpm)	Nm	293
Max. brake mean effective pressure (1600 – 5400 rpm)	bar	23
Vehicle curb weight		
NEDC	kg	1760
WLTC _{high}	kg	2063
RTS 95	kg	1829
Transmission type	-	AT (8 gears)
Gear ratio		
1. / 2.	-	5.054 / 3.36
3. / 4.	-	2.213 / 1.735
5. / 6.	-	1.324 / 1.02
7. / 8.	-	0.847 / 0.667

In addition to the vehicle and engine data, the model requires engine maps of the fuel and water consumption. These engine maps were derived from the experimental results of the single cylinder engine testing, which are presented in the following section. Uncertainties remain regarding the different behavior of single and multi cylinder engines. This especially relates to the thermal management, gas exchange and boosting system. The influence of the different boosting

systems is reduced as the load points of the cycle simulations cover mostly areas of the engine map where a minor influence can be expected. Due to the water injection a strong reduction of the exhaust gas temperature applies reducing the critical high temperatures for multi cylinder engines at high engine speeds. Again, the concentration of load points in a range with rather low load reduces the influence of this issue. However, the differences between the single and multi-cylinder engine will affect the results of the vehicle simulations. Nevertheless, it is expected that still a valuable comparison of the different configurations investigated in this study is possible by this approach. The simulation model also accounts for the increased friction level due to the additional high pressure pump system necessary for the direct water injection. Therefore, an engine speed and mass flow depending friction level is applied to each load point. This results in an increase of the fuel consumption of $\sim 0.2\%$ in the cycle simulations.

For the simulations three driving cycles covering a low, intermediate, and high load demand were chosen:

1. New European Driving Cycle - NEDC
2. Worldwide Harmonized Light-Duty Test Cycle - WLTC
3. Standardized Random Test Sequence - aggressive - RTS 95

The WLTC includes adjustments of the vehicle curb weight according to e.g. additional equipment. For the investigations, a high curb weight of 2063 kg (see Table 2) was chosen and is thus referred to as $WLTC_{high}$ in the following investigations. Whereas for the NEDC, a value of 1760 kg and for the RTS 95, a value of 1829 kg was set. The RTS 95 was chosen to represent a driving cycle covering a highly dynamic operation under RDE boundary conditions [23]. Apart from the vehicle curb weight, all other boundary conditions were kept constant for the different driving cycles and engine configurations. However, the shifting strategy was adapted for minimum fuel consumption for each of the investigated combinations of driving cycle and engine configuration.

EXPERIMENTAL RESULTS

In the following section, the experimental results of different engine configurations with applied water injection will be discussed. As already described in previous publications [16, 17, 18, 22], the optimal timing for direct injection of water can be found during the compression stroke. It is almost independent from the intake valve closing (IVC), the engine load and the injected water mass. However, it was found that the optimal injection timing of water shifts to earlier timings with increasing engine speed [22]. Therefore, an engine speed related start of the water injection (SOI_{Water}) was set ranging from 90° CA BTDC at an engine speed of 1250 rpm to 120° CA BTDC at 4000 rpm, when operated with a CR of 13.5.

In addition, it was found that the CR also has an influence on the optimal timing of the water injection. Figure 1 depicts a variation of the start of water injection (SOI_{Water}) for the two CRs that were later analyzed in the engine map and cycle simulations.

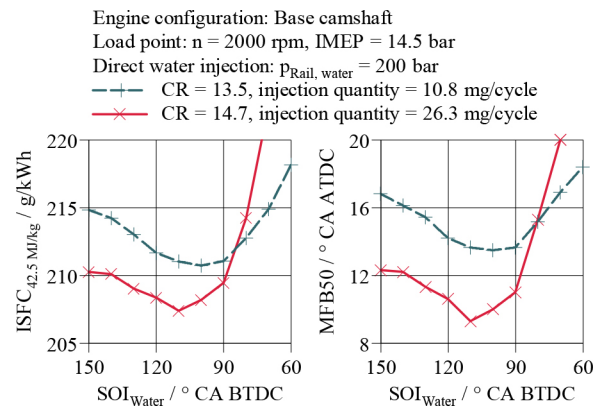


Figure 1. Variation of the start of water injection (SOI_{Water}) for different CRs

Due to the higher injected water mass, corresponding to an increase of the injection duration from $\sim 12^\circ$ CA to $\sim 29^\circ$ CA, earlier MFB50s and thus, together with the higher CR, lower minimal fuel consumption (indicated specific fuel consumption corrected to a heating value of 42.5 MJ/kg - $ISFC_{42.5 MJ/kg}$) were measured with the higher CR. A 10° CA earlier injection timing at 110° CA BTDC for the higher CR (14.7) compared to 100° CA BTDC for the lower (13.5) was found to be optimal in case of highest knock reduction and thus an early MFB50 as well as lowest ISFC. The reason for this can be seen in the higher cylinder pressure and temperature at a certain $^\circ$ CA during the compression for the higher CR which reduces spray penetration length and thus shifts the trade-off for the optimal injection timing to slightly earlier SOIs. Apart from this general shift between the CRs, the engine speed related adaption of the injection timing was kept constant.

As also presented earlier, a high water injection pressure is beneficial for a better primary break-up and evaporation of the injected water [18]. Figure 2 depicts a variation of the water rail pressure at three loads, ranging from moderate engine speed and load to increased engine speed and high load. The end of the water injection was kept constant during the measurement series to avoid an additional effect of the shorter injection timing with increasing injection pressure.

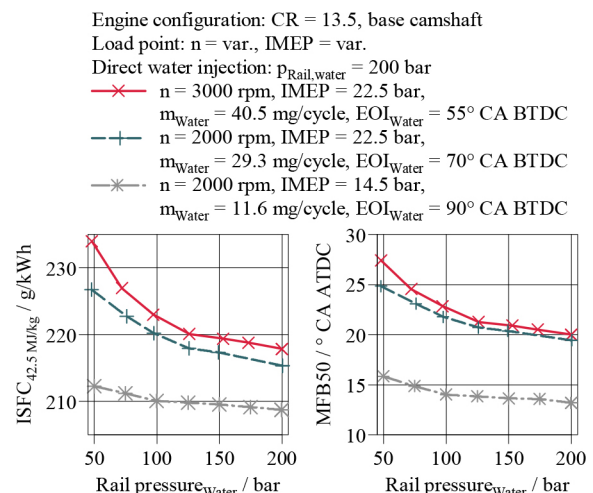


Figure 2. Variation of the water injection pressure for different load points

A constant increase of the knock reduction with higher rail pressure is already visible at the lowest load point. With increasing load and thus injected water mass, the benefit grows. Especially at high engine speeds, the higher rail pressure also helps to reduce the injection timing and thus to avoid unfavorable injection timings which reduce the knock reduction effect as depicted in Figure 1. Thus, the water injection pressure was kept constant at the highest possible system pressure of 200 bar for the following investigations in the whole engine map.

In the following section, the potential of direct water injection to reduce fuel consumption as well as to significantly increase the sweet-spot area in the engine map will be presented. To create the engine maps variations of the water/fuel-ratio up to 100 % were conducted for each engine configuration in various load points as presented in Figure 3 exemplarily for the engine configuration with a CR of 14.7 in combination with the base camshaft. Out of these variations in each load point, the water/fuel-ratio leading to the lowest fuel consumption was selected. This was either when the optimal MFB50 point of $\sim 7\text{--}8^\circ$ CA ATDC is achieved, the maximum water/fuel-ratio of 100 % was reached or the knock reduction of the additional water does not compensate for the additional dilution effect present with increased water/fuel-ratios due to mixture formation issues or saturation. The isoline of a MFB50 at 8° CA ATDC is added in the diagrams to represent the start of the knock restricted load area.

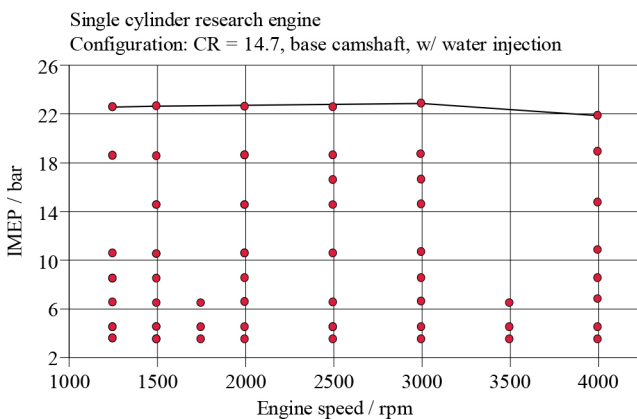


Figure 3. Distribution of engine load points for creation of engine maps

Operation with and without Direct Water Injection

To account for future legislation fuel enrichment and scavenging was avoided for all engine configurations in the complete engine map. The comparison between an engine operation without and with direct water injection is shown in Figure 4 and Figure 5 regarding the ISFC (top) and MFB50 (bottom). For this comparison, the engine configuration with the lowest effective CR of 13.5 and Miller camshaft was chosen. Although rather late MFB50s have to be applied for the operation without water injection (see Figure 4), the comparison allows a good evaluation of the potential of direct water injection to decrease knock limitations.

Without water injection, knock restrictions occur at loads higher ~ 12 bar IMEP due to the relatively high CR (see Figure 4). Only at an engine speed of 2000 rpm, the pursued full load of 22.5 bar IMEP is reached when considering 35° CA as latest acceptable MFB50 at

stoichiometric operation. At the low-end-torque with an engine speed of 1250 rpm, the occurrence of knocking despite the Miller effect and strongly retarded combustion limited further increase of the load above 21 bar IMEP. The reduction of the maximum engine load with increasing engine speed can be explained by the reduction of the Miller effect with LIVC due to the inertia of the air mass. The rather high CR allows for a lowest ISFC of 207 g/kWh at the border of the not knock-limited load area.

When water injection is applied, the knock tendency can be strongly reduced (see Figure 5). This is visible from the earlier MFB50s which are present at loads above 12 bar IMEP. The load point of minimum ISFC is shifted to a 6 bar higher value of 16.5 bar IMEP at the same engine speed of 3000 rpm. At the same time, the minimum ISFC is reduced by 2.7 % to 201 g/kWh. At higher loads, even with increased water injection, an optimal MFB50 cannot be maintained. However, due to the strongly reduced knock restriction and thus earlier MFB50, the sweet-spot area with ISFCs < 210 g/kWh is significantly enlarged reaching full load at ~ 1500 to 2500 rpm. This corresponds to a fuel consumption reduction of 14 % at 2000 rpm compared to an operation without water injection.

The increase of the sweet-spot area gives further advantages in addition to lower minimal fuel consumption especially at high dynamic engine operation, as will be depicted later. Furthermore, the full load can be maintained from 1250 to 4000 rpm as the water injection allows for $\sim 20^\circ$ CA earlier MFB50s. Thus, a load of 22.5 bar IMEP could be achieved with moderately retarded MFB50s.

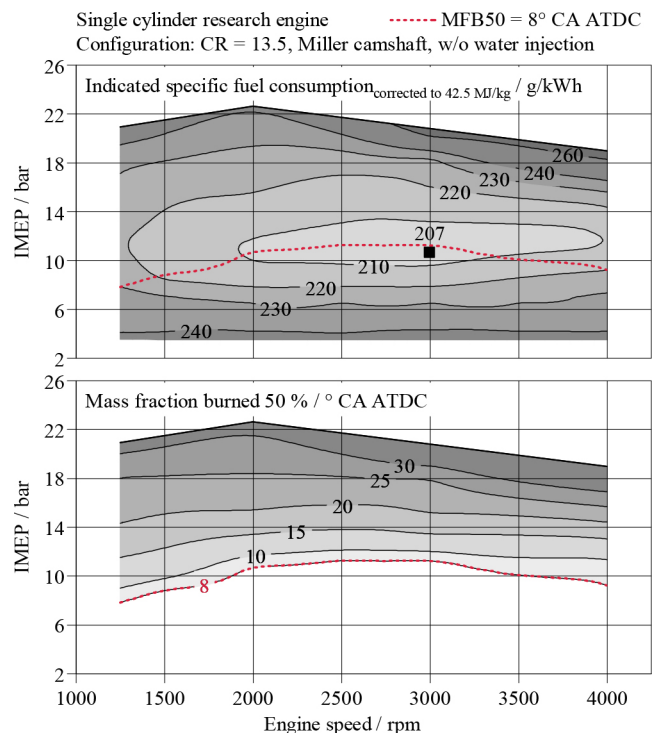


Figure 4. ISFC (top) and MFB50 (bottom) for CR = 13.5 and Miller camshaft without water injection