Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 148 (2018) 50–57

Energy Procedia

www.elsevier.com/locate/procedia

73rd Conference of the Italian Thermal Machines Engineering Association, ATI2018, 12-14 September 2018, Pisa, Italy

Basics on Water Injection Process for Gasoline Engines

Stefania Falfari[a,*], Gian Marco Bianchi[a], Giulio Cazzoli[a], Claudio Forte[b], Sergio Negro[c]

[a]*DIN, Alma Mater Studiorum University of Bologna*
[b]*NAIS srl*
[c]*Litens Automotive Partnership*

Abstract

Actual and future limits to the global CO_2 emissions and the necessity of a further reduction of the fossil non-renewable fuels have moved the automotive engine research toward new solutions. With focus on reciprocating internal combustion engines, the mass of CO_2 emitted in the atmosphere is a function of the fuel consumption. Therefore, the designers are focusing their attention on both the drop of passive resistances and the improvement of the engine efficiency. As far as the latter is concerned, the reduction of in-cylinder temperature and the adoption of stoichiometric combustion on the full range of engine operation map are the most investigated solutions. Water injection is thought to help in fulfilling these goals thus contributing towards more efficient engines. The aim of the present work is to understand the basic thermophysical and chemical fundamentals governing the water injection application in modern downsized spark ignited engines. The investigation has been carried out with aid of CFD simulation by using AVL FIRE v.2017 solver.

Keywords: SI Engines, Downsizing, Water injection, CO_2, Fluid-dynamic analysis

1. Introduction

In order to accomplish with CO_2 emission limit tightening, different technologies, or more likely a combination of them, might be used in the next generation S.I. engines: : i) further engine downsizing; ii) stoichiometric combustion;

* Corresponding author. Tel.: +39-0512093316
E-mail address: stefania.falfari@unibo.it

10.1016/j.egypro.2018.08.018

iii) new fuels (biofuels technology and/or synthetic fuels); iv) high-pressure injection systems; v) advanced combustions (gasoline compression ignition, for example); vi) advanced ignition system; vii) variable geometric and operative engine parameters. All these solutions carry open critical issues, such as making the engine more prone to detonation and pre-ignition as well as to the increase of the Turbine Inlet Temperature (TiT). The water injection is viewed as one of the possible technical solution since it promises to make the engine operating with larger compression ratios and stoichiometric combustions at high load, resulting in a drop of the specific fuel consumption on the overall engine map. Water injection has been investigated by many authors due to its anti-knock attitude in S.I. engine applications. Lanzafame et al. [1, 2] experimentally and numerically analysed the effects of both continuous and pulsed water injection in the intake duct at low pressure (10 bar), finding that the water injection successfully expand the engine knock limits. Boretti [3] presented a comparison between water injection in the intake duct or directly in the cylinder. The injection in the intake duct has a cooling effect on the intake air mass, reducing its density and enhancing engine permeability and power density. The injection directly in the chamber leads to a cooler combustion phase, followed by a lower exhaust temperature. Bhagat et al. [4] performed a numerical study after the model validation. The high-pressure (200 bar) water injector was centrally located in the combustion chamber. In [5, 6] the position of the water injector in the intake duct was studied, finding that a closer position to the intake valve reduces the water consumption to less than 50% compared to an upstream position. Considerations on the effect of water injection on combustion duration are also drawn. In [7, 8, 9] the authors studied the potential benefits of the water injection on a PFI turbocharged engine, combining the effect of the water injection with the effect of the exhaust gas recirculation for reducing the exhaust gas temperature.

Most of the water injection studies were performed for a Port Water Injection (PWI) solution, because it is less costly and the injector can be easily installed on a pre-existing engine [10]. Some authors [11, 12, 13] worked on a water direct injection system. In general, the injection pressure for PWI applications in literature ranges between 4 bar up to 10 bar [14, 7, 10]. In the current car market, BMW sells the M4-GTS equipped by three PWI water injectors, one every two cylinders. They located the water tank in the trunk, with a capacity of five liters. The injection pressure was set to 10 bar.

Since few data were reported on detailed investigation on the effect of water properties in engine applications, the present paper deals with analysis of the expected effect of the main physical and chemical properties of the water injection on the main engine processes toward its efficient and feasible applications to series production.

Despite the fact that many evidences show that the water injection can be considered as a promising candidate to improve the engine efficiency, on the other hand its application is faced with several open issues that must be addressed in order to:

- Minimizing water consumption;
- Minimizing costs;
- Improving reliability of injection systems;
- Minimizing lubricant contamination;
- Fulfilling turbine inlet temperature (TiT) maximum limit;
- Verifying its compatibility with cold climate temperatures.

Thus, a deep knowledge of the physical, thermodynamics and chemical processes is necessary to optimize the water injection operation in S.I. engines, including the best choice of water injector installation and the injection timing.

2. Water injection: physical and chemical properties

The final aim of the water injection is to cool a mixture for both reducing the unburnt gas temperature level during combustion (for reducing knock) and fulfilling the TiT maximum limit thus allowing the use of design solutions or calibrations that provide the increase of the engine efficiency. If stoichiometric operations are considered at full load, the water injection replaces the cooling effect of a part of the excess of fuel with further benefit in the *bsfc*.The water is injected in liquid phase in the air or in the air/fuel mixture during the intake or the compression phase, seeking for the highest cooling effect together with the necessity of avoiding the formation of water liquid film on walls. The injector can be located in the intake ports or manifold (namely Port Water Injection (PWI)) or can be mounted in the cylinder (namely Direct Water Injection (DWI)). The mass of the water injected per cycle is usually expressed by the parameter *s*, defined as the ratio between the injected liquid water mass m_W and the stoichiometric fuel mass m_{FUEL}, for a given mass of air trapped per cycle:

$$s = \frac{m_W}{m_{FUEL}} \quad (1)$$

Typically, the parameter *s* varies in the range 20% to 50%, which corresponds to a fraction of injected water mass approximatively equal to 6% of the total trapped mass per cycle.

In Table 1, the main physical properties of the water are compared with those of iso-octane and commercial gasoline RON 95. The **latent heat of vaporization** (LHoV) is the physical properties that makes the water injection so much attractive for S.I. engine applications since it is six times greater than that of gasoline RON95 at atmospheric pressure. It must be noted, however, that the LHoV is a function of the current ambient pressure and, in particular, it decreases when the pressure increases. In Figure 1 the evolution of the LHoV in a typical range of S.I. engine in-cylinder pressure trend at high load is depicted. If the ambient pressure, where the liquid water evaporates, increases from 1 bar to 20 bar, the LHoV reduces of about 19%. As far as the injection is considered, **the water surface tension** is also of worth importance since it is about four times greater than that of gasoline RON95 (or iso-octane), as discussed in [12]. The result is a weaker attitude of liquid water to atomisation, suggesting the adoption of higher injection pressures than those used for gasoline (the value depends on application considered: port or direct fuel injection). Figure 2 shows the different values of the characteristic liquid jet non-dimensional parameters (e.g., Ohnesorge and Reynolds numbers) for iso-octane and water in a direct injection application. Moreover, it is important to consider the variation of the polytropic compression index: it becomes higher as the mixture index is increased toward the stoichiometric value and it gets even greater if vaporized water is added. The whole result is an increase of the mixture temperature at TDC.

Table 1. Comparison between water, iso-octane and RON95 main properties at 25°C

	Water	Iso-octane	Gasoline RON95
Surface tension σ [N/m]	72.71e-3	18.32e-3	19.80e-3
Latent heat of vaporization [kJ/kg] at 1 bar	2257.00	307.00	397.00
Vapour pressure p_V [kPa] at 25°C	2.34	5.30	5.90
Viscosity ν [mPa·s] at 25°C	0.88	0.47	0.50
Density [kg/m^3] at 25°C	999.00	690.00	750.00

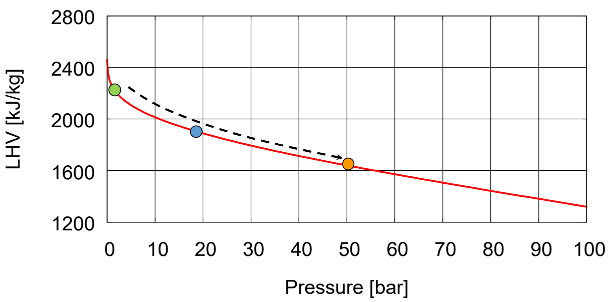


Figure 1: Effect of ambient pressure on the water latent heat of vaporization

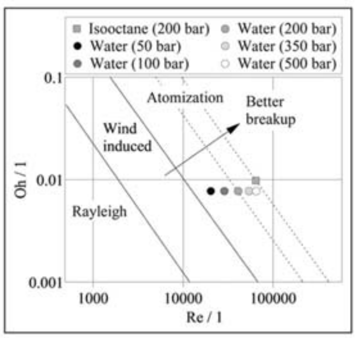
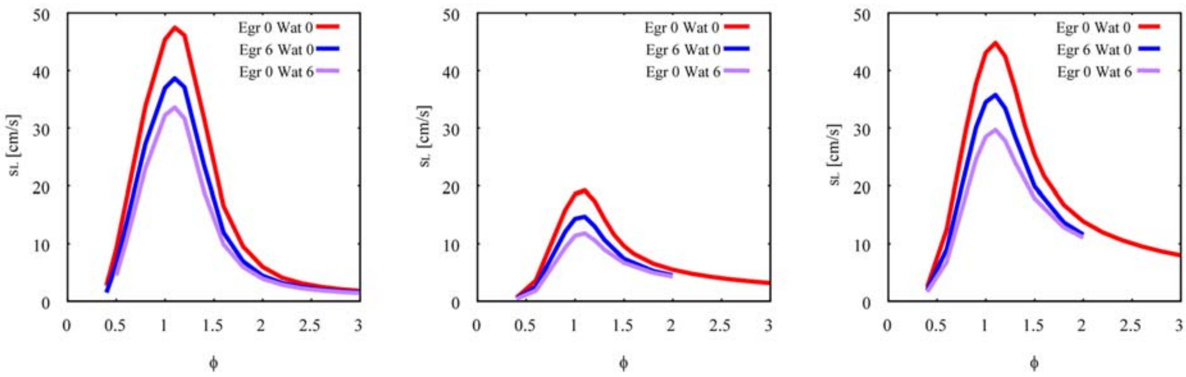


Figure 2: Liquid jet atomization regimes depending on the injection pressure level and fluid type [12]



a) b) c)

Figure 3: Laminar flame speed vs equivalence ratio at ambient conditions: a) 1 bar and 358 K; b) 50 bar and 358 K; c) 50 bar and 600 for different percentages of water (Wat) and residual gas fraction (Egr)

For a more comprehensive overview on the effect of the water injection, one has to consider how **the laminar flame speed** is affected by water since it becomes a mixture diluter (see [18, 20] for more details). Figure 3 shows influence of the water mass fraction on the laminar flame speed at given different pressures, temperatures and residual gas fractions (e.g., 1bar and 50 bar, 358K and 600 K, 0% and 6% EGR) versus the equivalence ratio (all parameters were numerically derived):

1. As one can expect, the laminar flame speed decreases of about minus 19% with respect to the reference case (0% EGR – 0% water) when the EGR mass fraction is increased from 0% to 6% (Figure 3a), without vaporised water (0%).
2. If one consider zero EGR mass fraction, increasing the vaporised water mass fraction from 0% to 6%, the drop of the laminar flame speed is even larger (minus 32%) with respect to the nominal case (0% EGR – 0% water).
3. The effect of the pressure on the laminar flame speed is more marked than the effect of the temperature. At the same ambient temperature, the more the ambient pressure, the less the laminar flame speed.

Therefore, the increase of the vaporised water mass fraction in the in-cylinder trapped mass drops the laminar flame speed, thus both lengthening the combustion duration and increasing the TiT more consistently than using the fuel for cooling mixture and exhaust gases. As a result, the combustion system has to be designed to counterbalance the longer combustion duration and the TiT increases. The experimental evidence shows that the disadvantage of the TiT increase usually disappears by simply injecting more water mass per cycle. Once again, the experiments show benefits but some details must be further investigated since complete evaporation of the injected mass might not be guaranteed providing several concerns on water consumption, as below it is discussed.

3. Water evaporation under S.I. engine conditions

The goal of the water injection is to replace the excess of fuel in order to achieve the largest mixture cooling effect with the lower water consumption for a given target of increase of *bme*p, reduction of *bsfc*, or both of them, at a given maximum turbine inlet temperature. Therefore, the injector location, the injection system specification, including the spray Sauter mean diameter, and the injection timings must be chosen in order to provide the fastest and most complete water mass evaporation over a wide range of engine speeds. The crank angle window available for the evaporation can be assumed no longer than 250 ca deg. (including intake phase and compression phase). Thus, the time available for evaporation can be easily estimated and reduces linearly as the engine speed increases. On the other hand, the time needed to complete the evaporation of the injected water mass is a function of the ambient (intake manifolds/ports or cylinder) conditions like pressure and temperature and, for fixed thermodynamic conditions, of the parameter s (e.g. the non dimensional mass of water injected). As a result, the evaporation rate and saturation limits depend on the water injector location or, more generally, the region where liquid water evaporates. According to the latter, water injection and water evaporation are faced with several physical constraints or processes and the most important of them are resumed below:

1. The risk of reaching the saturation limits is greater when the ambient temperature is lower. This outlines the requirement to avoid the water liquid film formation on intake manifold or port walls and to make water evaporate outside these low temperature regions.
2. The need to avoid water liquid film formation on wall to maximize the cooling effect and to avoid lubricant contamination with water if the liquid film is formed onto the cylinder liner surface. Thus, in the case of intake manifolds/ports water injection solution, the injection time must be set to seek for a complete evaporation of the water inside the cylinder avoiding wall impact.
3. The requirement to achieve the best balance between the achievement of the maximum cooling effect and the need to promote a complete water evaporation since water consumption is a concern.
   This point can be divided into two sub-topics: (a) the influence of injection timing on the latent heat of vaporization; (b) the effect of ambient conditions and injection system specifications on the evaporation rate (e.g. the time required to achieve a complete evaporation).
   As far as the first topic is considered, one can see that an earlier water End Of Injection (EOI) allows one to inject under lower ambient pressure conditions taking the advantage of the larger latent heat of vaporization (Figure 4 and Figure 5). On the other hand, a later set of the EOI enhances the vaporisation rate (Figure 6) since the mixture temperatures are higher. In order to highlight these opposite effects, a simple ideal thermodynamic analysis is presented taking as a reference the in-cylinder pressure and temperature evolution of a typical S.I. downsized GDI engine at full power obtained without considering the water injection (Figure 7 and Figure 8). The non dimensional parameter *s* is fixed at 0.2. The ideal and best condition would require an instantaneous (isochoric) water evaporation at IVC when the system becomes closed. Under real operating conditions, a finite evaporation time should be considered and the compromise above mentioned must be found. In order to assess the influence of delaying injection (e.g. evaporation) on the mixture cooling effects at TDC, the isochoric evaporation at different crank angles of a fixed water mass has been considered, as one can see in Figures 7 to 9, where the evaporation crank angles are highlighted by coloured circles. The mixture cooling effect at TDC changes when the water evaporation is delayed and, in particular, it drops of about 6 K moving the isochoric evaporation from IVC to 690 ca deg. ATDC (Figure 9).

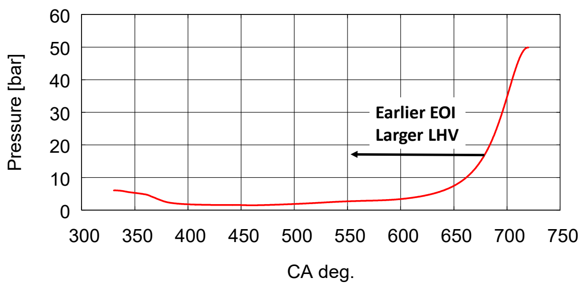


Figure 4: Typical in-cylinder pressure evolution of a SI engine at full power

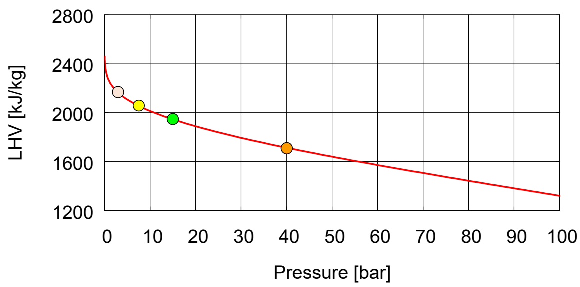


Figure 5: Water latent heat of vaporization Vs pressure

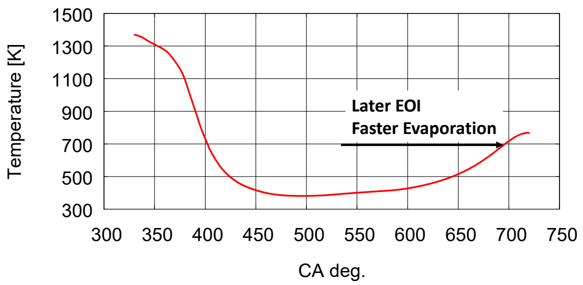


Figure 6: Typical in-cylinder temperature evolution of a SI engine at full power

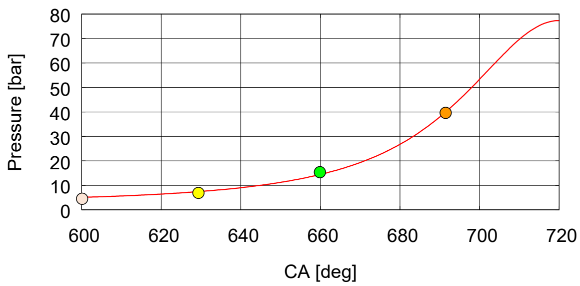


Figure 7: In-cylinder pressure evolution of a SI engine at full power

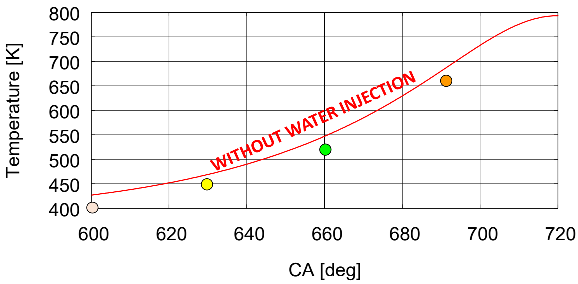


Figure 8: In-Cylinder temperature evolution of a SI engine at full power with and without water injection cases compared

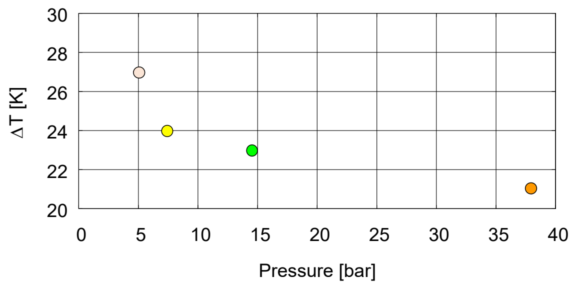


Figure 9: Mixture cooling effect at TDC Vs pressure at evap.

As far as the second topic is concerned, one has to consider that both ambient conditions and injection system specifications play a role in defining the evaporation rate. The latter is of worth importance since the overall evaporation time available is limited by the engine speed and, in particular by the maximum power engine speed and the target non-dimensional water mass injected *s*. In order to assess the effect of the air (mixture) thermodynamic conditions in the engine during water evaporation, of the injection system characteristic on the water evaporation rate and of the injected water mass on the evaporation time, a matrix of 27 cases of CFD simulation of water injection into a quiescent air filling a vessel were carried out. The matrix of tested cases includes: (1) *Three different injection pressure intervals and three corresponding mean spray SMD values*, representing of both Port Water Injection and Direct Water Injection systems, at a fixed value of parameter *s* (e.g. water mass). (2) *Three different representative engine conditions*, selected based on Figures 4 and 5: 300 K & 2.5 bar, 450 K & 4.5 bar, 650 K & 22 bar. They are assumed to represent port and direct water injection conditions at different E.O.I. (3) *The effect of the injection system in terms of spray droplet sizing*, by injecting sprays having different SMD depending on the selected injection pressure interval: a) 30, 50 and 90 μm for injection pressure interval 5-10 bar; b) 20, 30 and 40 μm for injection pressure interval 25-50 bar: c) 10, 15 and 20 μm for injection pressure interval 150-200 bar. All these spray SMD values, and so the corresponding different injection pressures, are considered representative of Port Water Injection systems (5-25 bar) or Direct Water Injection solutions (50 to 200 bar).

The spray breakup model was disabled in simulation in order to isolate the effect of the droplet size. As far as the evaporation rate is considered, Figure 10 depicts the effect of the ambient temperature, the injection duration (level of injection pressure) and the spray SMD of the injected water on the evaporation rate. It is very clear that the evaporation rate at 300 K is very slow even with the smallest droplet sizing considered. This confirms the evidence that in the case of port injection, the water indeed has to evaporate at higher temperatures likely those occurring into the cylinder. The possibility to operate with the largest pressure and lowest SMD allows one to increase the evaporation rate of almost one order of magnitude (Figures 10a-10c) moving toward the ideal isochoric evaporation. Moreover, larger the evaporation rate, larger is the maximum value of *s* allowed and larger is the allowed rated power engine speed for a fixed *s*. It is to mention that there is also a strong dependence of the evaporation rate on the ambient temperature (Figure 11) and that water should evaporate at temperature larger than 450 K. Finally, Figure 12 reports the evaporation rates corresponding to the finest granulometry of all the considered sprays at the three injection pressure levels considered versus the air (mixture) temperature. It is clear that at 5-10 bar, injection pressure interval representative of the water injection process in intake manifolds/ports, the evaporation rate appears always to be too very slow, also at the temperatures of 450 K and 650 K because of the spray SMD. Even the value of 0.4 mg/ms, which is the minimum evaporation rate at 2000 rpm (the lowest considered engine speed), is too much if compared to the feasible evaporation rate at 300 K. At 6500 rpm, the necessary evaporation rate (1.38 mg/ms in Figure 12) is feasible for both injection pressure levels typical of high pressure PWI or DWI solutions at 650 K (delayed EOI) or at 450 K (earlier EOI) for injection pressure level 150-200 bar, because the evaporation rate fulfils the minimum required. This simple analysis is effective to highlight the scenario which might be beyond a choice of the water injector location (PWI/DWI) and the operating pressure and it provides suggestions to injection time setting.

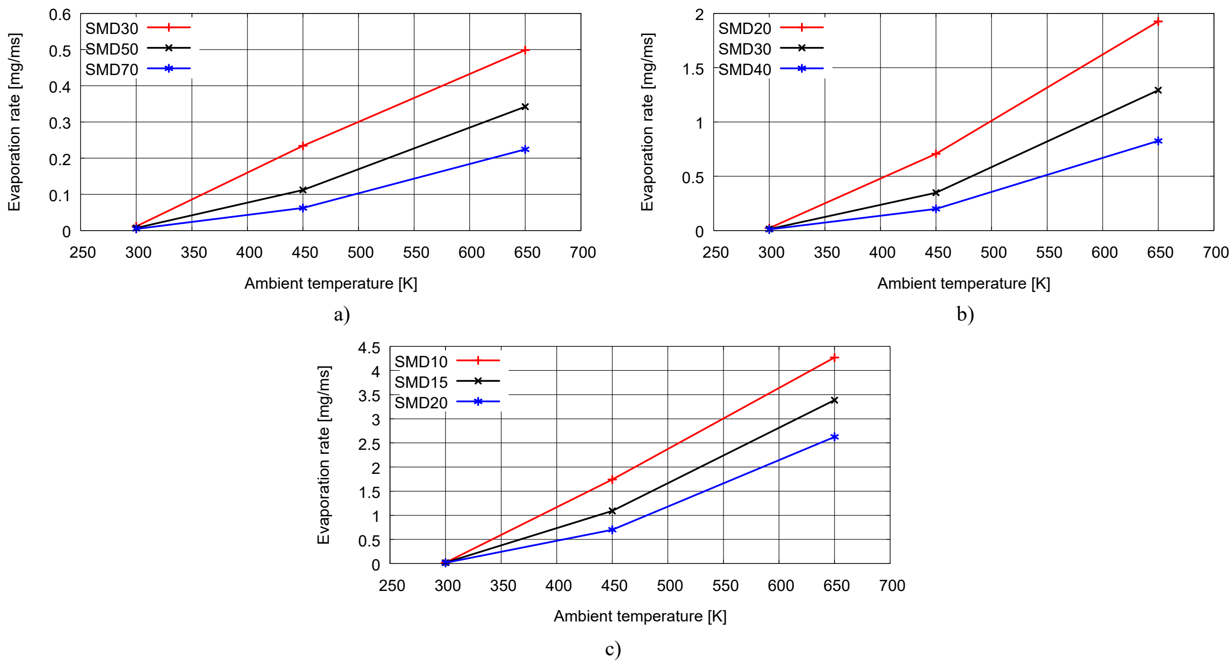


Figure 10: s=0.4 and different injection pressures: a) 5-10 bar; b) 25-50 bar; c) 150-200 bar

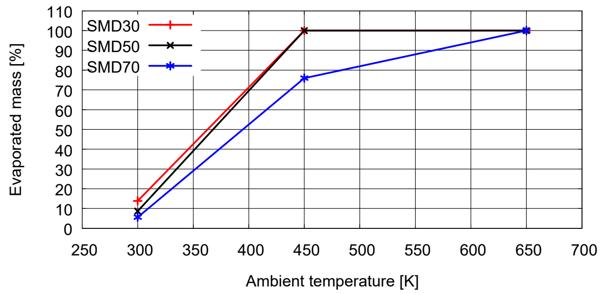


Figure 11: Percentage evaporated mass at 5-10 bar

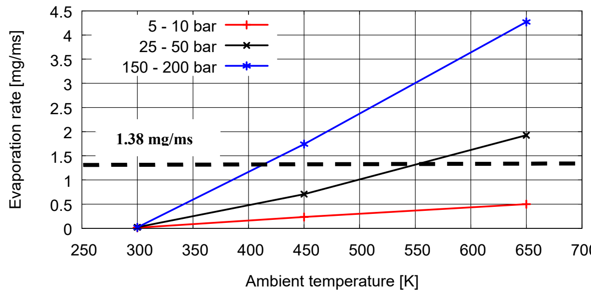


Figure 12: Evaporation rate for the finest spray SMD - s=0.4 at different injection pressure intervals

4. Water injection for engine application – Some issues

From the above considerations, arise some recommendations:

1. The need to find the best trade-off between the necessity of lowering the fuel consumption and the need of limiting the water consumption, in order to increase vehicle autonomy and reduce weight/bulk of the water tank. Consider that the vehicle might undergo a recovery condition if the water tank gets empty;
2. Taking into account that the air temperature plays a key role in achieving a complete water evaporation, thus suggesting that, in case of water port fuel injection, water must evaporate inside the cylinder where temperatures are much higher than within intake manifold or ports.
3. The impact of the water on the wall must be avoided for achieving the largest cooling effect, avoiding the risk of both local saturation and lubricant contamination (impact on cylinder liner).
4. These results suggest that to achieve the most flexible and efficient operation with water the Direct Water Injection solution would be preferable. DWI is faced with cost limits and reliability issues to the large thermal load the injector suffers anytime is switched off.
5. In case of PWI solutions, injection timing should promote the water evaporation inside the cylinder.

5. Conclusions

The present work is a preliminary study oriented to the comprehension of the physical, chemical and thermodynamic properties of the water. From this study, some important issues emerged:

a. The more the percentage of vaporized water in the chamber, the less the laminar flame speed;
b. The higher surface tension of the water would require larger injection pressure for atomizing the spray;
c. The latent heat of vaporization is larger than the RON 95 gasoline but it is pressure dependent and this could affect the choice of the SOI/EOI.

The most important parameters than need to be accounted for in the definition of water injection systems are:

1. The choice of the type of injection system: PWI or DWI;
2. The minimum mass of water to be injected for the focused cooling effect on the fresh mixture;
3. The injection pressure and injector technology (spray SMD) has a huge effect on the evaporation rate;
4. Depending on the injection solution and the injection timing, it must be considered the possible effects of the injection on the main in-cylinder turbulent intensity.

References

[1] Lanzafame, R., "Water Injection Effects In A Single-Cylinder CFR Engine," SAE Technical Paper 1999-01-0568, 1999, https://doi.org/10.4271/1999-01-0568.

[2] Brusca, S. and Lanzafame, R., "Water Injection in IC - SI Engines to Control Detonation and to Reduce Pollutant Emissions," SAE Technical Paper 2003-01-1912, 2003, https://doi.org/10.4271/2003-01-1912.

[3] Boretti, A., “Water injection in directly injected turbocharged spark ignition engines”, Applied Thermal Engineering, 52(1):62 - 68, 2013, https://doi.org/10.1016/j.applthermaleng.2012.11.016.

[4] Bhagat, M., Cung, K., Johnson, J., Lee, S. et al., "Experimental and Numerical Study of Water Spray Injection at Engine-Relevant Conditions," SAE Technical Paper 2013-01-0250, 2013, https://doi.org/10.4271/2013-01-0250.

[5] D'Adamo, A., Berni, F., Breda, S., Lugli, M. et al., "A Numerical Investigation on the Potentials of Water Injection as a Fuel Efficiency Enhancer in Highly Downsized GDI Engines," SAE Technical Paper 2015-01-0393, 2015, https://doi.org/10.4271/2015-01-0393.

[6] Pauer, T., Frohnmaier, M., Walther, J., Schenk, P. et al., “Optimization of Gasoline Engines by Water Injection,” in: 37th International Vienna Motor Symposium, 2016.

[7] Worm, J., Naber, J., Duncan, J., Barros, S. et al., "Water Injection as an Enabler for Increased Efficiency at High-Load in a Direct Injected, Boosted, SI Engine," SAE Int. J. Engines 10(3):951-958, 2017, https://doi.org/10.4271/2017-01-0663.

[8] De Bellis, V., Bozza, F., Teodosio, L., and Valentino, G., "Experimental and Numerical Study of the Water Injection to Improve the Fuel Economy of a Small Size Turbocharged SI Engine," SAE Int. J. Engines 10(2):550-561, 2017, https://doi.org/10.4271/2017-01-0540.

[9] Iacobacci, A., Marchitto, L., and Valentino, G., "Water Injection to Enhance Performance and Emissions of a Turbocharged Gasoline Engine under High Load Condition," SAE Int. J. Engines 10(3):928-937, 2017, https://doi.org/10.4271/2017-01-0660.

[10] Cavina, N., Rojo, N., Businaro, A., Brusa, A. et al., "Investigation of Water Injection Effects on Combustion Characteristics of a GDI TC Engine," SAE Int. J. Engines 10(4):2209-2218, 2017.

[11] Jaeheun Kim, Hyunwook Park, Choongsik Bae, Myungsik Choi, Younghong Kwak “Effects of water direct injection on the torque enhancement and fuel consumption reduction of a gasoline engine under high-load conditions”, International Journal of Engine Research, Volume: 17 issue: 7, page(s): 795-808, https://doi.org/10.1177/1468087415613221.

[12] Hoppe, F., Thewes, M., Baumgarten, H. Dohmen, J., “Water injection for gasoline engines: Potentials, challenges, and solutions”, International Journal of Engine Research, 2015, Volume: 17 issue: 1, page(s): 86-96, https://doi.org/10.1177/1468087415599867.

[13] Hoppe, F., Thewes, M., Seibel, J., Balazs, A. et al., "Evaluation of the Potential of Water Injection for Gasoline Engines," SAE Int. J. Engines 10(5):2500-2512, 2017.

[14] Busuttil, D., Farrugia, M., “Experimental investigation on the effect of injecting water to the air to fuel mixture in a spark ignition engine”, March 2015MM Science Journal, 2015(01):585-590, doi: 10.17973/MMSJ.2015_03_201510.

[15] Russ, S., “A review of the effect of engine operating conditions on borderline knock”, in SAE Technical Paper, SAE International, 02 1996, https://doi.org/10.4271/960497.

[16] Caris, D. and Nelson, E., "a new look at High Compression Engines," SAE Technical Paper 590015, 1959, https://doi.org/10.4271/590015.

[17] Gerty, M. and Heywood, J., "An Investigation of Gasoline Engine Knock Limited Performance and the Effects of Hydrogen Enhancement," SAE Technical Paper 2006-01-0228, 2006, https://doi.org/10.4271/2006-01-0228.

[18] Cazzoli, G., Forte, C., Bianchi, G., Falfari, S. et al., "A Chemical-Kinetic Approach to the Definition of the Laminar Flame Speed for the Simulation of the Combustion of Spark-Ignition Engines," SAE Technical Paper 2017-24-0035, 2017, doi: 10.4271/2017-24-0035.

[19] Jun-Kai Wang, Jing-Lun Li, Ming-Hsun Wu and Rong-Horng Chen, “Reduction of Nitric Oxide Emission From a SI Engine by Water Injection at the Intake Runner”, Paper No. IMECE2009-12517, pp. 335-340; 6 pages, doi:10.1115/IMECE2009-12517.

[20] Giulio Cazzoli, Gian Marco Bianchi, Stefania Falfari, Claudio Forte, “Development of a chemical-kinetic database for the laminar flame speed under GDI and water injection engine conditions”, submitted to ATI 2018.