Comparative Evaluation of Water-Fuel Emulsion and Intake Air Humidification: Effects on HD DI Diesel Engine Performance and Pollutant Emissions

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ABSTRACT

A theoretical study is conducted to examine the effect of water addition on a heavy-duty (HD), direct injection (DI), diesel engine's performance and pollutant emissions. Three water addition strategies are examined, i.e. water fuel emulsification keeping constant injection duration as the water percentage increases, water fuel emulsification with varying injection duration as the water percentage increases, and inlet air humidification. To investigate the effect of water content on engine performance and pollutant emissions three water percentages are examined, namely 10%, 20% and 30% by weight. For all cases examined, the engine brake power output has been kept constant and equal to the one corresponding to the baseline case with no water addition. From the computational study conducted, it is revealed that intake air humidification reduces NO (nitric oxide) emissions but it leads to significant increase of soot emissions and to a slight increase of bsfc (brake specific fuel consumption). On the other hand, using water/fuel emulsion a slight improvement on bsfc is achieved, while a significant reduction is observed on NO and soot emissions compared to the baseline case with no water addition. As water percentage increases, the beneficial effects of water addition are enhanced. However, when water fuel emulsion is used with varying fuel injection duration, and the water percentage is higher than 20%, a slight increase is observed in bsfc and soot emissions, a fact owed to the significant increase of injection duration.

Keywords: diesel engine; water emulsified fuels; air humidification; performance; emissions

INTRODUCTION

Today's diesel engines are well established as the dominating power train solution in the world market [Benson and Whitehouse 1979), Heywood (1988)]. Especially in the area of transportation, diesel engines are considered as one of the main power sources. This is owed to the substantial improvements of these engines, especially the DI ones, in terms of brake efficiency and reduction of pollutant emissions (mainly nitric oxide and soot particulates). It is expected that these improvements will secure the domination and further expansion of the use of diesel engines in the forthcoming years.

Remarkable advantages towards the development of cleaner diesel engines have been achieved, over the last years, by following various engine-related techniques, such as for example the use of common-rail systems, fuel injection control strategies, exhaust gas re-circulation, exhaust gas after-treatment, oxygen enriched intake-air, etc [Ferguson (1986), Heywood (1988), Stone (1992)].

Moreover, especially for the reduction of pollutant emissions, researchers have focused their interest on the domain of fuel-related techniques, such as for example the use of alternative fuels often in fumigated form [Kouremenos and Rakopoulos (1986), Kouremenos et al. (1990)], or gaseous fuels of renewable nature that are friendly to the environment [Rakopoulos and Kyritsis (2001), Rakopoulos et al. (2006a)], or oxygenated fuels that present reduced particulate emissions [Xiao et al. (2000), Rakopoulos et al. (2004)] usually with an escorting increase of the emitted nitrogen oxides.

Moreover, considerable attention has been paid on the development of alternative fuel sources in various countries, with particular emphasis on the bio-fuels that posses the added advantage of being renewable fuels that can be replenished through the growth of plants or production of livestock, showing an *ad hoc* advantage in reducing the emitted carbon dioxide [Rickeard and Thompson (1993), Rakopoulos et al. (2006b), Hansen et al. (2005)]. Among these, vegetable oils or their derived bio-diesels (methyl or ethyl esters) and bio-alcohols are considered as very promising fuels [Graboski and McCormick (1998), Scholl and Sorenson (1993), Rakopoulos (1992), Rakopoulos et al. (2006c), (2006d), Ecklund et al. (1984), Rakopoulos et al. (2006e)]. Vegetable oils and bio-diesels are good diesel engine fuels owing to their reasonably high cetane number, while ethanol is a good spark-ignition engine fuel because of its high octane number, though it has been considered recently and as an alternative fuel (in blends) for diesel engines.

The present work corresponds to the group of techniques with the purpose of affecting the combustion mechanism and consequently of reducing the formation of pollutant emissions. Water/fuel emulsions have been studied extensively during the last years as an alternative to conventional diesel fuels [Henningsen (1994), Wirbeleit et al. (1997), Song et al. (2000), Armas (2005), Samec (2002), Abu-Zaid (2004), De Fries (2004), Matthews et al. (2004), Musculus et al. (2002), Weber et al. (2004), Ahern et al. (2001)]. The presence of water inside the engine cylinder is known to result to reduction, especially of nitric oxide (NO) and particulate matter (PM) in some cases. However, significant problems had kept the application of this technology away from commercial engines. Nonetheless, during the last years research in this field has been intensified, aiming at the optimization of the relevant systems for maximum emissions reduction potential.

In the literature, it has been reported that the water fuel emulsion improves the fuel-air mixing process. This improvement is ascribed to the increased momentum of the water emulsified fuel spray and the micro-explosion phenomena [EPA Report (2002)]. Micro-explosions (also called secondary evaporation) help to accelerate the evaporation of fuel droplets in emulsions, and are strong enough to eject fragments of torn droplets several millimeters away from the limits of the spray cone at high speeds, a fact that can help to improve the air-fuel mixing.

The effect of the percentage of water added to diesel fuel is of high importance. Moreover, significant engine modifications may be necessary before using emulsions. One example is the adjustment of the injection timing to optimize engine performance when using emulsified fuel.

In the present study, the authors use a multi-zone phenomenological combustion model to examine how engine performance and emissions are affected by water addition. The engine used for the simulation is a heavy-duty, DI diesel engine. Two different techniques are examined for the water addition into the engine cylinder. According to the first one the water/fuel emulsion is injected directly into the combustion chamber, while with the second one water is injected separately into the intake pipe and consequently it mixes with the air inside the combustion chamber. The model is calibrated and then validated via available experimental data for the reference case (with zero water percentage). Four different engine loads at a speed of 1800 rpm are examined, in order to cover the whole engine operation field. Moreover, three different water percentages are examined in each case to investigate the optimum water amount concerning NO emission reduction, in combination with the resulting penalty in specific fuel consumption and soot emissions.

Since the main scope of the present work is to investigate the effectiveness of water addition to reduce pollutant emissions, a prerequisite has been set that, for all cases examined, the engine power output is equal to the one corresponding to the base case (no water addition). When using emulsified fuel blends, owing to their lower heating value compared to the one of the pure fuel blends, special measures have to be taken in order to maintain engine power unchanged. In this study, two different techniques have been examined. According to the first one, the injection duration extends as the percentage of the water added into the fuel blend increases so that, irrespectively of the amount of water injected into the cylinder, the engine power output remains unchanged. Alternatively, one could maintain the injection duration constant and compensate for the negative effect of water/fuel emulsion on engine power output, by increasing the diameter of the injector's nozzle holes, retaining injection pressure constant. Although the later is not easily applicable, it is interesting to investigate how it affects engine performance and emissions.

ENGINE SIMULATION MODEL

OUTLINE OF THE MODEL

The authors have already given in the past a comprehensive description of the multi-zone simulation model used in the present study [Rakopoulos and Hountalas (1998), Hountalas et al. (2002a), (2004)]. Thus, only a brief outline of its main principles and components is presented in the following, giving emphasis on the modifications necessary to account for the effect of water addition in the combustion chamber. The model used is a three-dimensional multi-zone one. The fuel jet after injection is divided into discrete volumes (zones) in the three dimensions via a concentric consideration. The number of zones in the axial direction of fuel spray is determined by the duration of fuel injection and the time step used for the simulation. Each zone has its own temperature and composition. The first law of thermodynamics and the conservation equations of mass and momentum are used [Rakopoulos and Giakoumis (2006)] to calculate gas properties inside each zone. The pressure inside the engine cylinder is considered to be uniform [Fitzgeorge et al. (1962)]. The simulation covers the entire engine cycle, taking into account the gas exchange period using the “filling and emptying” technique.

HEAT TRANSFER

The characteristic velocity, necessary for the heat transfer calculations, is determined using a turbulent kinetic energy viscous dissipation rate $k\sim\varepsilon_t$ model [Rakopoulos et al. (1995)]. Then, the heat transfer coefficient is estimated from a widely tested correlation as described in [Annand (1963)]. The heat exchange rate obtained is distributed among the jet zones according to their mass, temperature and specific heat capacity [Rakopoulos and Hountalas (1998), Hountalas et al. (2002b)].

AIR SWIRL

A hybrid scheme consisting of a solid body core surrounded by a potential flow region is considered to describe the motion of air as it enters the engine cylinder from the intake valve [Dent and Derham (1974), Ramos (1989)].

SPRAY MODEL

After initiation of fuel injection, zones start to form and penetrate inside the combustion chamber. The initial conditions at the nozzle tip are obtained using the experimental injection rate. The velocity of each zone along the jet axis is obtained from correlations providing the penetration length of the fuel jet inside the cylinder. The zone velocity on the jet periphery is estimated using the methodology described in detail in [Rakopoulos and Hountalas (1998), Hountalas et al. (2004)], by consideration of their radial distance from the jet axis. The effect of air swirl upon the jet is also considered, using a calculation of the local components of the air velocity in the radial and axial directions and the momentum conservation equations in both axes.

AIR ENTRAINMENT

Estimation of the air entrainment is based on momentum conservation. The simulation was modified to consider for the effect of water on the fuel. The expression describing the momentum conservation in this case is the following:

$$
\begin{aligned}
&(m_f + m_w)u_{inj} = (m_w + m_f + m_\alpha)u_{spr} \Rightarrow \\
&\Rightarrow m_a = \frac{(m_f + m_w)u_{inj} - (m_w + m_f)u_{spr}}{u_{spr}} \Rightarrow \\
&\Rightarrow m_a = \underbrace{m_f\left(\frac{u_{inj}}{u_{spr}} - 1\right)}_{I} + \underbrace{m_w\left(\frac{u_{inj}}{u_{spr}} - 1\right)}_{II}
\end{aligned}
\tag{1}
$$

where m_w is the amount of water in the fuel. From the previous expression, it is revealed that using water/fuel emulsion the air entrainment into the fuel spray is increased. To make this more clear, in EQ(1) the term *(I)* corresponds to the mass of air that would be entrained into the fuel spray if only fuel was injected, while the term *(II)* corresponds to the increase of air entrainment due to the existence of water inside the fuel spray. Therefore, the momentum of the injected water is actually utilized to increase the specific air entrainment rate (i.e. per unit of fuel mass). This is expected to have a positive effect on combustion and soot oxidation mechanism.

DROPLET EVAPORATION AND BREAKUP

The injected fuel or emulsion of water/fuel is distributed to the zones according to the instantaneous injection rate [Siebers (1998)]. Inside each zone, a chi-squared distribution is used to describe the distribution of the fuel droplet diameter. The evaporation process of pure diesel fuel is modeled using the theoretical approach suggested in [Borman and Johnson (1962)]. As known, the presence of water in the mixture affects droplet size and evaporation rate. For droplet evaporation, a uniform mixture of water and fuel is considered in this preliminary investigation. The physical and thermodynamic properties of fuel/water mixture are estimated using the following relation:

$$PR = PR_w(1 - X) + PR_f X \tag{2}$$

where ‘PR’ is the value of the property and ‘X’ is the water percentage inside the mixture. Hence, the Sauter Mean Diameter (SMD) is estimated from the following relation:

$$
\begin{aligned}
D_{SM,1} &= 0.38\, Re_{inj}^{0.25}\, We_{inj}^{-0.32} \left(\frac{v_f}{v_\alpha}\right)^{0.37} \left(\frac{\rho_f}{\rho_\alpha}\right)^{-0.47} d_{inj} \\
D_{SM,2} &= 4.12\, Re_{inj}^{0.12}\, We_{inj}^{-0.75} \left(\frac{v_f}{v_\alpha}\right)^{0.54} \left(\frac{\rho_f}{\rho_\alpha}\right)^{0.18} d_{inj}
\end{aligned}
\tag{3}
$$

where subscripts “1” and “2” stand for complete and incomplete sprays, respectively. The Sauter mean diameter is taken to be the maximum of the above two values. As observed from EQ(3), the effect of added water on SMD is expressed through its effect on fuel/water mixture density and viscosity. Then, the model of Borman and Johnson (1962) is used to calculate the evaporation of emulsified fuel. Thermophysical properties of fuel/water blend employed in the evaporation model are estimated using EQ(2). A more realistic approach for describing the evaporation process of the fuel/water mixture would be based on bi-component evaporation. According to this, the evaporation rate of diesel fuel (represented by n-dodecane) and water would be calculated separately. However, due to the preliminary nature of the present analysis and the fact that for high pressure injection systems, as the present one, evaporation is extremely fast, this is not expected to have a serious effect on the overall results.

COMBUSTION MODEL

The amount of air entering a zone is considered to mix with the evaporated fuel, and the local reaction rate depends on the local concentration of fuel, oxygen and the local temperature. Ignition commences after an ignition delay period, which is estimated using the local conditions inside each zone [Hountalas et al. (2002), Nishida and Hiroyasu (1989), Kadota et al. (1976)].

FUEL INJECTION

In the present work, the injection pressure history was measured during the experiment and used as an input to the simulation. From this, the instantaneous injection rate of fuel can be calculated.

GAS EXCHANGE

The proven successful method of "filling and emptying" is used to calculate the mass exchange rate between the cylinder and inlet and exhaust manifolds [Benson and Whitehouse (1979), Heywood (1988), Hountalas et al. (2002), (2004)]. The simulation of the intake manifold is modified to cater for the air humidification. Hence, the inlet air temperature is estimated considering the enthalpy addition from the injected water and the partial evaporation (up to saturation) of injected water. A serious effort has been made to maintain inlet and exhaust manifold pressure at the same level for all cases examined, in order to have comparative results between the examined cases. Hence, the values corresponding to the reference case (engine operation without water addition) were considered.

SOOT FORMATION AND OXIDATION

The semi-empirical model of Hiroyasu and co-workers [Hiroyasu and Kadota (1976), Nishida and Hiroyasu (1989)] is used to predict the soot formation rate in each zone [Rakopoulos et al. (1995), Hountalas et al. (2002)]. The model is based on the assumption that the net soot formation rate is equal to the difference of soot formation and soot oxidation rates. According to [Nishida and Hiroyasu (1989)], both rates of soot formation and soot oxidation depend on local temperature and mean cylinder pressure. In addition, the rate of soot formation depends on the mass of evaporated fuel inside each zone, whereas the soot oxidation rate depends on the local partial pressure of oxygen and local soot concentration. In the present study, it is taken into account the contribution of OH radicals on soot oxidation rate in order to describe more accurately the effect of water addition on soot formation. Up to now, for cases without water addition, this has been neglected since oxygen has the most significant effect and the overall soot oxidation rate was adjusted via a proportionality factor. Obviously, this is not adequate for the case of water addition since the proportionality factor of soot oxidation once determined, at an operating case, is maintained as it is for all the other cases. The OH radicals play an important role in soot oxidation especially at higher temperatures, because they become the dominating oxidation means of soot, whereas the flame consumes oxygen in its molecular form [Hountalas et al. (2001)]. The addition of water to the fuel results to higher OH radical production, due to the following reactions:

$$\begin{aligned} H_2O + O &\rightarrow OH + OH \\ H_2O + H &\rightarrow OH + H_2 \end{aligned} \tag{4}$$

At the same time, OH concentration drop is expected because of its consumption during soot oxidation due to the reaction:

$$C_n + OH \rightarrow CO + C_{n-1} + H \tag{5}$$

The above results in reduced soot emission concentration in the combustion products.

FORMATION OF NITRIC OXIDE – Eleven chemical species are considered, calculated assuming the chemical equilibrium scheme of [Vickland et al. (1962)]. The formation of nitric oxide is controlled by chemical kinetics [Heywood (1988)]. For its calculation, the extended Zeldovich mechanism is used [Lavoie et al., (1970)]. The formation of NO is mainly temperature affected and thus the effect of water is mainly due to its influence on the local gas temperature.

ENGINE DESCRIPTION

The engine considered herein is a standard production heavy-duty, DI diesel engine. The engine is a supercharged one with a bore of 130 mm, a stroke of 150 mm and a compression ratio of 18:1. The engine is equipped with a common-rail fuel injection system.

MODEL VALIDATION

Before using the multi-zone model for the analysis of the effect of added water into the combustion chamber, it is necessary to calibrate and validate it. For the present application, we used a model calibration procedure that has been developed by the authors and has already been described in detail in [Pariotis et al. (2006)]. After this procedure is completed, the model constants are retained unchanged during the application of the model, at all engine operating conditions examined, with and without water addition. In the present study, the model's constants have been estimated using the aforementioned calibration procedure at 1800 rpm engine speed and full engine load with no water addition (reference engine operating point). The model was then validated against available experimental data for engine performance, soot and NO emissions, covering the whole range of engine's operating conditions without water addition. Theoretical results indicated a good agreement with the corresponding experimental data, but these are not presented herein for the sake of brevity of space of this paper.

WATER ADDITION STRATEGIES CONSIDERED

Three water addition strategies have been considered in the present work, and their effect on engine performance and pollutant emissions is comparatively evaluated. In Figure 1 the strategies followed are shown schematically, and their abbreviations used for each of them are given, i.e. IM, DI-A and DI-B. The abbreviation 'IM' stands for the first technique used to add water into the combustion chamber, which is injection of water in the Intake Manifold, while 'DI' corresponds to the second technique examined where water is Directly Injected into the combustion chamber where it partially evaporates up to saturation and mixes up with fresh air; with the rest of it evaporating during the compression stroke. For all water addition strategies followed, the engine brake power output has been kept constant and equal to that of the baseline case (with no water addition).

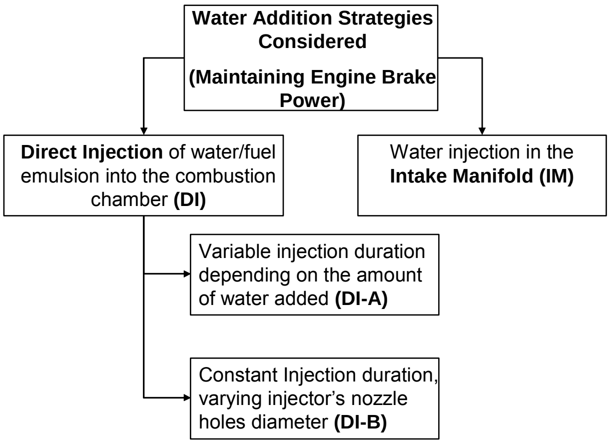

Figure 1. Water addition strategies considered.

When water/fuel emulsion is used, since the specific heating value of the mixture (fuel/water) is lower compared to the baseline case (pure diesel fuel), two alternative strategies have been investigated in order to keep the engine brake power constant as the water percentage increases.

According to the first one (named DI-A), the injection duration varies proportionally to the percentage of water added, while in the second one (named DI-B) the diameter of the injector's holes increase as the water percentage increases, keeping injection duration constant. Finally, when water is injected into the intake manifold, the injection mechanism is not affected, since the variation of injection duration required to maintain the engine brake power output is extremely low.

To comparatively evaluate the effect of each of these strategies on engine performance and emissions, the percentages of water added for all strategies are the same and correspond to 10%, 20% and 30% of the fuel mass injected in the baseline case (no water addition). In all cases examined, the injection timing is maintained the same as in the case with no water addition (reference case).

The computational study refers to an engine speed of 1800 rpm and four different loads, namely 25%, 50%, 75% and 100% of full load operation. In Table 1 the operating data for the aforementioned operating cases are given. As shown, when the DI-A strategy is used, injection duration is increased as the percentage of water increases, to obtain the same engine brake power output as for the reference case without water. Obviously, the injection duration at 100% load and

30% water percentage is unrealistic, i.e. extremely high, but it is still considered here for the sake of completeness. This also reveals the necessity for modifying the fuel injection system if high water percentages are to be used at full load operation, to obtain the same power output. This is required since the injection rate (of emulsion) is practically the same as for the case without water, because injection pressure is the same and the small difference in density cannot compensate for the decrease of net fuel flow rate (due to the presence of water).

Moreover, in Table 1 the diameters of the injector's nozzle holes are shown, which have been used for the computational study, when the DI-B strategy is followed. Comparing nozzle's hole diameters as water percentage increases between the various engine loads examined, it is noticed that the nozzle hole diameter remains almost unchanged at all engine loads examined and it only varies with water percentage. This remark is of special importance regarding the applicability of this strategy (DI-B) in real engine operating conditions since, having determined the optimum water percentage, there is no need to vary nozzle's holes diameters with engine operating conditions. Finally, as far as the strategy of water injection in the intake manifold (IM) is concerned, the engine operating conditions correspond to the ones without water injection, since in this case the injection mechanism is not affected.

Table 1. Engine operation cases considered in the present work.

Engine Mode	Engine Speed (rpm)	Engine Load (%)	Water (%)	Inj. Duration (^{o}CA) Water Addition Strategy DI-A	Inj. Hole Diam. (mm) Water Addition Strategy DI-B	Inlet Pressure (bar)	Inj. Pressure (bar)
B25	1800	25	0	11	0.134	1.90	900
			10	12	0.142		
			20	13	0.150		
			30	14	0.156		
B50		50	0	17	0.134	2.35	1200
			10	18	0.140		
			20	20	0.146		
			30	23	0.156		
B75		75	0	23	0.134	2.77	1300
			10	26	0.141		
			20	29	0.147		
			30	33	0.156		
B100		100	0	31	0.134	3.18	1400
			10	34	0.139		
			20	38	0.147		
			30	45	0.156		

COMPARATIVE EVALUATION OF WATER ADDITION STRATEGIES EXAMINED

The main scope of the present work is to comparatively evaluate the effect of three different water addition strategies applied on a heavy-duty supercharged DI diesel engine, as far as the combustion and pollutant formation mechanism is concerned. Through this computational study, a better understanding of how water addition affects engine performance and pollutant emissions is obtained, and some guidelines for the optimum technique that should be followed to reduce pollutant emissions without affecting seriously engine performance can be drawn. For the sake of brevity of space, in the following paragraphs the results are shown for the engine performance and emissions that correspond to the low and full load cases only.

SPECIFIC FUEL CONSUMPTION

Taking into account that for all the strategies examined the engine power output has been kept the same, the bsfc is a representative engine parameter that can be used to evaluate the effect of each water addition strategy followed onto engine performance. In Figure 2a the effect of the percentage of water added, for each of the three water-addition strategies considered at low engine load, is shown. As observed, using water emulsified fuel leads to a small reduction of the bsfc as the water percentage increases, a fact that is more pronounced for water percentage greater than 20%, while when water is injected into the intake manifold an opposite trend occurs.

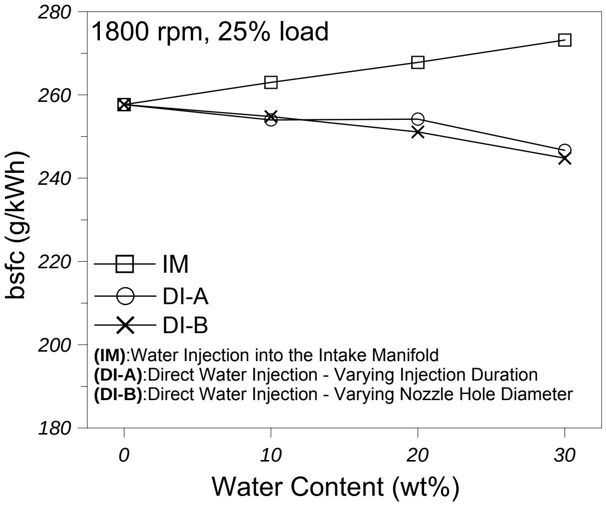

Figure 2a. Effect of water addition technique on the variation of calculated bsfc with water percentage, at 1800 rpm, 25% of full engine load

The slight increase of the bsfc when water is injected into the intake manifold can be attributed, mainly, to the replacement of a portion of the inlet air charge by water, leading to reduced oxygen availability inside the fuel spray. Taking into account that the amount of fuel injected into the cylinder is kept constant, it is concluded that the reduction of oxygen content leads to lower AFR (air-to-fuel ratio) inside the fuel spray region, which in turn affects adversely the combustion rate and consequently the bsfc. On the other hand, when water/fuel emulsion is used (strategies DI-A and DI-B), it improves the air-fuel mixing mechanism due to the additional momentum of the water injected inside the combustion chamber, which leads to higher air entrainment rates EQ(1) and consequently higher AFR inside the fuel spray and better bsfc.

At full engine load (Figure 2b), similar dependence of bsfc to the water percentage is observed as for the low load case, with the exception of DI-A strategy where an opposite trend is observed. As the percentage of water increases, the bsfc slightly increases. This can be attributed to the fact that, at full engine load, the injection duration increases considerably, shifting the combustion towards the expansion stroke. Therefore, it appears that irrespective of the strategy followed to add water into the combustion chamber, there is a slight effect on bsfc. Moreover, when water fuel emulsion is used, keeping injection duration constant (DI-B), the bsfc is slightly decreased as the water percentage increases at low and full engine load. However, since water addition is a technique used to reduce mainly pollutant emissions, it remains to be examined how exhaust tailpipe emission values (NO and Soot) are affected using the aforementioned water addition strategies.

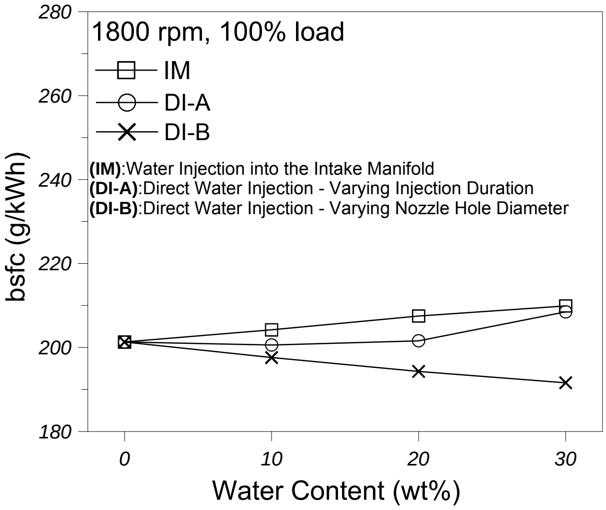

Figure 2b. Effect of water addition technique on the variation of calculated bsfc with water percentage, at 1800 rpm, 100% of full engine load

NO EXHAUST TAILPIPE EMISSIONS

According to Figure 3a,b, either water fuel emulsion or intake air humidification act beneficially on the reduction of NO tailpipe values at both low (Figure 3a) and high (Figure 3b) engine loads. This is because NO formation mechanism is highly dependent on local gas temperature, which lowers as the percentage of water added into the combustion chamber increases. At low engine load (Figure 3a) when DI-A strategy is used, this reduction of NO exhaust emissions is stronger for water contents between 10% and 20%, while a further increase of water content is not followed with a proportional decrease of NO emissions. On the other hand, at full engine load (Figure 3b), an almost linear reduction of NO tailpipe emissions is observed for all the strategies of water addition examined, as the water percentage is increased from 10% to 30%.

Furthermore, the effect of water addition on NO emissions is stronger at full engine load, compared to the corresponding one at low engine load. Based on the calculated results shown in Figures 3a,b, comparing the effectiveness of each of the water addition strategies examined for NO reduction, the intake air humidification has the strongest effect at low engine load, whereas, at full engine load (Figure 3b), the best strategy seems to be the usage of water-fuel emulsion by varying the injection duration, so as to keep the engine brake power output constant (DI-A).

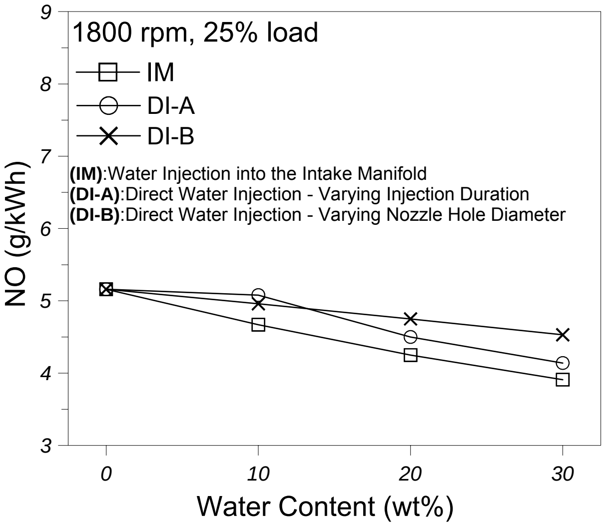

Figure 3a. Effect of water addition technique on the variation of calculated NO tailpipe value with water percentage at 1800 rpm, 25% of full engine load

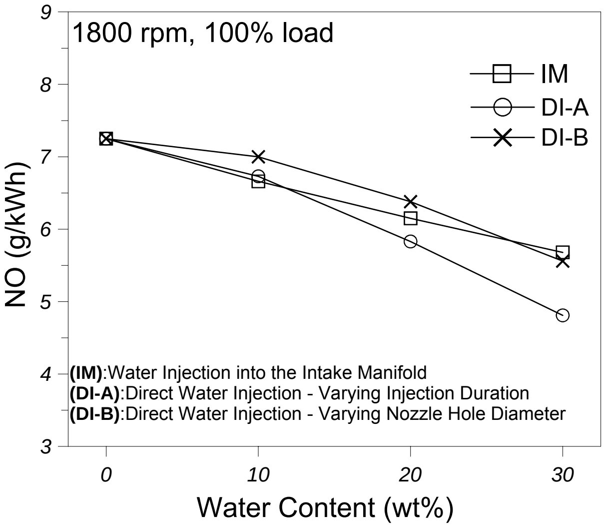

Figure 3b. Effect of water addition technique on the variation of calculated NO tailpipe value with water percentage at 1800 rpm, 100% of full engine load

SOOT EXHAUST TAILPIPE EMISSIONS

Since the mechanisms of NO and Soot formation are controversial, it is interesting to examine how soot tailpipe emissions are affected by water addition into the combustion chamber. In Figure 4a is shown the variation of soot exhaust tailpipe emission for the three water addition strategies examined, as the water content increases from 0% to 30% at low engine load. As observed, when water is added directly into the fuel spray (DI-A or DI-B), it leads to a reduction of soot tailpipe emissions, while when the intake air is humidified (IM) it has a strong negative effect (increase) on soot emission. The reduction of soot emission when water-fuel emulsion is used, is attributed mainly to the increase of the air entrainment rate and consequently of the AFR, as already described in EQ(1). This reduces the soot formation rate and results in higher soot oxidation due to the increased oxygen availability. On the other hand, when intake air humidification is used, the AFR inside the fuel spray is reduced compared to the baseline operating conditions (with no water addition), which results to a considerable increase of soot emission as the water content increases.

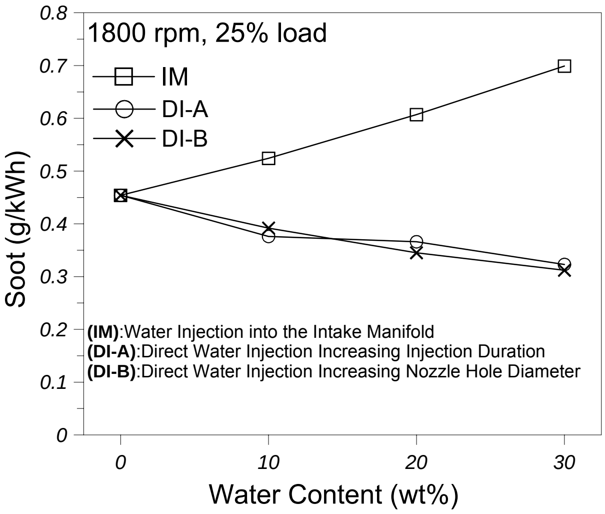

Figure 4a. Effect of water addition strategy on the variation of calculated Soot tailpipe exhaust value with water percentage, at 1800 rpm, 25% of full engine load

At full engine load, the corresponding effect of the variation of water content on soot emissions for all the water addition strategies examined is practically negligible, as shown in Figure 4b.

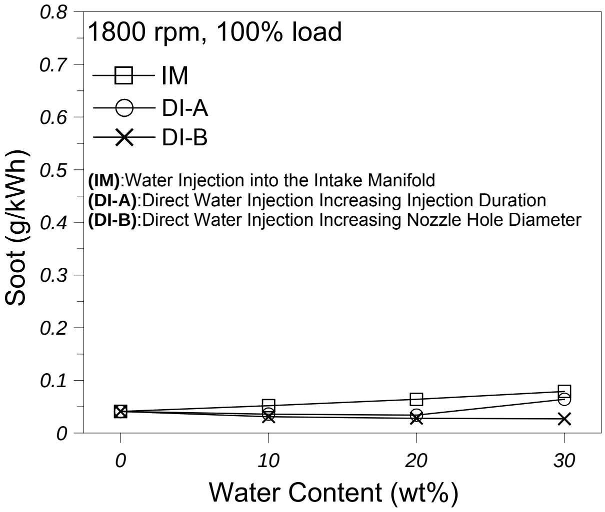

Figure 4b. Effect of water addition strategy on the variation of calculated Soot tailpipe value with water percentage at 1800 rpm, 100% of full engine load

Again here, the usage of water fuel emulsion leads to a reduction of soot emissions as water content increases, while the humidification of the intake air has a negative effect on them. An exception to the aforementioned trend is observed when the water content is higher than 20% and the DI-A strategy is used. In this case, the exhaust soot tailpipe emission increases. This is due to the considerable extension of the injection duration (Table 1), which leads to late soot formation during the expansion stroke that cannot be properly oxidized.

From the previous analysis is revealed that it is preferable to use water fuel emulsion (DI) as water addition strategy, since it results to a simultaneous reduction of NO and Soot exhaust emissions. However, when the DI-A strategy is used, its positive effect on the air fuel mixing mechanism and consequently on soot exhaust emissions is superseded by the impact of increased injection duration when the water content exceeds 20%, which is in accordance to what has already been reported in the literature [Song et al. (2000), Abu-Zaid et al. (2004), Ishida and Chen (1994)].

EFFECT OF WATER ADDITION ON BULK GAS TEMPERATURES AND POLLUTANT FORMATION HISTORIES

In the previous sections it was shown how the exhaust tailpipe emissions values are affected as water percentage is varied, for the three water addition strategies examined. In this section an attempt is made to explain the effect of water addition on NO and Soot formation mechanisms, examining the in-cylinder temperature and pollutant formation histories. For the brevity of space, only a limited number of characteristic cases are presented via which general conclusions can be drawn.

Figure 5 shows the calculated in-cylinder temperature history corresponding to the three strategies examined at low engine load and for 20% (wt) water content. When intake air humidification is used, the lowest temperatures are observed during the first part of combustion compared to the other two strategies examined, while this reduction is more evident at peak combustion temperatures. This is due to the fact that, using this strategy (IM), the total mass of water is added at the inlet valve closure, while when fuel emulsification is used, water is added gradually as fuel injection takes place. Thus, during the initial part of combustion, the mass of water inside the combustion chamber is higher when air humidification is used, resulting to lower bulk gas temperature. When water is injected directly into the combustion chamber together with the fuel, the higher temperatures are observed when injection duration remains constant (case DI-B). This is expected, since in this case the air entrainment rate is significantly boosted as shown by EQ(1), enhancing the combustion mechanism.

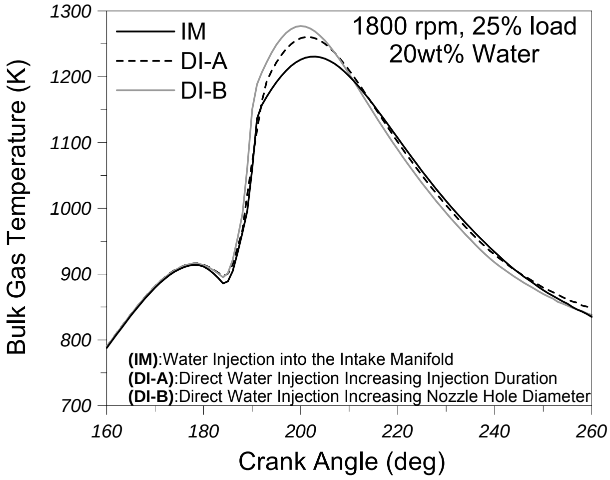

Figure 5. Calculated in-cylinder temperature history for the three water addition strategies examined at 1800 rpm, 25% of full engine load and 20% water content.

On the other hand, when injection duration varies (case DI-A), as the water percentage increases, combustion is shifted towards the expansion stroke, a fact explaining why lower temperatures are observed at this first part of combustion. Taking into account that almost all of the NO forms within the first 20 deg following the start of combustion [Heywood (1988)] and that temperature is the dominating factor affecting the NO formation mechanism, it is expected that the NO concentration inside the combustion chamber as well as its exhaust tailpipe values will follow the same trend as the temperature profiles shown in Figure 5. Indeed, as shown in Figure 3a, the classification of the exhaust tailpipe NO values are in accordance with the corresponding one as far as temperatures is concerned, during the initial part of combustion (Figure 5).

The temperature history at full engine load, with water percentage equal to 20% for the three water addition strategies examined, is shown in Figure 6a. As observed, during the first part of combustion (which is crucial for the NO formation mechanism), the lowest gas temperatures correspond to the DI-A strategy, the IM strategy comes next and finally the highest temperatures are observed when the DI-B strategy is applied. After this initial phase, the temperature classification is reversed. The in-cylinder gas temperature, when the DI-A strategy is followed, takes the highest values compared to the rest cases examined. This is due to the considerable increase of the injection duration, which shifts the combustion towards the expansion stroke.

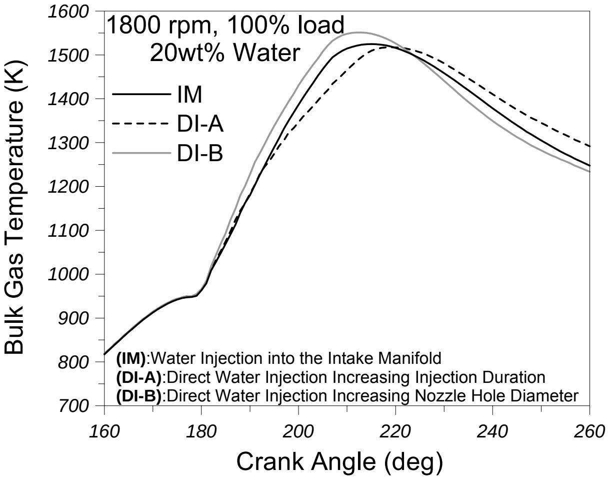

Figure 6a. Calculated in-cylinder temperature history for the three water addition strategies examined at 1800 rpm, 100% of full engine load and 20% water content.

Comparing the temperature profiles shown in Figure 6a, with the corresponding history of NO mass formation shown in Figure 6b, it is also observed that the in-cylinder NO concentration history and the exhaust tailpipe values for the three strategies examined follow the same classification as the one corresponding to the in-cylinder temperatures during the first part of combustion (Figure 6a). This confirms that the dominant factor controlling NO tailpipe exhaust emission values is the bulk gas temperature, especially during the first part of combustion, which is affected by adding water inside the combustion chamber.

As far as the soot mass history is concerned, in Figure 6c the soot (net) mass formation inside the combustion chamber is shown for the three water addition strategies examined, at full engine load and for water percentage equal to 20%. As observed, by adding water in the intake manifold (air humidification) results to significantly higher in-cylinder concentrations of soot compared to the other two strategies examined, where fuel is emulsified. This difference is mainly ascribed to the lower AFR inside the fuel spray when air humidification is used, which leads to increased soot formation rates.

Moreover, comparing the two strategies of water emulsification examined, it is observed that when the DI-B strategy is followed (constant injection duration), the in-cylinder peak soot concentration is reached earlier compared to the DI-A case. This is due to the fact that, in this case (DI-B), the amount of fuel injected in this first part of combustion (until peak in-cylinder soot concentration is reached) is higher compared to the DI-A case where injection duration is extended. Thus, higher concentration of fuel is expected leading to higher soot formation rates. However, the peak soot concentration and its final exhaust tailpipe value are lower compared to what is observed for the DI-A strategy, since the in-cylinder gas temperatures are higher (Figure 6a) due to the higher air entrainment rate obtained, so that the soot oxidation is enhanced.

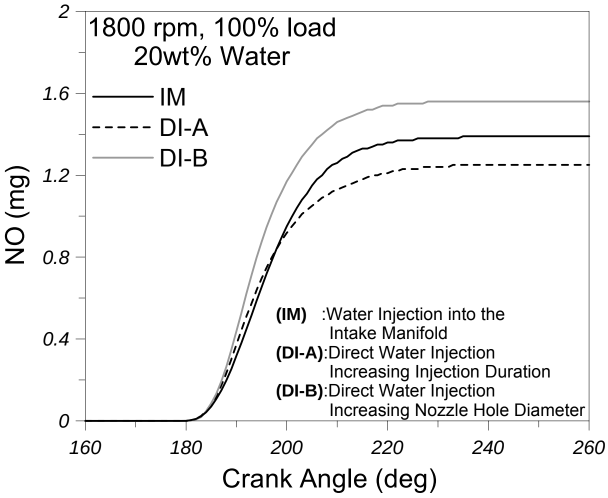

Figure 6b. Effect of water addition strategy on the in-cylinder NO mass formation history at 1800 rpm, full engine load and 20% water content.

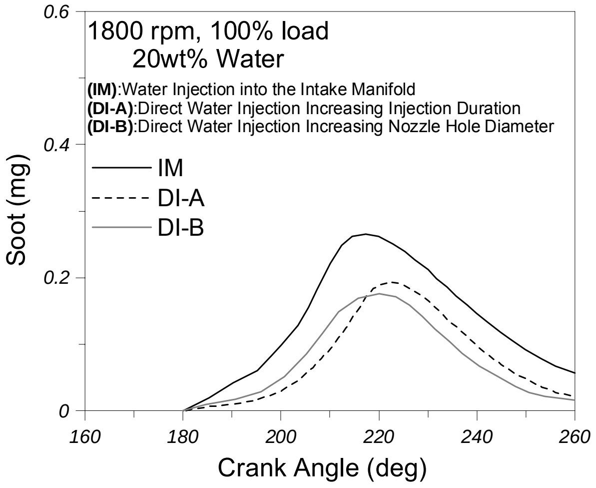

Figure 6c. Effect of water addition strategy on the in-cylinder Soot mass formation history at 1800 rpm, full engine load and 20% water content.

CONCLUSIONS

In the present study an attempt has been made to evaluate the use of water for the reduction of NO emissions in DI diesel engines. With reference to existing experience, two strategies were comparatively evaluated, viz. injection of emulsified fuel directly into the combustion chamber and injection of water into the engine intake manifold (air humidification). Two techniques have been examined for the water/fuel emulsification: The first one (called DI-A), where injection duration is varied as a function of water percentage to keep engine brake power constant and equal to the baseline case (with no water addition), and the second one (called DI-B), where fuel injection duration and brake engine power output are kept constant (and equal to the baseline case) for all water percentages examined, modifying appropriately the injector's holes diameter.

The simulation was based on a multi-zone model developed in the past by the authors, which was modified to account for water addition. Three different water content percentages were examined (10%, 20% and 30% by weight) at 1800 rpm engine speed and for engine loads ranging from 25% to 100% of full load. The main findings of the investigation can be summarized as follows:

- The introduction of water into the combustion chamber offers clear benefits towards the reduction of emitted NO levels. This reduction is more evident at high engine load and for water content percentages higher than 10%, when fuel emulsification is used, while when inlet air is humidified the effect of water percentage on NO emission is independent of load. The way that water addition mainly affects NO emissions is through the reduction of bulk gas temperature due to water evaporation, the increase of specific heat capacity of the cylinder content and dissociation of combustion products.

- The investigation revealed that the two main strategies for the introduction of water into the engine combustion chamber have opposite effects concerning the local AFR values inside the zones of the fuel jet. When water is added as an emulsion, it increases local AFR values due to the increased momentum of the injected fuel/water mixture spray. On the contrary, when air humidification is used, the local AFR in each zone of the fuel jet decreases (since a portion of the inlet air charge is replaced by the water added). This effect on AFR has a direct impact on bsfc and soot formation, favoring or opposing their values in comparison to the corresponding ones for the baseline engine operation (with no water addition). An exception regarding the effect of water addition on bsfc and soot emission is observed when fuel emulsification with varying injection duration (DI-A) is used, at full engine load. In this case, although the AFR is enhanced as the water percentage increases, bsfc and soot emission are increased. This is ascribed to the significant increase of the injection duration, which shifts combustion into the expansion stroke. Thus, in general, with water addition it is possible to reduce pollutant emissions (mainly NO) without affecting seriously engine performance (and even, in some cases, slightly improving it). This is very challenging, since with most conventional techniques used to reduce pollutant emissions, there is a penalty on bsfc or it is not possible to simultaneously reduce both NO and Soot emissions.

- As far as soot exhaust tailpipe emission is concerned, injecting water directly into the combustion chamber leads to reduced soot tailpipe exhaust values at both low and high engine load conditions. An exception to this trend is observed for the DI-A strategy at full engine load, when water percentage is equal to 30%. On the other hand, air humidification affects negatively the soot tailpipe exhaust emissions, especially at full engine load. From the theoretical analysis conducted, it has been revealed that water addition affects primarily the AFR inside the fuel spray region, which has a direct effect on the soot formation mechanism and to a lesser extent on the soot oxidation mechanism through the reduction of the bulk gas temperature. When fuel emulsification is used, the AFR increases leading to lower soot formation rates, while when air humidification is used, AFR decreases (since a portion of the air entrained into the fuel spray is replaced by water) enhancing soot formation rate.

- Comparatively evaluating the water addition strategies examined, it can be concluded that the usage of water emulsification with varying injection duration (DI-A) is beneficial from the perspective of NO reduction, since it offers a high reduction of NO emissions at low engine load and the highest one at full engine load where the problem is more intense. However, when water percentage exceeds 20% at full engine load, a slight increase is observed at bsfc and soot tailpipe emission, mainly due to the practically unacceptable increase of injection duration, which shifts combustion into the expansion stroke.

- Another interesting outcome of the computational study conducted is that, following water addition strategy DI-B (maintaining injection duration constant), it is possible to reduce bsfc, NO and Soot emissions simultaneously, although on a percentage basis the NO reduction achieved is significantly lower compared to the corresponding one following the DI-A strategy. However, for applications where the effect of water addition on bsfc is important and in order to avoid problems related to highly extended fuel injection duration at full engine load conditions, this strategy seems to be appropriate. Moreover, it is noticed that the required injector nozzle hole diameter, in order to keep the engine brake power output constant for each water percentage, is almost the same at all engine operating conditions. This is important since otherwise, this technique (DI-B) would not be feasible for practical applications.

- Finally, intake air humidification is beneficial to applications where modifications to the engine's fuel injection system should be avoided and where the consequent increase of soot tailpipe emissions can be faced up with an adequate after-treatment technique. However, attention should be paid to prevent possible adverse effects of the presence of water droplets on the engine structure (e.g. corrosion).

Concluding, through this extensive computational investigation conducted in the present work, it has been shown that the multi-zone simulation model used can be a valuable tool for engine design and development, by being suitable to provide reliable predictions of the engine performance and emissions in an affordable computational time, making thus parametric studies feasible.

NOMENCLATURE

d	diameter (m)
D_{SM}	Sauter mean diameter (m)
m	mass (kg)
u	speed (m/s)
v	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)

Subscripts

a	air
f	fuel
inj	injection, or injector
spr	spray
w	water

Abbreviations

AFR	air-to-fuel ratio
bsfc	brake specific fuel consumption
DI	direct injection
HD	heavy duty
Re	Reynolds Number
SMD	Sauter mean diameter
We	Weber Number

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