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Experimental study of inlet manifold water injection on combustion and emissions of an automotive direct injection Diesel engine

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ABSTRACT

This paper describes an experimental study conducted on a modern high speed common-rail automotive Diesel engine in order to evaluate the effects on combustion and pollutant emissions of water injected as a fine mist in the inlet manifold.

First, a literature survey describing the several ways to introduce water in an internal combustion engine and reporting the main results from previous studies is presented. It is followed by a short description of the engine and experimental set-up.

After that, various results are presented. A special focus is made on water injection (WI) cooling effect. Then, the influence of WI on ignition delay, rate of heat release, nitrogen oxides (NOx) and particulate matter (PM) emissions and engine efficiency is analysed, for various engine operating conditions (speed and load) and various amount of water (up to 4 times the amount of fuel injected). A comparison is made with exhaust gas recirculation to evaluate the potential of inlet WI as an in-cylinder emissions reduction device for automotive application.

1. Introduction

In light of the current requirements as regards the reduction of pollutant emissions of automotive Diesel engines such like EURO 6 in Europe, manufacturers have to develop new in-cylinder strategies and/or aftertreatment devices [1]. With the upcoming pollutant emission regulations, nitrogen oxides (NOx) emissions will become particularly critical on automotive Diesel engines.

Exhaust gas recirculation (EGR) into the engine intake is the most used and studied technology as regards the in-cylinder strategies aiming at reducing NOx emissions. The decrease of NOx emissions with EGR is the result of complex and sometimes opposite phenomena occurring during combustion [2–15]. The main effect is the decrease of local temperatures in the combustion chamber, in particular those corresponding to zones where NO is produced (on the lean side of the diffusion flame during fuel injection [16] and in the combustion products after the end of injection). The main drawback of EGR is the increase of particulate matter (PM) emissions in the classical high temperature Diesel combustion and the need to increase boost pressure at middle and high loads when using EGR while maintaining air–fuel ratio (AFR) at a suitable level.

Another in-cylinder strategy to reduce local temperatures and consequently the NO production rate is the injection of water (WI), either into the engine inlet [17–24], directly in the combustion chamber with the fuel injector [23–27,30], with a separate injection system [30,31], or in emulsion with the fuel [18,19,21–23,33–46]. One advantage of WI as compared with EGR is the possible reduction of NOx emissions either at low loads and high loads without a substantial increase in PM emissions.

1.1. Inlet WI [17–24]

The probably easier way to inject water in the engine is inlet WI [17–24]. This technique has been largely used on large marine Diesel engines [18]. Various strategies to inject water in the inlet air are presented in the literature: multipoint WI in the intake pipes close to the inlet valves [17–20], single point WI upstream the compressor [19,21] or downstream the compressor [21,22]. In all cases, the water-to-fuel ratio was increased of up to maximum 50%. By comparing multipoint WI in the intake pipes and single point WI upstream the compressor on a 7118 cm^3 four-cylinder air cooled DI Diesel engine, Samec et al. [19] found that both WI strategies show practically the same NOx emissions reduction (at the maximum power engine operating condition). Regarding engine thermal loading, better results have been obtained applying the second WI system [19]. Most studies show a minor or only slight increase in

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Nomenclature

Dr	dilution ratio (%)
$\dot{m}$	mass flow rate (kg/h)
NOx	nitrogen oxides (g/h)
P	pressure (bar)
PM	particulate matter (g/h)
T	temperature (°C)
X_{egr}	EGR rate (%)
X_{mass}	mass percentage in mixture (%)
λ	air excess (−)

Subscripts

1	atmospheric conditions
2	post-compressor
2′	post-intercooler
2″	post-EGR and air/spray mixer
3	exhaust − pre-turbine
4	post-turbine
5	post-DOC No 1
6	post-DPF
air	air
O_{2inlet}	oxygen at inlet
O_{2ref}	atmospheric oxygen (reference)
a/s_mixer	air/spray mixer
rail	fuel rail

Abbreviations

BMEP	brake mean effective pressure
BSFC	brake specific fuel consumption
CA	crank angle (°)
CO	carbon monoxide
DOC	diesel oxidising catalysts
DPF	diesel particulate filter
EGR	exhaust gas recirculation
FSN	filter smoke number
HC	hydrocarbon
HP	high pressure
HSDI	high speed direct injection
ID	ignition delay
IVC	inlet valve closure
LP	low pressure
LTC	low temperature combustion
NEDC	new European driving cycle
PM	particulate matter
ROHR	rate of heat release (W)
SOC	start of combustion
SOI	start of injection
TDC	top dead centre
VGT	variable geometry turbine
WI	water injection

PM emissions [17,19,22], and a significant increase in CO and HC emissions while increasing the water quantity [17,22].

1.2. Direct WI [25–32]

Different strategies have been proposed to inject water directly in the combustion chamber, with the aim at reducing NOx emissions while limiting the water quantity as compared with inlet WI. For instance, Mitsubishi Heavy Industries have developed a stratified fuel–water injection (SFWI) system [25]. In this method, fuel oil and water are injected from an injection valve in the form of stratification in the order of fuel oil–water–fuel oil [25]. Owing to this direct WI strategy, the ignition delay can be maintained at a low level even if a large quantity of water is injected and great NOx reduction is obtained [25]. One advantage of direct WI as compared with water-in-diesel emulsion is the possibility to change the water-to-fuel ratio, while varying engine parameters (speed and load) or during engine warm-up (cold start) [27–29]. Stanglmaier et al. [28] have tested a system for coinjecting mixtures of diesel fuel and water into a heavy-duty Diesel engine developed at the Southwest Research Institute. This system allows varying the percentage of water in the mixture on a cycle-resolved basis. With this system, Stanglmaier et al. [28] show a considerable improvement in both NOx and PM emissions, both under steady-state and transient conditions. The control of water percentage on a cycle-resolved basis was shown to be an effective method for mitigating NOx and smoke emissions over step-load transients.

1.3. Water-in-diesel emulsion and microemulsion [18,19,21–23,33–46]

Most engine experiments and numerical studies using water-in-diesel emulsion technique show that the NOx reduction is accompanied with a large reduction of PM and soot emissions. This decrease of PM emissions with the use of water-in-diesel emulsion has been explained as follows:

- At a given fuel injection rate, the use of water-in-diesel emulsion leads to an increase of the total injected mass, of which a consequence is an increase of the mixing rate between fuel and air, thus increasing local AFRs and consequently reducing PM production.
- The vaporisation of water and the dilution effect of water lead to a decrease of the temperatures within the core spray where soot is produced, thus reducing the soot production rate [34]. Indeed, soot production rate increases with temperature [50].
- The increased presence of water within the combustion jet may affect the chemical kinetic mechanisms of soot formation (in the core spray) and soot oxidation (at the jet periphery) [34,46].
- The presence of water in the emulsion has a tendency to increase the ignition delay. Thus, the mass fraction of fuel that burns under a premixed combustion is increased, of which a direct consequence is the decrease in soot production rate [35,37,38,45].
- The reaction zone of a diesel fuel jet stabilizes at a location downstream of the fuel injector once the initial autoignition phase is over. This distance is referred to as flame lift-off length [47]. It has been shown that, for diesel fuel jet, this parameter plays an important role in the soot formation process, by allowing fuel and air to mix upstream of the lift-off length (i.e. prior any combustion) [45–48]. Just downstream of the lift-off length, the partially premixed air–fuel mixture undergoes a premixed combustion that generates a significant local heat release and fuel-rich product gas that becomes the ‘fuel’ for the diffusion flame at the jet periphery. The soot formation was shown to be directly dependant on the equivalent fuel–air ratio at the lift-off length [47–50]. Local studies in an optically-accessible engine show that that flame lift-off length is significantly increased with water-in-diesel emulsion, leading to leaner mixtures within the jet during the diffusion stage of combustion [38].
- The water droplets contained in the emulsion have a lower and sharply defined boiling point. According to some researchers,

the sudden and dramatic expansion of vaporising water (called micro-explosion) would enhance the mixing process between air and fuel [21,42–44].

As mentioned above, the most efficient WI technologies to reduce NOx emissions are the use of water-in-diesel emulsion or direct WI because the water is injected directly into the combustion zone, allowing a large decrease of combustion temperature. As a consequence, at a given quantity of injected water, the NOx reduction with direct WI or water-in-diesel emulsion is around twice as high as with inlet WI [18,23]. On the other hand, theses techniques have some drawbacks:

- Complex implementation and thus increased engine cost for the direct WI technique.
- Need of a more extended and developed distribution network of water-in-diesel emulsion, or implementation of a complex on-board water-in-diesel emulsion production system on the engine, also increasing the engine cost.

As a matter of fact, the aim of this study is to investigate the capability of a relative simple-to-implement inlet WI system to reduce NOx emissions on a modern automotive common-rail DI Diesel engine for future emissions standards. Also studied are the effects of inlet WI on mean in-cylinder temperature at the end of the compression stroke, ignition delay, combustion, PM emissions, and brake specific fuel consumption (BSFC).

2. Experimental apparatus and procedure

2.1. Description of the engine

The engine used for the experiment is a 2.0 l water-cooled HSDI Diesel engine which conforms to Euro IV standards. It is equipped with a Variable Geometry Turbine (VGT) turbocharger, an intercooler, two Diesel Oxidising Catalysts (DOC) and a Diesel Particulate Filter (DPF) (see Fig. 1). Engine specifications are given in Table 1.

2.2. Operating points

The study is conducted for different operating conditions at low and part load, such as those encountered in most normalized automotive driving cycles (NEDC in Europe). The engine speed, torque, pilot and main injection quantities, brake mean effective pressure (BMEP) and rail pressure are given in Table 2. Injection quantities are held constant for each operating point. The changes in BMEP are thus a result of the various tested modifications (inlet temperature, WI rate, EGR rate, etc.).

2.3. Modifications of EGR system

The engine was originally equipped with a High Pressure (HP) EGR loop. A Low Pressure (LP) EGR loop has been added to the engine in order to reach high levels of recirculation (see Fig. 1). The recirculated gases are taken downstream of the DPF to avoid the fouling of the compressor and intercooler with soot. To avoid uncontrolled condensation downstream the introduction of recirculated gases, water from the recirculated gases is removed owing to a large LP EGR cooler that maintains the EGR temperature T_{egr_LP} under 50 °C. These are thus dry and free of PM. Condensed water is separated by gravity and collected in a catch tank. A valve has been added at the end of the exhaust line to control exhaust back pressure and thus permit high rates of EGR using the LP EGR loop.

In this study, only the LP loop is used to compare effects of WI and EGR.

The original air/air intercooler was turned into a water/air intercooler allowing the air temperature T_2' and/or T_2'' to be controlled separately.

2.4. WI system

The WI system was designed in order to inject large amounts of water while ensuring a (to a high degree) homogenised mixture of air and water droplets. The system is installed after the intercooler just before the inlet manifold (see Fig. 1). It comprises the following components:

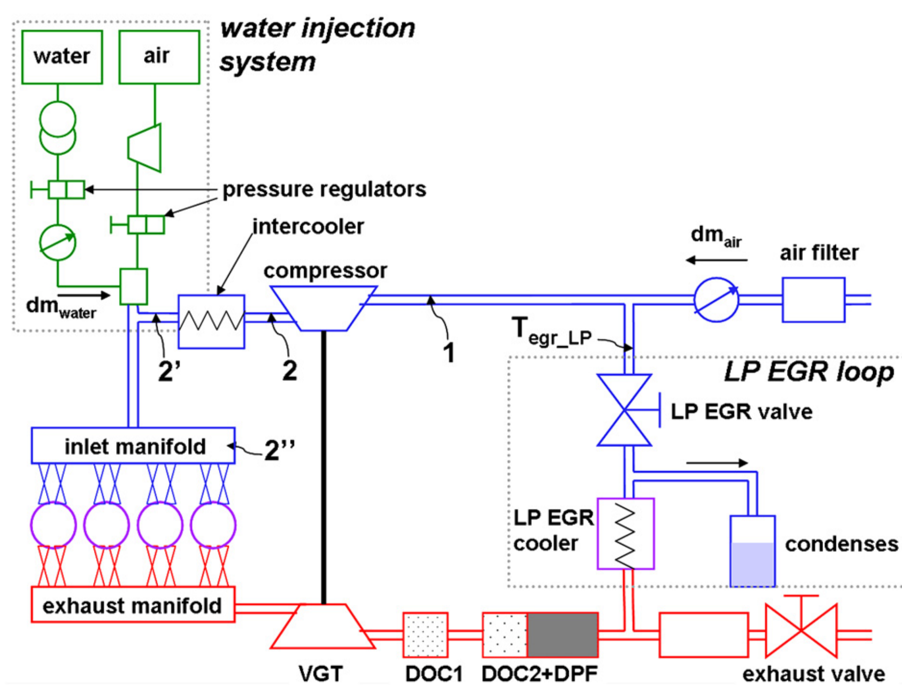

Fig. 1. Engine configuration.

Table 1
Engine specifications.

Compression ratio	18:1
Number of cylinders	4
Number of valves per cylinder	4
Combustion chamber	Re-entrant bowl-in-piston
Injection system	Common-rail piezoelectric 2nd generation
Maximum injection pressure	1600 bar
Number of injection holes	7
Injector nozzle diameter	0.150 mm

- Kärcher K2.01 High Pressure water pump,
- 8 bar air compressor,
- Parker Pneumatic 8 bar air pressure regulator,
- Radiospares Turbine Flowmeter.

The air/water injector consists of a modified petrol injector. Compressed air (supplied by an external compressor at a pressure from 5 to 6 bar) is used to atomise water and produce a fine mist. The air quantity used for this purpose was measured independently and always remains below 2% of total air mass flow aspirated by the engine. However, it was taken into account in charge mass calculation. The size of the droplets depends on the water flow rate, an increase in flow rate increasing the size of the water droplets. Photographs show the quality of the mist during system operation (Figs. 2 and 3).

2.5. Evaluation of dilution ratio

Generally, the dilution ratio can be defined as:

$$\mathrm{Dr} = 1 - \frac{X_{\mathrm{mass}}\mathrm{O}_{2\mathrm{inlet}}}{X_{\mathrm{mass}}\mathrm{O}_{2\mathrm{ref}}} \tag{1}$$

In the case of EGR the mass percentages of CO_2 and O_2 are calculated from the molar (volumetric) concentration percentages that are measured by the gas analyser.

In the case of WI the dilution ratio is defined using the mass flow rate of the intake charge and the mass flow rate of the diluent present in it and can be defined as:

$$X_{\mathrm{mass}}\mathrm{O}_{2\_\mathrm{a/s\_mixer}} = \frac{\dot{m}_{\mathrm{air}}}{\dot{m}_{\mathrm{air}} + \dot{m}_{\mathrm{WI}} \cdot X_{\mathrm{mass}}\mathrm{O}_{2\mathrm{ref}}} \tag{2}$$

$$\mathrm{Dr} = 1 - \frac{X_{\mathrm{mass}}\mathrm{O}_{2\_\mathrm{a/s\_mixer}}}{X_{\mathrm{mass}}\mathrm{O}_{2\mathrm{ref}}} \tag{3}$$

2.6. Evaluation of the mean gross rate of heat release (ROHR)

The gross ROHR is obtained for each operating condition thanks to a calculation procedure developed at the laboratory. The calculation is classically based on the in-cylinder pressure, measured with a Kistler 6055BB piezoelectric pressure transducer and an

Table 2
Operating points.

Point	A	B	C	D
Engine speed (rpm)	1500	1665	2050	2000
Torque (N m)	45	114	140	200
Pilot quantity (mg/stroke)	1.2	1.5	1.7	2.4
Principal quantity (mg/stroke)	11.2	22.8	30.7	39.7
BMEP (bar)	2.8	7.1	9.5	12.7
P_{rail} (bar)	600	865	1028	1154

Fig. 2. Spray quality of the injector.

encoder with a resolution of 0.36 CA degree. The cylinder pressure used is the mean value over 100 consecutive cycles, which was found to be enough for the mean value to be reliable. It must be underlined that the mean ROHR gives no indication on eventual cycle to cycle dispersions. The gross ROHR was extracted from the net ROHR by calculating the heat transfer to the combustion chamber walls with Hohenberg's model [51].

Since valve overlap is negative on this engine, trapped mass of fresh air can be deduced directly from air mass flow measured at the engine inlet and eventually air mass flow used for WI (see Section 2.4). The mass of residual gas is calculated with perfect gas law applied at EVC, from volume, measured in-cylinder pressure and in-cylinder temperature estimated from exhaust gas temperature. The residual gas composition is deduced from exhaust gas analysis. When EGR is used, EGR mass flow is calculated from CO_2 concentrations measurements at the engine inlet and engine exhaust (that gives the ratio between EGR mass flow and total (air + EGR) mass flow, the air mass flow being measured at the engine inlet). EGR composition is deduced from exhaust gas analysis. When WI is used, the trapped mass of water is evaluated from water flow measurement. The in-cylinder temperature at inlet valve closure (IVC) is deduced from trapped mass, volume at IVC and in-cylinder pressure measurement (with perfect gas law for all gases except water for which steam table are used). This temperature is used to verify that the air is still unsaturated while all injected water has evaporated during intake stroke.

Fig. 3. Spray quality of the air/spray mixer.

2.7. Evaluation of fuel proportion injected during ignition delay

In order to interpret NOx and soot emissions it can be helpful to know the proportion of fuel that burns during the premixed phase and that which burns during the diffusion phase. In fact, it is almost impossible to access directly this proportion. Thus, the proportion r of fuel injected during ignition delay was calculated for each operating condition tested (operating point, water quantity, EGR rate, inlet temperature,…). The injection rates were first measured for each operating point on an injection test bench, thus providing the instantaneous proportion of injected fuel at any time after start of injection (SOI), in particular at start of combustion (SOC). The ignition delay and thus the SOC are deduced from ROHR diagrams. The ignition delay is supposed to end when combustion acceleration (differential of ROHR, in W/s) reaches a critical value, which was arbitrarily fixed at 2×10^8 W/s. The ratio of injected fuel at SOC over the total amount of fuel injected at each cycle is given in brackets on ROHR diagrams for each operating condition tested. It must be stressed that a part of the fuel injected during ignition delay has not yet mixed with air at SOC and will thus not undergo a premixed combustion. As a consequence, the ratio *r* can be considered as an upper bound of the proportion of fuel that burns under a premixed mode.

2.8. Emissions measurement

NOx emissions are measured with an ECO PHYSICS CLD 700EL gas analyser, which uses the chemical luminescence detector (CLD) method. PM emissions at the exhaust are measured with an AVL 415S smoke-meter. Inlet O_2 and CO_2 concentrations as well as CO emissions are measured with a SIEMENS ULTRAMAT 23 gas analyser, which uses the non-dispersive infrared (NDIR) measurement technique. Each gas analyser is calibrated every 4 h of experiments with specific gas standards. If the necessary shift is under 0.3%, then the experiments done since the previous calibration are validated. The conversion of emissions from ppm to g h^{-1} and the calculation of PM emissions in g h^{-1} starting from the filter smoke number (FSN) are presented in a previous paper [5].

2.9. Error analysis

Table 3 sums up the measurement technique, calibrated range, accuracy and relative error of various instruments involved in the experiment for various parameters. Errors in experiments can arise from instrument conditions, calibration, environment, observation, reading and test planning. The accuracy of the experiments has to be validated with an error analysis. That was performed here using the method of differential method of propagating errors based on Taylor' theorem. It gives the maximum error *u* of a function $f(x_1, x_2,\ldots,x_n)$ as follows:

$$u(f(x_1, x_2, \ldots, x_n)) = \sqrt{\sum (c_i \cdot u(x_i))^2} \quad (4)$$

As a result, the maximum relative errors for X_{EGR}, NOx (g/h), PM (g/h), and λ are 1.4%, 1.5%, 2.3%, and 1.05% respectively.

Table 3
Relative measurement error.

Instrument	Calibrated range	Accuracy	Relative error
Inlet gas temperature (k-type thermocouple)	0–1000 °C	±1 °C	±0.75%
Inlet gas pressure (2 bar piezoresistive relative pressure sensor HCS Sensor Technics)	0–2 bar	±5 mbar	±0.25%
Air mass flow (hot wire air flowmeter)	0–1000 mg/str	±5 mg/str	1%
Water mass flow (Radiospares Turbine Flowmeter)	3–30 kg/h	±0.3 kg/h	2%
Fuel consumption (PIERBURG PLU 401/121)	0.05–23 kg/h	±37 g/h	±0.16%
NOx (ECO PHYSICS CLD 700EL)	0–1000 ppm	±5 ppm	1%
Smoke (AVL 415S)	0–10 FSN	±0.1 FSN	2%
Inlet O_2 (SIEMENS ULTRAMAT 23)	0–25%	±0.025%	0.2%
Engine torque	0–700 N m	±3.5 N m	1%
Engine speed	0–5500 rpm	±5 rpm	0.2%
In-cylinder pressure (Kistler 6055BB)	0–200 bar	±0.5 bar	1%

3. Cooling effect of inlet WI in the inlet manifold

A well known effect of WI into the intake air is cooling of the intake charge. This effect has been used for several decades to increase the power and prevent knock in spark-ignited engines for specific applications such as military airplanes or racing cars. Here, a test was conducted to try to isolate this cooling effect and evaluate its consequences on combustion and emissions. For operating point C, a first test (C1) was carried out by de-activating the intercooler (see Table 4). In that case, the inlet temperature reaches 91 °C. Then a 3 kg/h flow of water at 20 °C was injected into the intake manifold, causing the intake charge air temperature T_2'' to fall to 60 °C (configuration C2). An energy balance (between the hot charge air and the water) indicates that the inlet charge remains unsaturated after the water is injected and confirms that the temperature drop measured in the engine intake corresponds to the whole quantity of the injected water being evaporated. Finally the same temperature T_2'' was obtained using the intercooler, without WI, for comparison purposes (configuration C3).

3.1. Influence of inlet temperature on combustion

The ROHR curves are given in Fig. 4. The difference between the three configurations is very slight. Indeed, since a pilot injection is used during these tests, the main combustion is almost purely diffusive: owing to pilot combustion, the ignition delay and amount of fuel injected before SOC are drastically reduced (no more than 3%). For this kind of combustion, intake air temperature has little influence on ROHR. Temperature reduction increases in-cylinder air density and thus theoretically enhances air/fuel mixing (a given volume of gas entrained by fuel spray contains a bigger mass of air) but this can be partly compensated by a slower spray penetration, so that diffusive combustion speed only increases slightly. Normally, diffusive combustion speed should be reduced for configuration C2 compared to C3 due to charge dilution by water. However, the water flow rate is moderated for this test, and dilution is kept to a low level, as indicated by λ variations in Table 4.

Further in the paper, higher rates of WI are presented, leading to a greater influence on the combustion process.

3.2. Influence of inlet temperature on NOx and PM emissions

The corresponding influence is summarized in Table 4. Compared to basic configuration (C1), NOx emissions are reduced

Table 4
Comparison of cooling effect of WI and the intercooler.

Configuration	C1	C2	C3
Cooling method	–	WI (3 kg/h)	Intercooler
P_2'' (mbar)	1800	1800	1800
T_2'' (°C)	91	60	60
NOx (g/kg fuel)	27.1	17.8	24.6
PM (g/kg fuel)	0.28	0.35	0.26
Rel. NOx reduction	–	34%	9%
Rel. PM increase	–	26%	−8%
λ	1.78	1.82	1.86

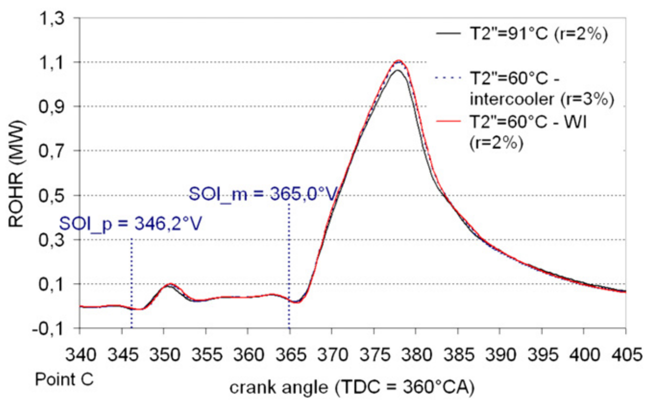

Fig. 4. Comparison of cooling effect of WI and the intercooler – influence on ROHR.

by 34% with WI (C2). The reduction obtained with intake air cooled by the intercooler (C3) is only of 9%. Thus the cooling effect of WI is only responsible for around 30% of NOx reduction, the other 70% being attributed to other effects of WI, such as dilution.

Concerning PM emissions, a 26% increase is observed between the basic configuration (C1) and WI (C2). However, the same T_2'' reduction obtained without WI (C3) can reduce PM emissions by 8%. This means that the potential PM reduction due to the cooling effect of WI is overcome by other effects of WI which finally result in a PM increase (decrease of flame temperature that results in a decrease of soot oxidation at the jet periphery, variation in the global AFR that influences soot oxidation as well, variation of flame lift-off length and corresponding AFR as well as core spray temperature that both affect the soot production rate in the core spray [47–50,52,53]). These various effects of WI on PM emissions are discussed in more details in Section 4.2. Some local experiments in the combustion chamber (on an optical Diesel engine) would help to understand the various effects of WI on PM production and oxidation into the combustion chamber.

It is important to note that these figures are indicative and the relative part of charge air cooling among the various effects of WI may vary depending on WI flow rate or engine operating conditions.

4. Influence of WI at a constant inlet air temperature

Four different operating points (A, B, C and D) have been studied.

4.1. Influence of WI on combustion

4.1.1. Influence of WI on ignition delay (ID)

The ROHR curves for operating points A to D are given in Fig. 5. The influence of WI on ignition delay for main injection is given in Fig. 6. As depicted in Figs. 5 and 6, the very first effect of WI on combustion is the increase in ignition delay. This is in agreement with previous researches [20,22]. This can be observed for both pilot injection and main injection (Fig. 5). The greater the quantity of water used, the greater is the ignition delay. For instance, for operating point B, the ignition delay is increased from 0.23 ms to 0.82 ms while increasing the dilution ratio from 0% to 10.2% (Fig. 6). This evolution can be explained easily by the cooling effect of water since air temperature is known to have a major influence on ignition delay. Fig. 7 shows the in-cylinder average temperature deduced from the in-cylinder pressure measurement, for operating points B and D. WI reduces the in-cylinder temperature at SOC of about 100 K for the maximum water flow rate, 21 kg/h corresponding to the highest dilution ratio in Fig. 7. Water may also have a chemical and/or diluting effect on ID. To study such effects, combustion with WI should be compared to combustion without WI but with the same in-cylinder temperature at SOI. This was performed above (Section 3) but with very little amount of water, no noticeable effect of water on ID (except for cooling effect) was observed. A comparison with a higher WI rate would be interesting. But it is hard to perform practically because it would require a very low air temperature at cylinder inlet (T_2'') which is difficult to attain with a classical intercooler.

4.1.2. Influence of WI on ROHR

The ID increase has a direct influence on combustion: the proportion r of fuel injected during ID is increased while increasing WI rate, of which a consequence is an increase in the premixed part of combustion. For instance, for operating point A, r is increased from 40% to 57% while increasing the dilution ratio from 0% to 6.2%. The higher the engine load, lower will be the ratio r (for a given dilution ratio). For the highest load tested here (operating point D), r is increased from 3% to 11% while increasing the dilution ratio from 0% to 13.0%.

The first peak of heat release is little delayed but becomes higher with moderate amounts of WI. This is very noticeable at low load, when injection duration is shorter and thus premixed part has a greater relative importance (operating points A and B, see Fig. 5a and b respectively). However, when in-cylinder temperatures become too low, premixed combustion speed may decrease due to chemical reactions which are slowed down, resulting in a decrease of the total ROHR (premixed + diffusion combustions). This can be observed for operating point A (see Fig. 5a), as well as for operating point B when large amount of WI are used (dilution ratio above 6.8%, see Fig. 5b).

At higher loads, the combustion is almost purely diffusive, the proportion of fuel injected during ID being very low. In that case, the influence of WI on ROHR is negligible. Indeed, diffusive combustion speed is governed by the amount of air entrained by the fuel spray per unit of time. When WI is used, the spray entrains a water–air mixture instead of pure air, so that a decrease in combustion speed could be expected. Such a decrease is observed for diffusive combustion with EGR when EGR rate is increased at constant boost pressure [5,6]. In the case of WI however, water does not replace air but is added to it: even if boost pressure is kept constant, the cooling effect produced by water vaporisation increases the density of in-cylinder gas content. In other words, when WI is used, assuming that fuel spray entrains the same volume of gas per unit of time than without WI, this volume contains a greater mass of gas (air + water vapour) and almost the same mass of air, so that combustion speed does not change noticeably.

4.1.3. Influence of WI on in-cylinder pressure and engine efficiency

When WI deeply modifies the ROHR, it follows that in-cylinder pressure traces are also altered. As a result, the engine torque is reduced when the combustion is delayed (the fuel mass injected being kept constant). As depicted in Fig. 10, this engine global efficiency reduction is very marked at low and middle load conditions (operating points A, B, and C).

For operating point A (see Fig. 8a), for which WI rate is limited by combustion stability (for a dilution ratio above 6.2%, corresponding to a water flow of 6 kg/h), combustion quality is deteriorated with production of large amounts of CO and HC and even some misfired cycles. The engine efficiency is decreased from 27.5% to 22.9% while increasing dilution ratio from 0% to 6.2%.

At higher loads, WI has very little influence on ROHR and thus on in-cylinder pressure (see Fig. 8b for operating point D).

4.1.4. Influence of WI on in-cylinder temperature

Considering in-cylinder average temperature after SOC (Fig. 7), as expected, the greater the mass of water injected, the lower is the

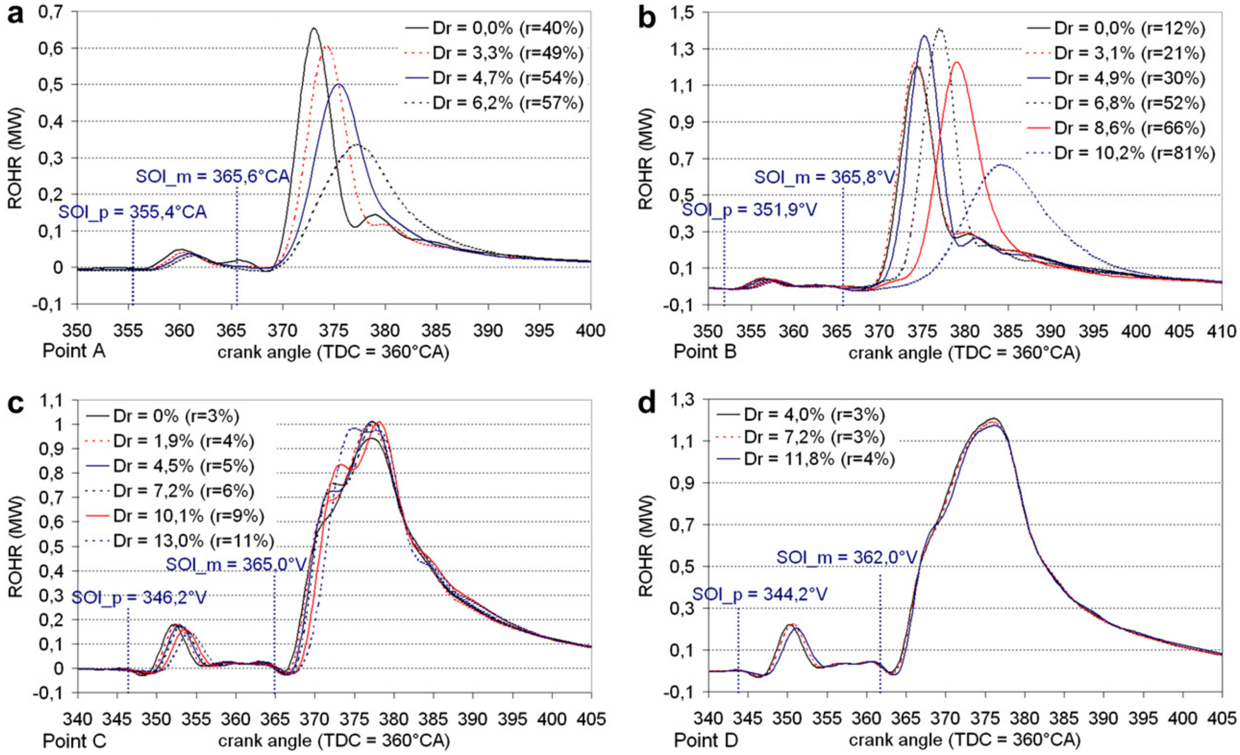

Fig. 5. ROHR for various WI rates.

temperature. However, the temperature difference between WI and the reference case increases during combustion (it may reach 200 K against 100 K at SOC), in particular at high dilution ratio (high rates of WI). Several explanations can be proposed. First, the trapped mass increases with WI rates (boost pressure is kept constant and temperature at IVC decreases due to water vaporisation) thus the in-cylinder heat capacity is greater. ROHR modifications (combustion is delayed with WI) also alter in-cylinder temperature evolutions.

A third reason appeared during ROHR analysis. In order to ensure a correct cumulative heat release it was found that it is necessary to increase heat losses at cylinder walls (all the more since a higher WI rate is used). Fig. 9 depicts the calculated cumulative combustion gross heat release while varying WI rate for operating point B. The final value decreases with WI whereas CO, HC and PM emissions show that combustion efficiency is not noticeably altered. This means that the calculation of combustion heat release is underestimated, which in turn can be attributed to an underestimation of heat losses (combustion gross heat release is equal to net heat release (from in-cylinder pressure measurement) plus heat exchange at the wall (estimated with a model)). Two elements could explain an increase of heat losses at cylinder wall when WI is used and water flow rate is increased:

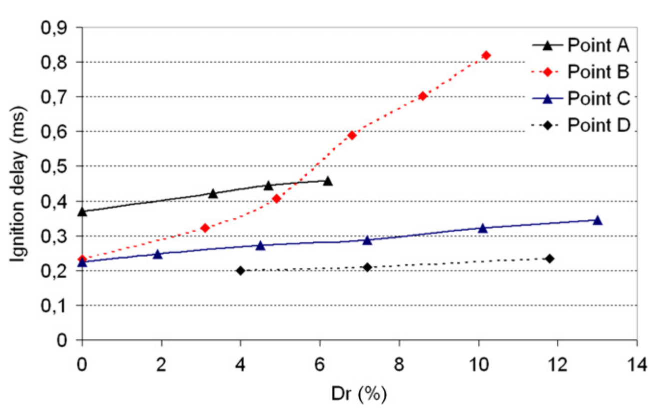

Fig. 6. Influence of WI on ignition delay (main combustion).

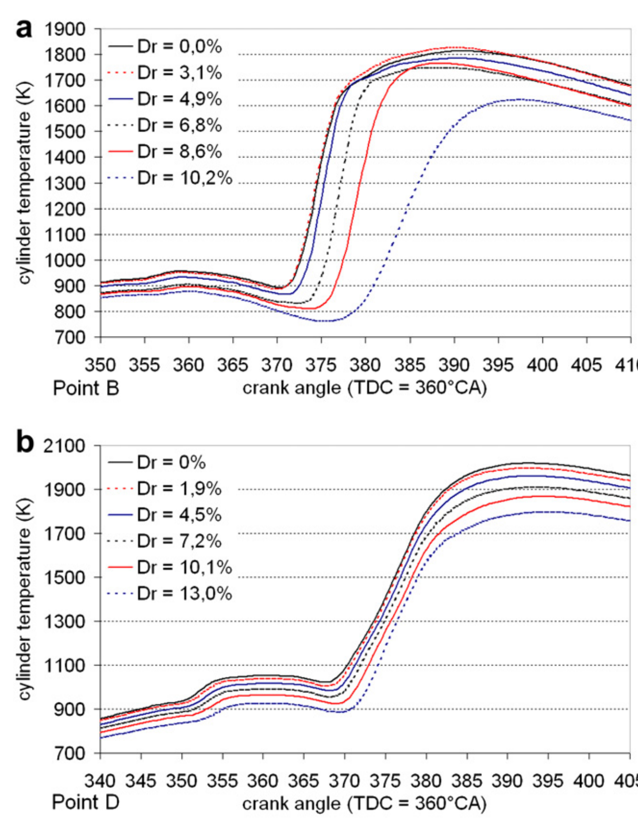

Fig. 7. In-cylinder average temperature for various WI rates.

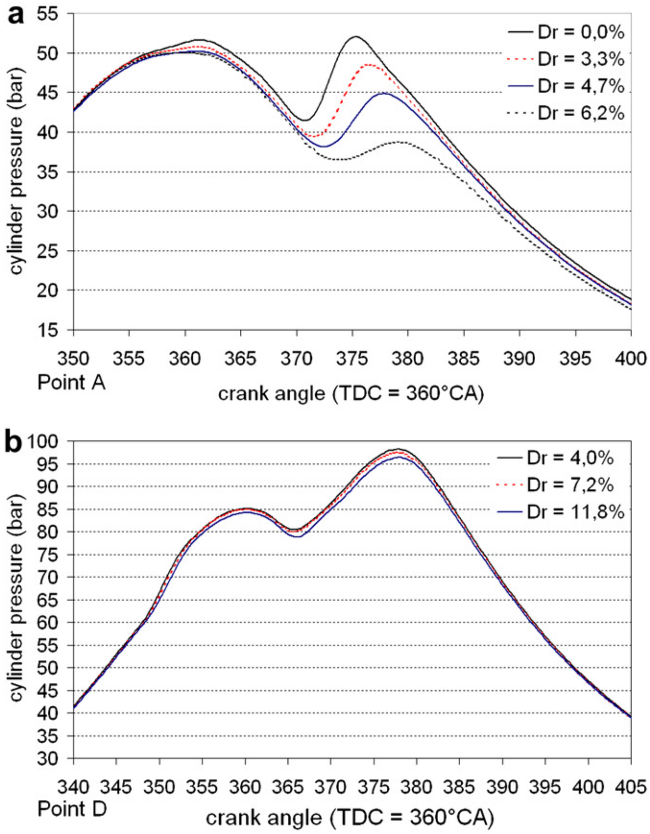

Fig. 8. In-cylinder pressure for various WI rates.

- As depicted above, WI has a cooling effect during water vaporisation which may decrease wall temperature during inlet and at the beginning of compression stroke, which in turn would increase the heat exchange when in-cylinder temperature becomes higher than wall temperature, in particular during combustion and expansion stroke.
- Water vapour has a much higher convective coefficient than air; that could justify a more intense heat exchange between in-cylinder gas content and the wall.

4.1.5. Comparison of WI and EGR effect on combustion

Fig. 11 shows a comparison of WI and EGR effects on ROHR for the same dilution ratio, for operating points A and B. Both EGR and WI cause ID to increase. For EGR, this increase can be attributed to the dilution effect: O_2 density decreases and heat produced by initiation reaction is absorbed by a larger mass, thus the temperature increase is smaller; in-cylinder gas temperature at SOI is not significantly modified compared to reference case, since inlet gas temperature (T_2'') is the same. For WI, as said before, the main effect on ID should be the decrease in gas temperature at SOI, due to the WI cooling effect. The result is that the ignition delay increase is more significant with WI than with EGR (for the same Dr). However, the premixed peak is not amplified (it is even reduced for operating point A) because lower temperatures tend to reduce chemical reaction rates. EGR also leads to lower temperatures during combustion compared to reference case, but to a lesser extent than WI. Actually, it is only due to dilution, whereas WI also decreases initial temperature (before SOC) as can be seen on Fig. 12.

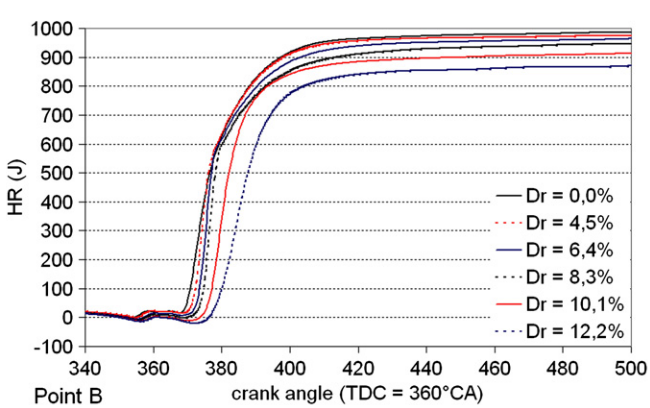

Fig. 9. Calculated combustion cumulative heat release for various WI rates.

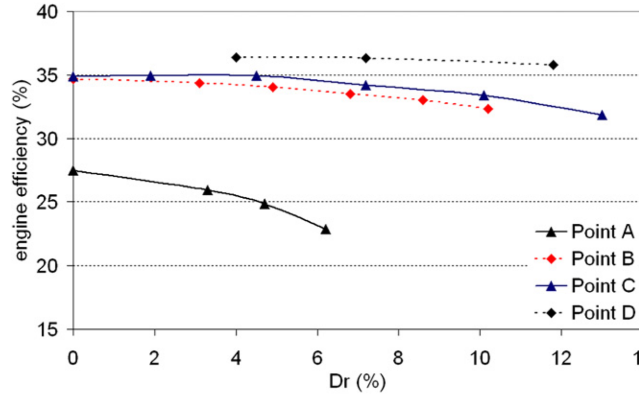

Fig. 10. Influence of WI on engine efficiency.

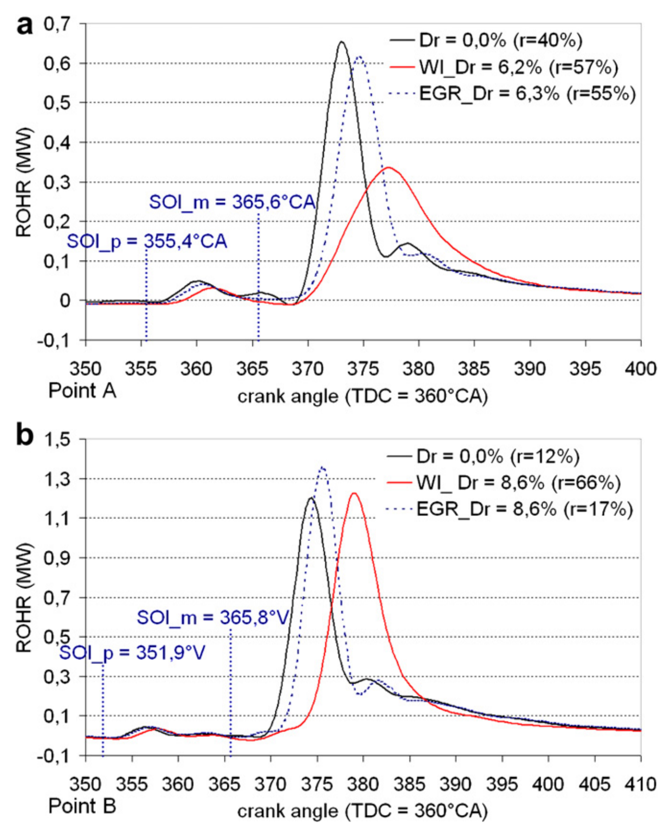

Fig. 11. ROHR comparison of EGR and WI.

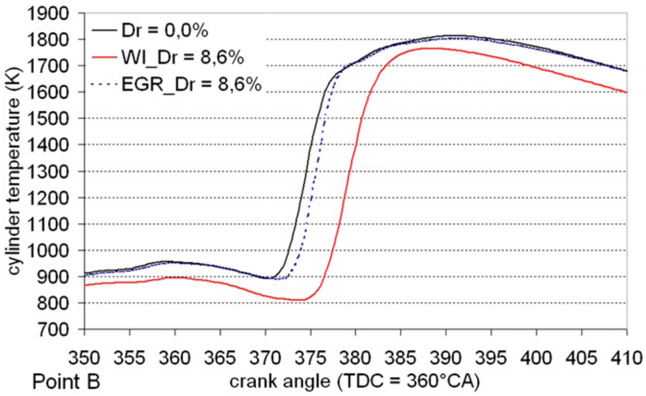

Fig. 12. In-cylinder average temperature comparison of EGR and WI.

4.2. Influence of WI on NOx and soot emissions

4.2.1. Influence of WI on NOx emissions

The influence of WI on NOx emissions is given in Fig. 13 for operating points A to D. As expected, NOx emissions decrease when WI rate increases. This is due to temperature reduction which itself has several causes as discussed in the above section: water cooling effect due to vaporisation, increase of heat capacity due to higher trapped mass, increase of specific heat capacity due to air dilution with vapour, increase of heat losses at the wall, combustion delay due to increase in ID and eventually decrease of chemical reactions rates,… Here it should be noted that some effects of WI (such as a possible increase of premixed combustion peak) may have a tendency to increase NOx emissions but they are largely overcome by other effects.

The maximum NOx reduction depends on the maximum WI rate which is limited by combustion stability. It reaches 77, 76, 89 and 83% for operating points A to D respectively, with a corresponding WI rate of 6, 11, 19 and 21 kg/h respectively. The maximum water/fuel mass ratio varies from 2.3 to 4.0 and the water /intake air mass ration varies from 0.10 to 0.15.

NOx emissions obtained with the LP EGR system (described in Section 2) are plotted on the same figures. EGR also causes NOx emissions to decrease. It must be underlined that the nature of diluent (water or dry exhaust gas), which was studied by Ladommatos et al. [7] is not the only difference between the two cases: WI cooling effect leads to lower temperatures before SOC compared to EGR, water is added to air (supplemental) while EGR replaces some air (substitution), ROHRs are also different,… However these differences seem to compensate for each other so that for a given dilution ratio the NOx reductions with EGR and WI are of the same magnitude, generally a little lower with EGR.

4.2.2. Influence of WI on PM emissions

For PM emissions, the global trend is an increase of emission with WI of 23%, 141% and 502% for operating points B, C and D (see Fig. 14b–d), thus leading to a well known trade-off between NOx and PM emissions. This evolution can be explained by the temperature decrease which limits soot oxidation. A slight decrease of λ is also observed (of 5–7%) which may also limit PM oxidation process. PM production (in the core spray) may also be altered by WI, for instance by the increase of premixed part of combustion which generally produces no PM, or by a change in richness at lift-off length that has a strong effect on the PM production rate [45–47]. But this later point requires optical access to be investigated experimentally.

On the same figures, it can be noticed that the increase in PM emissions with EGR is much more important than with WI, (for a given dilution ratio). Even if the nature of diluent (water or dry EGR) may play a role [7,8], a major explanation for the trend between EGR and WI seems to be the difference in global air/fuel

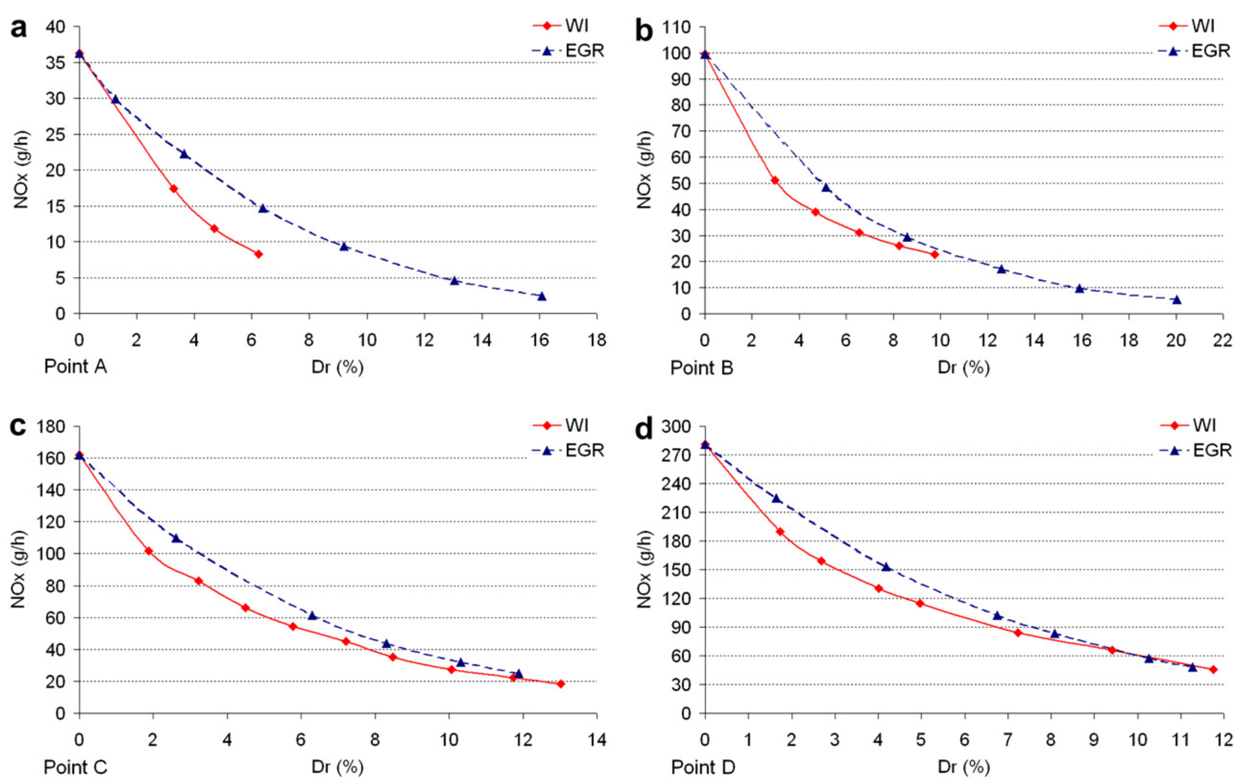

Fig. 13. Influence of WI on NOx emissions – comparison with EGR.

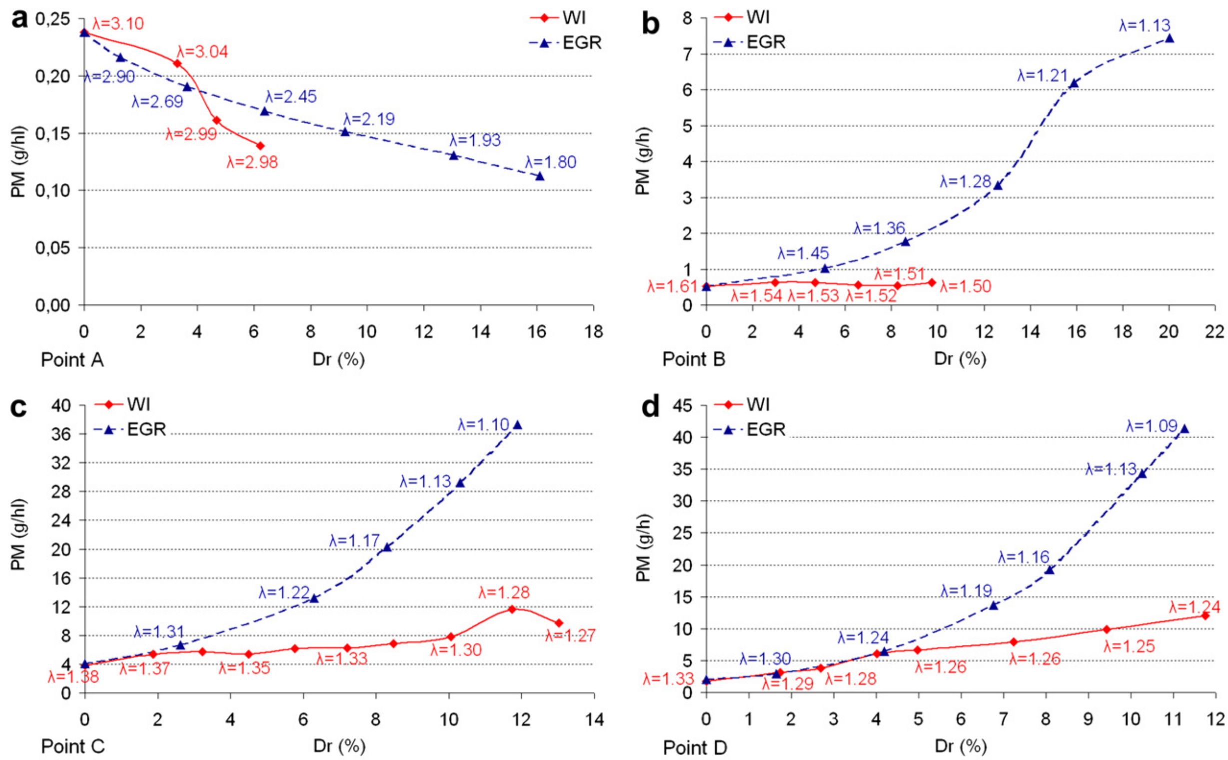

Fig. 14. Influence of WI on PM emissions – comparison with EGR.

equivalence ratio (λ). Indeed, for WI, the cooling effect compensates for the dilution effect so that the trapped mass of air does not vary much with WI rate. Where recirculated gases replace some of the air when EGR rate increases (substitution EGR) since it is used here at constant density at engine inlet (boost pressure and inlet air temperature being kept constant). The decrease of λ is likely to limit PM oxidation, thus explaining the higher final emissions. PM production may also be different between WI and EGR, in particular

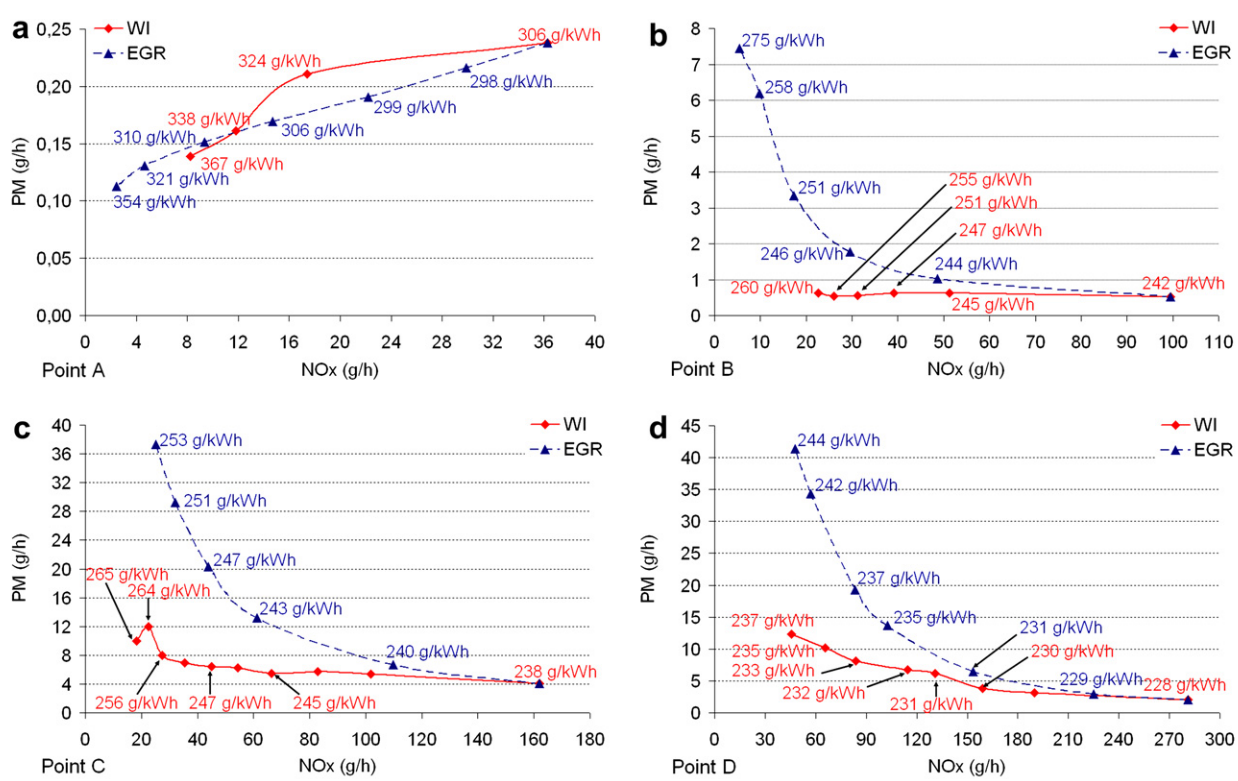

Fig. 15. Influence of WI on NOx–PM trade-off and BSFC – comparison with EGR.

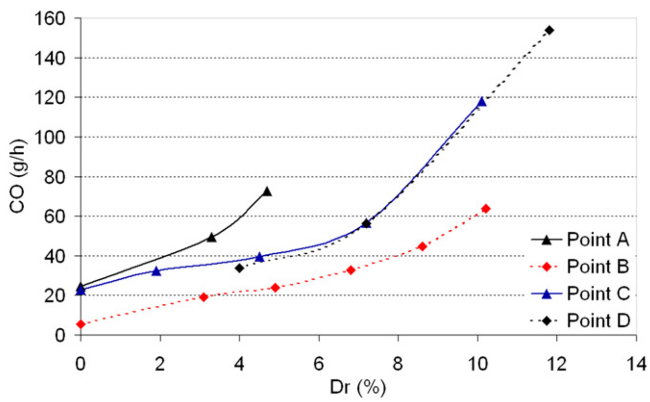

Fig. 16. Influence of WI on CO emissions.

due to differences in temperatures in the core spray or local fuel–air ratio at lift-off length. However, as stated just above, this point requires local experiments or calculations which are beyond the scope of this paper.

A different trend is observed for operating point A (see Fig. 14a): PM emissions decrease when WI rate is increased. This can be explained by the fact that at low load, combustion temperatures are already low without WI, thus WI may cause the engine to enter a low temperature combustion (LTC) mode. In that case, a decrease in PM emissions is classically observed while temperatures are decreased (generally with large amount of EGR [6]). The temperatures then become too low for PM to form, so that engine-out emissions can be reduced, although PM oxidation is also drastically reduced. A major drawback is that at these temperatures CO and UHC emissions increase a lot, as well as BSFC, while combustion efficiency decreases. In that case, PM emissions are similar with both WI and EGR for a given dilution ratio.

4.2.3. *Influence of WI on NOx–PM trade-off*

Finally NOx/PM trade-off curves obtained with WI are presented in Fig. 15 for operating points A to D. BSFC is also presented as well as the NOx–PM trade-off obtained with LP EGR. For operating point A, a simultaneous reduction of NOx and PM can be achieved, both with WI and EGR. This is due to a particular combustion mode (LTC) as described above. For the other three points, a more classical trade-off is obtained which is much better with WI than with EGR: this means that for a given NOx level, PM emissions will be lower with WI than with EGR.

As depicted in the previous part, WI delays combustion because ignition delay increases and sometimes chemical reactions rates are slowed by lower temperatures. This off-phasing effect has positive influence on NOx emissions which are reduced (in-cylinder pressure and in-cylinder temperature are lower during combustion) but a major drawback is the increase of BSFC (see Fig. 15), the cycle thermal efficiency being reduced as described earlier.

4.2.4. *Influence of WI on CO emissions*

The influence of inlet WI on CO emissions upstream the DOC is given in Fig. 16. As can be observed, the increase of dilution ratio results in an increase of CO flow rate upstream the DOC. This may affect the final CO emissions (downstream the DOC) if the DOC is unable to oxidize a higher CO flow rate. Furthermore, the decrease of exhaust gas temperature induced by WI may reduce the conversion efficiency of the DOC. The impact of WI on CO and hydrocarbon emissions as well as their after-treatment in the DOC should be further investigated before any industrial application.

5. Conclusion

The effects of inlet WI have been investigated on a modern, small, automotive HSDI Diesel engine. The effects of water as vapour and as liquid on combustion have been presented and compared with EGR. Here are the main conclusions:

- A large reduction of NOx emissions can be achieved with high WI rates, at low load as well as high load conditions. A water mass of about 60–65% of the fuel is needed to obtain a 50% NOx reduction.
- At low load conditions when air excess is naturally high, EGR has the capability to reduce NOx emissions without increasing PM emissions too much. At these conditions, from a practical point of view, EGR seems to have an advantage compared to WI because it does not require a second liquid (water). At higher loads, WI has the capability to reduce NOx emissions without a large increase of PM emissions because the air flow rate remains approximately unchanged. At these operating points, NOx emissions cannot be largely decreased by EGR because the air flow rate cannot be reduced without a large increase in PM emissions. Thus, the WI technique has a clear advantage in terms of NOx reduction as compared with EGR at high load.
- A new effect of inlet WI as compared with EGR is the reduction of inlet temperature due to vaporisation, which a consequence is a large decrease of in-cylinder temperature at SOI and during combustion, as well as exhaust temperature (which will have consequences on the design and matching of the turbine and the exhaust aftertreatment devices).
- However, WI seems to increase heat losses at cylinder wall, which can affect negatively the engine global efficiency.
- Finally, the influence of WI on combustion (pilot + main injections) has been also studied and show the same effects as for EGR (increase of ignition delay and consequently of the premixed part of combustion, off-phasing effect on combustion). These effects are much more significant for WI than for EGR for a given dilution ratio. To maintain the cycle efficiency, the readjustment of fuel injection while increasing the dilution ratio is thus higher with WI than with EGR.

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