

Experimental Study of Intake Conditions and Injection Strategies Influence on PM Emission and Engine Efficiency for Stoichiometric Diesel Combustion

2011-01-0630

Published
04/12/2011Pierre Solard, Alain Maiboom and Xavier Tauzia
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doi: [10.4271/2011-01-0630](https://doi.org/10.4271/2011-01-0630)

ABSTRACT

Pollutant emissions standards (such like EURO 6 in Europe) are increasingly severe and force a search of new in-cylinder strategies and/or aftertreatment devices / schemes at a reasonable cost. On a conventional Diesel engine an excess of air is usually used to allow very high combustion efficiencies and reasonable levels of soot which can then be after-treated in a diesel particulates filter (DPF). As a consequence, NOx emissions cannot be easily after-treated (lean NOx trap (LNT) and selective catalytic reduction (SCR) are quite expensive even if effective, solutions), as a result they are generally controlled by means of internal measures such as High Pressure (HP) or Low Pressure (LP) exhaust gas recirculation (EGR). In light of ever more stringent NOx emissions regulations, NOx aftertreatment devices seem to be becoming unavoidable.

The goal of the paper is to test stoichiometric combustion on an automotive Diesel engine (with the aim of reducing NOx emission with simple 3-way catalyst) while trying to overcome the traditional drawbacks of this combustion mode (large increase of Particulate Matter (PM), CO and unburned HC emissions and noticeable decrease of engine global efficiency).

An experimental study is carried out on a 2.0L automotive common-rail High Speed Direct Injection (HSDI) Diesel engine, equipped with a variable geometry turbocharger and an intercooler. Two air loop strategies to get stoichiometric conditions are investigated: low temperature throttled fresh air and high temperature EGR. In each case, various injection strategies are tested, especially while varying the timing of the injection splits and the fuel mass distribution. A particular focus is made on in-cylinder pollutant emissions production (PM, NOx), combustion efficiency, and thermal efficiency.

Characteristics and advantages of each air intake strategies are studied and compared. A heat release analysis helps to understand the various combustion modes, in particular to determine the proportion of fuel which burns in premixed conditions. Finally, the most promising paths towards stoichiometric Diesel combustion are underlined. For instance, it seems that when stoichiometry is obtained by intake air with EGR, the combustion processes is more efficient (about 93%). In turn, the throttling of the intake air allows the most advantageous level of PM emissions (soot emission below 2 FSN was achieved). For the best cases, stoichiometric diesel combustion sacrifices specific fuel consumption by around 6%. As a conclusion, some evolutions on engine design are proposed to further improve the concept.

INTRODUCTION

Several technologies (EGR, Common Rail (CR) injection,...) have been progressively introduced in the last few decades for the in-cylinder reduction of NOx emission in automotive Diesel engines. However, emission standards have become so strict that the use LNT or SCR associated with a DPF is now necessary on some applications and will probably tend to generalize in the future. These after treatment devices are efficient to cut-down NOx emission but have some drawbacks, in particular their cost [1, 2, 3, 4, 5]. An alternative solution to remove NOx from exhaust gas could be the use of a much cheaper three-way catalyst (TWC), but this requires that the diesel engine operate with stoichiometric combustion, similar to spark-ignited engines. Traditional Diesel engines on the other hand, use a lean Air/fuel ratio to enable a good quality of combustion (with lambda ranging from high values at low load to minimum values of typically 1.15 at full load/low rotation speed on automotive engines). Only few studies were found

in the literature concerning Diesel engines operating under stoichiometric conditions.

Lee et al. [6-7] conducted experimental studies on a single cylinder engine equipped with a variable helical air intake, a special piston developed using Computational Fluid Dynamics (CFD) optimization to implement Narrow Angle Direct Injection (NADI) concept [8], and a nozzle with a 130 degree included spray angle. The results [7] show that the premixed charge compression ignition (PCCI) mode [5,7,9, 10, 11, 12, 13, 14] allows low soot emission level (0.2 g/kWh) at stoichiometric operation, although it induces approximately a rise of 7% in fuel consumption, resulting from incomplete combustion. CO and HC emissions are thus at higher levels, but can be treated with a TWC. Moreover, the authors notice that the phasing of the heat release rate has a minor influence on the brake thermal efficiency, because a better phasing was accompanied by longer combustion durations.

Experiments indicate that stoichiometry allows brake thermal efficiency of about 35%, relatively independent from the combustion phasing and the combustion regime [6]. Moreover, high boost pressure and high injection pressure seems to improve fuel consumption. This trend was confirmed by others studies [9,15-16]. A statistical analysis shows that the amount of injected fuel and the Start of injection (SOI) are the most significant parameters influencing soot emission. Finally, multiple injection seems to be promising to reduce fuel consumption, when the injection is divided into one or two pilot injections phased early before the main injection.

Kim et al. [17] compared experimentally and numerically (with KIVA CFD code) the engine-out emissions and the combustion process on a single cylinder DI Diesel engine operating at stoichiometric conditions obtained either with a throttled air intake or with a high rate of EGR (and boost pressure). In the case of throttled intake, the portion of premixed combustion is more important than with EGR, because ignition delay and spray penetration are increased. Although combustion and emission characteristics are different, brake thermal efficiencies are very close with the two configurations.

Under the appellation of “rich diesel combustion” [5,11,14,18, 19, 20, 21, 22] some papers study excursions on both rich and lean side of stoichiometry. They show that a higher equivalence ratio leads to an increase of soot, hydrocarbon (HC) and carbon monoxide (CO) emissions, along with a degradation of combustion efficiency.

Some other studies, although not focused on stoichiometric combustion, occasionally reach stoichiometric conditions and can propose some interesting results that could eventually help improve stoichiometric Diesel combustion. In particular, since the in-cylinder soot formation is due to the diffusive part of the combustion and especially to rich local spots [23], various studies show that the promotion of premixed combustion can

reduce, sometimes drastically, PM emission. Homogeneous charge compression ignition (HCCI) engines [24, 25, 26] produce very little soot, but with rather poor combustion efficiency, in particular because of the high spray penetration which can induce the formation of a fuel film on the cylinder walls and the piston. Moreover, this concept still faces difficulties regarding the engine control and is limited to low loads. Some intermediate concepts [21-22,27, 28, 29] consist of a premixed charge that - although not fully homogenous - burns at low temperature, and produces low NOx and PM emission. To avoid hot flame regions, Akihama et al. [18] and Kimura et al. [27] suggested applying high EGR rates to increase ignition delay, thus allowing a better A/F mixing. These EGR levels guarantee low local combustion temperatures. However, an increase of CO and HC emission is also reported.

Finally, previous studies on Diesel stoichiometric combustion relate the influence of some engine parameters but do not examine all possible combination of parameters setting : for instance when intake conditions are varied injection parameters are kept constant. The aim of this paper is to present a systematic experimental study, showing cross-interaction between parameters. For that purpose, three injection parameters (pilot and main injections timing, pilot fuel mass) are varied with four different values being tested for each parameter, in combination with two different intake conditions (throttled air and EGR) thus leading to 128 different experiments. The evolutions of combustion heat release, PM emission, combustion efficiency, and engine efficiency are successively analysed.

EXPERIMENTAL SETUP

TEST ENGINE

The experiments are conducted on a 2.0 L water-cooled HSDI Diesel engine which originally conforms to Euro IV standards. It is equipped with a Variable Geometry Turbine (VGT) turbocharger, an intercooler and a high pressure EGR system (see Figure 1). The original DPF was removed from the engine for the tests presented in this paper. Engine specifications are given in Table 1.

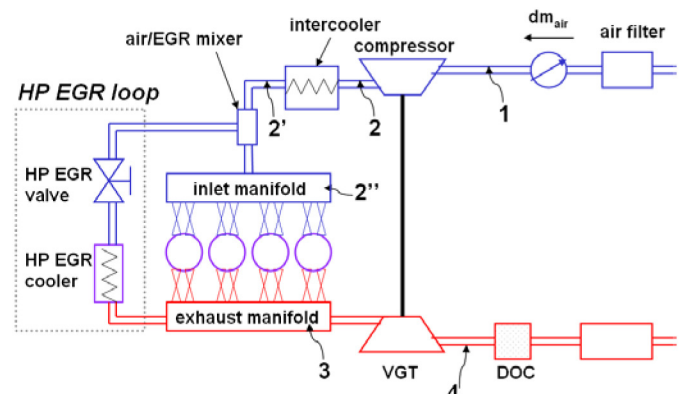


Figure 1. Engine configuration

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 2 nd generation
Maximum injection pressure	1600 bar
Number of injection holes	7
Injector nozzle diameter	0.150 mm

A modified EGR circuit with an air/EGR mixer has been used to ensure that air and recirculated gases are perfectly mixed, so as to keep the cylinder-to-cylinder dispersions low. An independent water circuit through the EGR cooler is used, in order to regulate the temperature of recirculated exhaust gases before reintroduction. The original air/air intercooler was turned into a water/air intercooler, allowing the temperatures T_2' and T_2'' to be controlled separately. The EGR rate is defined as follows:

$$X_{\text{EGR}} (\%) = 100 \cdot \frac{X_{\text{CO}_2\text{-in}}}{X_{\text{CO}_2\text{-ex}}} \quad (1)$$

Where $X_{\text{CO}_2\text{-in}}$ and $X_{\text{CO}_2\text{-ex}}$ are measured CO_2 concentrations in the inlet and exhaust manifolds respectively.

OPERATING POINTS

The study has been conducted for only one operating point (speed and torque). Both injection strategy and air intake conditions are varied. Preliminary experiments have been done to quantify the influence of each engine parameter on PM emissions, combustion efficiency, and thermal efficiency. These first tests have shown that in all cases, the higher the injection pressure, better are the results (decreased PM emissions and improved efficiencies). Actually, the main challenge of Diesel stoichiometric combustion is to promote the mixing process between fuel and the air, which strongly depends on the momentum introduced in the combustion chamber by fuel injection. At a given injection quantity, the total momentum is increased by increasing injection velocity, i.e. injection pressure for a given nozzle geometry. As a matter of fact, it has been chosen to maintain the injection pressure at a high level in the following experiments. Moreover, the injection has been split into two injections: a pilot injection and a main injection. As a consequence, three injections parameters have been varied in this study: the mass of fuel injected in the pilot injection (m_{f_p}), the start of main injection (SOI_m), and the duration between pilot and main injection (SOI_{dif}). Four values have been chose for each injection parameter (given in [Table 2](#)). It must be underlined that the

starts of injection that are given in the table correspond to the starts of injector actuation, the real starts of injection (introduction of fuel into the combustion chamber) being thus delayed by 3.3°CRK .

Table 2. Injection conditions

Engine speed (rpm)	2050
Approx. BMEP (bar)	8.5 bar
Total quantity (pilot + principal) (mg/stroke)	32.5
m_{f_p} (mg/stroke)	2.5 / 4 / 6 / 8
SOI_m ($^\circ\text{CRK}$ ATDC)	0 / -4 / -8 / -12
SOI_{dif} ($^\circ\text{CRK}$)	15 / 18 / 23 / 28
P_{rail} (bar)	1500

Moreover, two intake conditions have been tested (given in [Table 3](#)):

- First condition, called “air” consists in controlling the inlet pressure so as to reach a stoichiometric air-fuel ratio. Since the operating point studied here is at middle load, the inlet pressure is only slightly lower than ambient pressure, equal to 980 mbar. This value is obtained by fully opening the Variable Geometry Turbine (VGT) vanes and slightly throttling the air after the compressor.
- Second condition, called “EGR” consists in using EGR to reach a stoichiometric air-fuel ratio. In parallel, the intake temperature has been increased as compared with first strategy.

Table 3. Intake conditions

	“air”	“EGR”
EGR rate	0 %	8 %
Inlet temperature T_2''	35 $^\circ\text{C}$	75 $^\circ\text{C}$
Inlet pressure P_2''	980 mbar	1200 mbar

EMISSIONS MEASUREMENT

PM emissions at the exhaust are measured with an AVL 415S smoke-meter. NO_x emissions are measured with an ENVIRONNEMENT SA TOPAZE 32M gas analyzer, which uses the chemical luminescence detector (CLD) method. Inlet and exhaust O_2 and CO_2 concentrations are measured respectively with a CAPELEC CAP3200 gas analyzer and a SIEMENS ULTRAMAT 23 gas analyzer, which both use the non-dispersive infrared (NDIR) measurement technique. Each gas analyser is calibrated every hour of experiments with specific gas standards. If the necessary shift is under 0.3 %, then the experiments performed since the previous calibration are validated. The calculation of PM emissions in g/h using the filter smoke number (FSN) is presented in a previous paper [30].

EVALUATION OF THE RATE OF HEAT RELEASE (ROHR)

In order to study the combustion process, a real-time procedure developed at the laboratory calculates the instantaneous gross ROHR on the engine test bench. The calculation is classically deduced from the in-cylinder pressure via a thermodynamic analysis. The in-cylinder pressure is measured with a KISTLER 6055BB piezoelectric pressure transducer and an encoder with a resolution of 0.36 CA degree. The cylinder pressure used is the mean value over 50 consecutives cycles, which was found to be enough for the mean value to be reliable. It must be underlined that the mean gross ROHR gives no indication on eventual cycle to cycle dispersions. The gross ROHR is extracted from the net ROHR by calculating the heat transfer to the combustion chamber walls with Hohenberg's model [31]

EVALUATION OF VARIOUS EFFICIENCIES

The combustion efficiency has been defined here as follows:

$$\eta_c (\%) = \frac{\dot{m}_{CO_2_{ex}}}{\dot{m}_{CO_2_{ex_{th}}}} \quad (2)$$

Where:

$\dot{m}_{CO_2_{ex}}$ is the CO₂ mass flow rate at the engine exhaust, calculated with the exhaust mass flow rate (sum of fuel flow rate and intake mass flow rate) and the CO₂ concentration measured at the exhaust,

$\dot{m}_{CO_2_{ex_{th}}}$ is the theoretical CO₂ mass flow rate at the engine exhaust supposing a complete combustion, deduced from the engine fuel consumption.

In order to compare the two intake conditions ("air" and "EGR"), it is interesting to separate their effects on pumping losses on one hand and on the "closed" part (also called high pressure loop) of the cycle on the other hand. For that purpose, we use the high pressure indicated thermal efficiency is defined by [32]:

$$\eta_{th_{HP}} (\%) = \frac{W_{gross}}{Q_{comb}} = \frac{W_{gross}}{LHV * m_f * \text{combustion efficiency}} \quad (3)$$

Where W_{gross} is the gross indicated work per cycle corresponding to the work given by the HP loop in the P-V diagram (calculated with measured in-cylinder pressure during the HP loop) and m_f is the mass of fuel injected per cycle (pilot + main).

Finally, the brake thermal efficiency is calculated as follows:

$$\eta_{brake} (\%) = \frac{P_{brake}}{LHV * FFR} \quad (4)$$

Where P_{brake} is the brake power and FFR is the fuel flow rate.

ERROR ANALYSIS

Table 4 sums up the measurement technique, calibrated range, accuracy and relative error of various instruments involved in the experiments 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. It is performed here using the differential method of propagating errors based on Taylor' theorem [33]. It gives the maximum error u of a function $f(x_1, x_2, \dots, x_n)$ as follows :

$$u(f(x_1, x_2, \dots, x_n)) = \sqrt{\sum (c_i \cdot u(x_i))^2} \quad (5)$$

As a result, the maximum relative errors for X_{EGR} , PM (g/h), Combustion efficiency, HP indicated thermal efficiency, and brake thermal efficiency are given in Table 5.

Table 4. 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 flow meter)	0-800 mg/str	± 4 mg/str	1 %
Fuel consumption (PIERBURG PLU 401/121)	0.05-23 kg/h	± 37 g/h	± 0.16 %
Nox (ENVIRONNEMENT S.A TOPAZE 32M)	0-10000 ppm	± 10ppm	1 %
Smoke (AVL 415S)	0-10 FSN	± 0.1 FSN	2 %
Inlet CO ₂ (CAPELEC CAP 3200)	0-20 %	± 0.15 %	1.5 %
Exhaust CO ₂ (SIEMENS ULTRAMAT 23)	0-20 %	± 0.1 %	1 %
In-cylinder pressure (Kistler 6055BB)	0-200 bar	± 0.5 bar	1 %

Table 5. Maximum relative error

X_{EGR}	1.4 %
PM(g/h)	2.3 %
Combustion efficiency	1.9 %
HP indicated thermal efficiency	2.4 %
Brake thermal efficiency	1.05 %

RESULTS AND DISCUSSION

HEAT RELEASE RATE

The combustion process is first analyzed using rate of heat release (ROHR) diagrams. They give information about the combustion speed, the ignition delay, the heat released by pilot and main combustions, the combustion durations, and more generally, the combustion mode. The effect of each injection parameter on the combustion process is analysed by plotting on the same graph the ROHR obtained while maintaining the others parameters constant. On all ROHR graphs, a narrow arrow indicates the pilot SOI and a wide arrow, the main SOI. As explained in the experimental setup section, the arrows indicate the real SOI (start of fuel introduction into the combustion chamber), which is delayed for this operating point by 3.3°CRK compared to the electrical SOI.

First, Figure 2 represents the influence of mf_p on combustion process for “air” (a) and “EGR” (b) strategies. SOI_m and SOI_{dif} are maintained constant (respectively -8° ATDC and 18° CRK). Same qualitative results have been obtained with other values of SOI_m and SOI_{dif} .

For both intake air strategies, the increase of mf_p results in a rise of the peak of pilot ROHR. Moreover, when increasing mf_p , the main ROHR peak is reduced. A first explanation could be that the fuel spray is introduced in a gas region that contains more burned gases (from pilot combustion), of which a consequence is the decrease of fresh air entrainment rate during main injection. In addition, main combustion is essentially mixing-controlled (because of the use of a pilot injection prior to main injection), so that the portion of premixed combustion is low (only a few percent of the total mass of fuel). The increase of mf_p has a tendency to decrease main ignition delay (ID) so that the premixed part is reduced. Actually, when increasing mf_p , the main fuel jet is injected in hotter in-cylinder gases. Finally, as the total fuel mass injected is constant, the main fuel quantity is reduced when mf_p is increased, so that the injection ends earlier.

Concerning the two intake air strategies, the use of EGR has a tendency to slightly reduce the ID (pilot ID and main ID). Classically, the use of EGR has opposite effects on ID:

- One one hand, the dilution effect of EGR results in an increased ID [32,34],

• One the other hand, the increase of inlet temperature because of the recirculation of hot exhaust gases has an opposite effect on ID [32,34].

• In addition to the temperature effect, the higher pressure at the SOI decreases the ID [32]. This higher pressure at the end of compression is due to the use of supplemental EGR (same mass of fresh air as without EGR)

In our case, the second effect seems to be predominant since EGR reduces ID. As a consequence, the premixed part of combustion is slightly reduced with EGR so that the combustion is more diffusive in nature with EGR (figure 2 b as compared with figure 2 a).

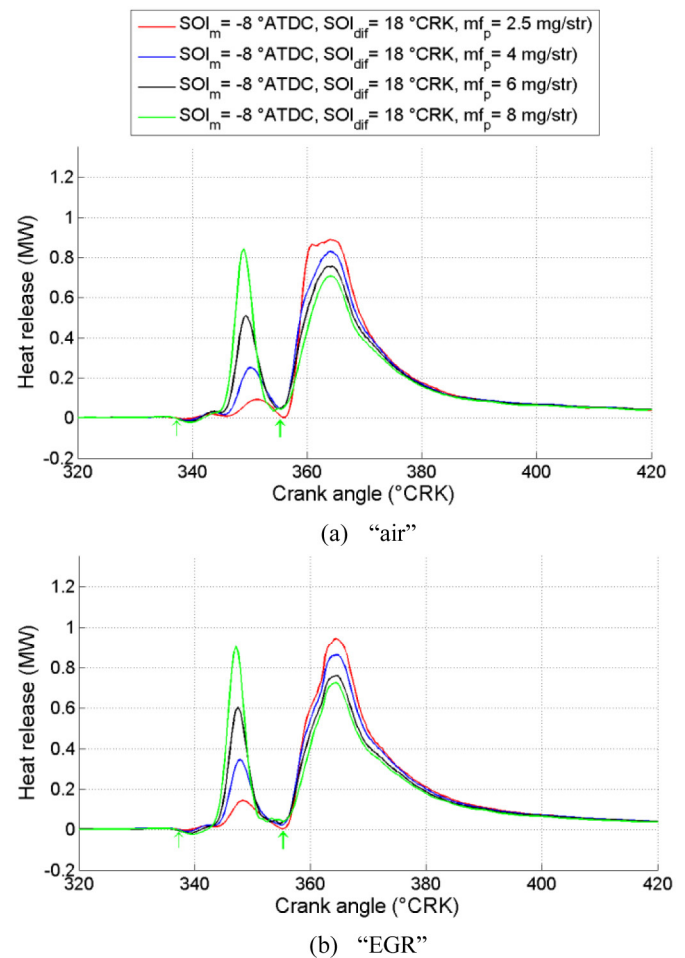


Figure 2. Influence of mf_p on ROHR at fixed SOI_m and SOI_{dif}

Figure 3 depicts the influence of SOI_m on the combustion process for “air” (a) and “EGR” (b) strategies. SOI_{dif} and mf_p are maintained constant (respectively 18° CRK and 4 mg/str). Same qualitative results have been obtained with other values of SOI_{dif} and mf_p .

The main ignition delay seems to be approximately constant, so that the main start of combustion (SOC) is advanced by approximately 4°CRK when SOI is advanced by 4°CRK. Moreover, when injection is advanced, the peak of main ROHR is slightly increased. This is due to the fact that the main combustion is essentially diffusive, the combustion speed being governed by the mixing process between air and fuel. When fuel is injected earlier (SOI_m going from 0°ATDC up to -8°ATDC), the injection event is centered around the top dead center (TDC) when the density of the gases in the combustion chamber reaches its maximum. As a consequence, the fuel spray entrains globally more gases per unit of time and results in an increased ROHR.

As regards the pilot combustion, the ID rises when advancing SOI_p whereas the pilot ROHR peak decreases. When advancing SOI_p, the temperature and pressure into the combustion chamber at SOI_p are retarding; with early injection timings, the pressure and temperature conditions at SOI_p are not sufficient for fast fuel ignition. As a matter of fact, the ID becomes longer, increasing the time for pilot fuel and air to premix before ignition. With the higher value of SOI_p (corresponding to SOI_m = -12° ATDC), the pilot ROHR has two peaks, showing that the pilot fuel is burning under a low temperature combustion (LTC) mode with a characteristic cool flame. It is possible that for this case, the fuel spray penetrates so far in the cylinder that it slightly impacts the cylinder walls. Higher values of SOI_p have been tested in the preliminary experiments, showing that with earlier timings, pilot injections do not even burn, probably because of wall impingement.

When comparing the two intake air strategies, the SOC of both injections are somewhat reduced by the use of EGR, which is certainly due to the higher in-cylinder temperatures at SOI (whose effect is slightly higher than the dilution effect as described earlier). Except for SOI_m = -12°ATDC, the main ROHR is only slightly modified with the “EGR” strategy as compared with the “air” strategy. As regards the influence of the “EGR” strategy on pilot combustion, it is shown that the pilot ID is lower and the ROHR peak is higher probably because of the higher temperature during pilot injection. Actually, the pilot combustion is governed by chemical kinetics, such that an increase of temperatures in the fuel spray reduces the ID and increases the ROHR. Another reason for an increased pilot ROHR peak with the “EGR” strategy is that the density of gases in the combustion chamber is higher than with the “air” strategy thus lowering the pilot fuel penetration and the risk of wall impingement. As a result, for SOI_m = -12° ATDC, the pilot combustion is more complete at main SOI with the “EGR” strategy, such that the main ID is decreased in the case of “EGR” as compared with the “air” strategy. As a consequence, the premixed part of main combustion as well as the ROHR peak are lower with the “EGR” strategy for SOI_m = -12° ATDC.

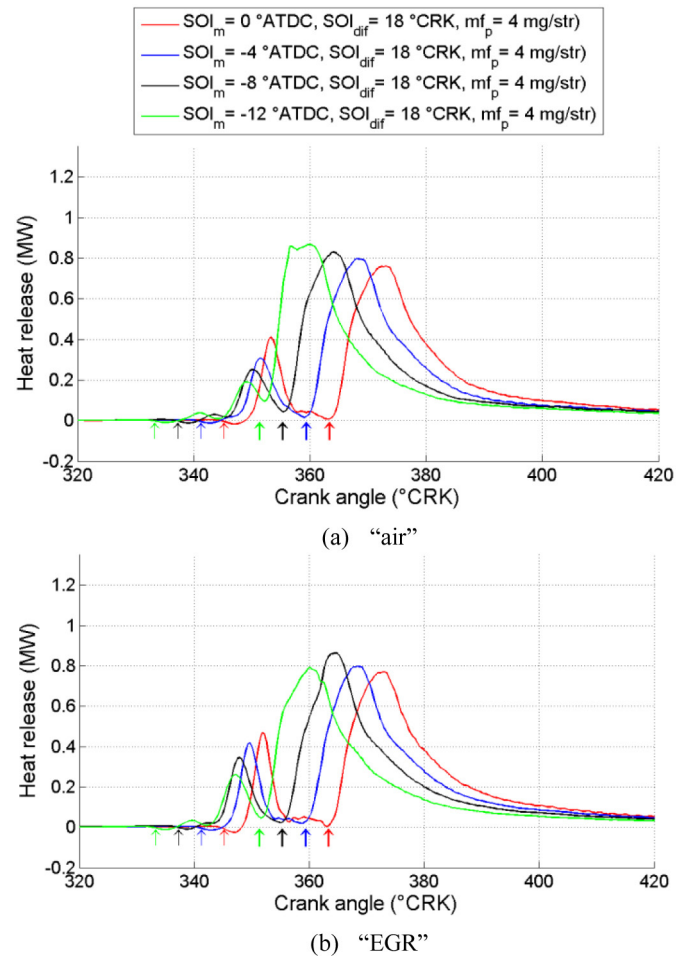


Figure 3. Influence of SOI_m on ROHR at fixed mf_p and SOI_{dif}

The influence of SOI_{dif} for “air” (a) and “EGR” (b) strategies is described in Figure 4. SOI_m and mf_p are maintained constant (respectively -12° ATDC and 6 mg/str). Same qualitative results have been obtained with other values of SOI_m and mf_p . For the pilot injection, the ignition delay increases with increased SOI_{dif} as explained earlier.

As regards the main combustion, there is a clear tendency of increase of the main ROHR peak when increasing SOI_{dif} for both intake air strategies. This can be explained as follows:

- First, when advancing pilot combustion, there is more time for the pilot fuel to mix with air. At main SOI local fuel-air ratios are thus surely lower and the main fuel spray has less probability to interact with fuel rich regions.
- Secondly, when increasing SOI_{dif}, the pilot fuel spray may penetrate further into the combustion chamber, such that the pilot and the main sprays are burning in separated zones of the combustion chamber. For very advanced pilot injection timings, a simple calculation based on the geometrical data of the combustion chamber and a penetration model for pilot injection [35] has shown that the pilot fuel can reach the