LATE DIRECT FUEL INJECTION FOR REDUCED COMBUSTION RATES IN A GASOLINE CONTROLLED AUTO-IGNITION ENGINE

by

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Homogeneous charge compression ignition concepts despite high efficiency and ultra-low nitrous oxides emissions, suffer from week controllability and load range limited by excessive pressure rise rates. In the present work, controlled auto-ignition is achieved via direct injection of gasoline into the exhaust gasses recompressed during negative valve overlap phase. Single cylinder engine experiments are designed to explore the potential of additional late post injection strategies for pressure rise rate and peak pressure suppressing. For two mixture strengths, fuel distribution is varied between 3 gasoline injection events. In-depth combustion analysis is supported by emission measurement results.

Increasing the amount of gasoline, post-injected during the main compression event, was proven to be an effective measure for reducing pressure rise rates, with over 50% reduction potential. The regulation capability however, is limited by typical tread-offs between stratified and homogenous fuelling concepts. Using post injection strategy results in decreased hydrocarbon emissions, but causes rapid increase in carbon monoxide and particulate matter emissions. Nitrous oxide increase rate is dependent on mixture strength with significantly higher sensitivities during lean operation.

Key words*: internal combustion engine, low temperature combustion, controlled auto-ignition, direct fuel injection, mixture stratification*

Introduction

Bad reputation of combustion engines, especially Diesel engines, results from inability to mitigate emissions of nitrogen oxides (NO_x) and particulate matter (PM) at the same time. The trade-off between NO_x and PM emissions calls into question application of Diesel engines as sources of propulsion for light-duty automotive vehicles. Spark ignition (SI) engines exhibit much lower emission rates when supplied with 3-way catalytic converters, however they suffer from lower thermal efficiency [1-3].

The engines utilizing low temperature combustion (LTC) can meet stringent demands concerning emissions in parallel with high fuel efficiency. One promising solution to realize LTC is so-called controlled auto-ignition (CAI) – the combustion system which utilizes high octane fuels in engines with geometrical parameters like that of SI engines [4, 5]. CAI combustion system relies on auto-ignition of the homogeneous air/fuel mixture and volumetric combustion controlled by chemical kinetics. Such combustion process is distinguished by low

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temperature and smokeless exhaust. Additionally CAI combustion is characterized by high thermal efficiency and reduced pumping losses, because engine load is governed by excess-air rather than throttling [6].

To realize CAI combustion in engine with compression ratio typical for SI engine it is necessary to increase compression temperature. This is achieved with the use of internally re-circulated exhaust via so-called negative valve overlap (NVO). Realization of NVO strategy requires reduction of valves' opening durations, thus exhaust valve is closed earlier, to trap some amount of residuals inside the cylinder, and intake valve opening is symmetrically delayed. This engine control strategy is a viable approach to achieve stable CAI combustion and extremely low emissions [7, 8]. However, the high rates of exhaust gas re-circulation (EGR) reduce amount of aspirated air and thus preclude achieving high loads. Numerous research results point out the maximum achievable load expressed by net indicated mean effective pressure (IMEP) at the level of approximately 0.5 MPa [9]. Additionally, high load operation is limited by excessive pressure rise rates (PRR) at high fuelling [10]. These disadvantages pose a challenge for development of a production-ready CAI engine.

Efficient solution to mitigate reduced amount of intake air, and thus enable higher fuelling is application of boost. Supercharging and turbocharging allow increasing the amount of aspirated air. However, it does not solve the issue of high PRR. Canakci [11] applied a moderate boost of up to approximately 0.14 MPa to gasoline port injected HCCI engine and found that increase of the intake pressure at constant fuelling enabled a substantial reduction in NO_X emissions; at the same time, however, an excessive PRR was noted. However, application of high boost pressure of up to 0.3 MPa allowed Kulzer *et al*. [12, 13] to conclude that increasing in-cylinder charge dilution by air enables substantial reduction in PRR. It should be noted, however, that achieving such high boost pressure in CAI engine operated in the NVO mode requires a mechanical boost device, which would reduce overall efficiency of the engine.

Another approach to reduce combustion rates, and thus increase attainable engine load is introduction of some degree of mixture stratification in spite of the fact that CAI engine concept relies on volumetric combustion of a nearly homogeneous mixture. If composition of in cylinder mixture is stratified, local temperatures differ as well. Thermal stratification affects auto-ignition delay as well as combustion rates. The effects of thermal stratification were confirmed by Sjoberg *et al*. [14]. The authors found that even 30 K width thermal stratification can substantially reduce combustion rates, and thus PRR. Recently, Lawler *et al*. [15] found that combustion phasing itself affects thermal stratification and resulting heat release rate.

A feasible method to achieve and control stratification in the combustion chamber is direct injection of some portion of fuel at late stage of the compression stroke. At late injection there is not enough time for mixing of the fuel vapour with remaining in-cylinder fluid [16]. Additionally, this process enhances thermal stratification, because the heat for fuel vaporisation reduces temperature locally. Dec *et al*. [17] applied port fuel injection and direct fuel injection to achieve partial fuel stratification and achieved reduction of PRR. However, this method turned out to be effective only for boosted engines. Mikulski and Wierzbicki [18] and Mikulski and Bekdemir [19] used detailed chemical kinetic model to understand combustion of partially stratified mixture. The authors used two fuels with different reactivity, where low reactivity (high octane number) fuel was injected to intake runner, while high reactivity fuel was injected directly to the cylinder. Turkcan *et al*. [20] applied only direct fuel injection and split injection technique where 80% of fuel was injected early, to create premixed charge, where the remaining fuel was injected late during compression stroke. The delay of the second injection reduced the PRR, however the emissions of CO, unburned hydrocarbons (UHC) and PM increased. The mutual effects of boost and stratification were investigated by Kwon and Lim [21] and Hunicz *et al*. [22]. Both studies found that increasing boost pressure reduced combustion time, whereas stratification increased it. Kodavasal *et al*. [23] performed a study to identify whether compositional or thermal stratification affect combustion to a higher extent. The effect of temperature stratification was found to be more significant than one of composition. Recently Hunicz *et al*. [24] applied variable direct gasoline injection strategies to control stratification under variable boost pressure. It was found that small quantities of fuel injected for stratification do not reduce PRR, while increase NO_x emissions.

The aim of the present study is to further investigate the complex effects of mixture stratification and its strength on combustion rates and emissions. On the basis of the results from the previous study [24], strategic injection timings were selected, to achieve stratification, however without emissions deterioration resulting from piston wetting by the fuel stream. In the case of this study stratification was controlled via the amount of fuel injected rather than injection timing.

Experimental set-up and procedure

Experimental test stand

The experiments were performed in the combustion engine laboratory located in Lublin University of Technology. The object was a single cylinder, 4-stroke, light-duty research engine with 0.5 L displacement volume. The engine was equipped with a swirl-type, single-stream, electromagnetic injector by Bosch. Boosting pressure was applied via an externally driven compressor. In order to realize NVO operation a variable valve actuation (VVA) system was used. The system, shown in fig. 1 and discussed in detail in the earlier works [24-26], allowed for independent adjustment of intake and exhaust valve opening phases and lifts. The regulation range was around 110 °CA (crank angle) and 9.4 mm for valve timing and lift, respectively. The detailed data of the test engine, along with the VVA system settings used in the present

Table 1. Research engine specification

Displaced volume	498.5 cm^3
Bore × Stroke	84 × 90 mm
Compression ratio	11.7
No. of valves	2
Intake valve opening (IVO) timing	87 °CA
Intake valve closing (IVC) timing	217 °CA
Intake valve lift	3.6 mm
Exhaust valve opening (EVO) timing	515 °CA
Exhaust valve closing (EVC) timing	–86 °CA
Exhaust valve lift	2.9 mm

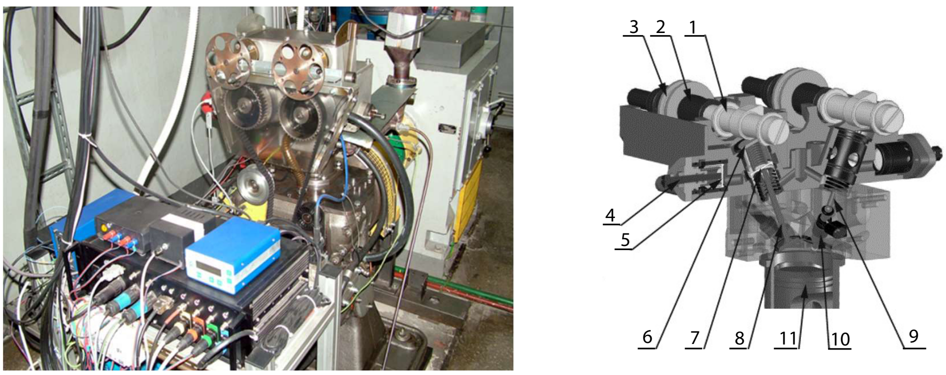

Figure 1. Experimental test stand (a), and design of the valvetrain control mechanism (b)
1 – camshaft, 2 – shaft drive, 3 – slider for valve timing adjustment, 4 – valve lift adjustment screw, 5 – piston of hydraulic accumulator, 6 – piston co-operating with the cam, 7 – piston co-operating with the valve, 8 – exhaust valve, 9 – intake valve, 10 – fuel injector, 11 – piston

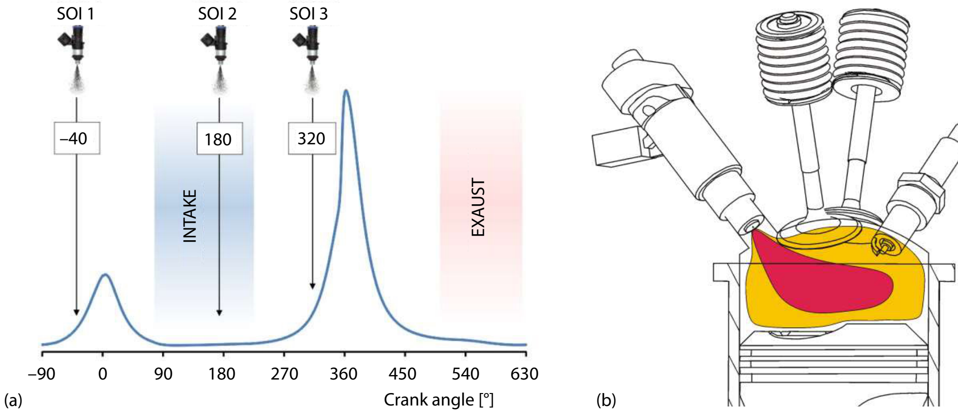

Figure 2. Exemplary in-cylinder pressure with presentation of fuel injection strategy (a), and concept of mixture stratification (b) (*for color image see journal web site*)

research are shown in tab. 1. For clarity, the values of valve actuation crank angles (tab. 1) are indicated in fig. 2, where an exemplary in-cylinder pressure trace is presented.

The engine was installed on test stand with direct current, dynamic dynamometer and equipped with all standard control and measurement devices. In-cylinder pressure was measured with the use of a miniature pressure transducer installed directly in the engine head. The pressure was acquired with constant angular resolution 0.1 °CA of crank angle. Intake air mass flowrate was measured by a thermal mass flow meter. Fuel consumption was provided by a fuel balance. Exhaust composition was measured using Fourier transform infrared (FT-IR) analytical system. Additionally, PM content was measured using a MAHA MPM4 meter. The excess air ratio (λ) was measured with the use of LSU 4.2 Bosch lambda probe and ETAS lambda meter. Numerous pressure and temperature transducers were used to control the thermodynamic conditions of intake air, exhaust, cooling liquid, lubricating oil, *etc*. The list of measured signals, used transducers and instrument accuracy is provided in tab. 2. The engine control system was computer-based and used dedicated software. It was connected to a timing module which controlled injection timings and durations. All the experiments were performed with the ignition system switched off.

Table 2. Measurement equipment and accuracy

Measured quantity	Transducer	Measured range	Accuracy
In-cylinder pressure	AVL GH12D	0-25 MPa	0.5-2.0%[(1)]
Fuel consumption	AVL fuel mass flow meter 735S	0-125 kg/h	0.12%
Excess air (λ)	Bosch LSU 4.2 / ETAS LA4	0.6-33	0.03%
Air mass flow rate	Bosch HFM5	8-370 kg/h	3%
Intake/exhaust press.	Freescale MPX4250AP	20-250 kPa	1.5%
Temperatures (ambient, intake air, cooling, oil, fuel)	Pt100 Czaki TP-361	–40-400 °C	0.2%
Exhaust temperature	Thermocouple K Czaki TP-204	0-1200 °C	0.8%
Exhaust composition (gaseous compounds)	AVL Sesam FT-IR	CO: 1-10000 ppm HC: 1-1000 ppm[(2)] NO_x: 1-4000 ppm	0.36% 0.1-0.49%[(3)] 0.31%
PM content	MAHA MPM4	0-700 mg/m^3	0.1 mg/m^3

[(1)] Depending on temperature; [(2)] Given measurement span relates to concentration of a single identified hydrocarbon;
[(3)] Depending on type of hydrocarbon species

Experimental procedure

The experiments were performed at constant rotational speed of the engine set to 1500 rpm at mid-load operation with wide open throttle to enable lean operation at high internal EGR. The engine was fuelled with European Euro Super commercial gasoline with a research octane number of 95. The valvetrain was set to achieve NVO (valve timings and lifts are shown in tab. 1), and thus internal exhaust gas re-circulation.

The design of experiment which allowed us to investigate the effects of fuel stratification on combustion and emissions was as follows. Fuel injected directly into the cylinder was split into three parts, as shown in fig. 2(a).

The first fuel dose was introduced into the cylinder during the exhaust compression. The first injection duration was fixed for all experiments and delivered approximately m_{F1} = 2.6 mg of fuel. The mentioned uncertainty results from the fact that the injector was operated on the shortest applicable actuation timings. Under such conditions the fuel delivery rate can be affected by variations in in-cylinder pressure and injector tip temperature. The role of the first injection was enabling of heat release during the NVO period and production of auto-ignition promoting species via exhaust-fuel reactions [25-27]. It was found that this approach allows this engine for stable operation at variable other control parameters [28].

The second fuel injection was applied at 180 °CA to create premixed charge. The third fuel injection, applied at fixed SOI_3 = 320 °CA was utilized to control mixture stratification via amount of fuel injected. This injection timing was selected, because previous studies demonstrated that it was late enough for mixture stratification, however with avoidance of piston impingement [24]. During the experiments the amount of fuel injected with the third injection was gradually increased to enhance mixture stratification. At the same time the amount of fuel delivered with the second injection was reduced to maintain constant overall mass of fuel injected during the engine cycle. The idea of mixture stratification on the background of the employed combustion system are shown in fig. 2(b). The area in yellow stands for a premixed mixture created with fuel injected in the 1st and 2nd injection, whereas the red area marks a higher fuel concentration prior to auto-ignition, created by the 3rd injection. The experiments were repeated for two fixed levels of λ value, which were controlled by correction of intake pressure delivered by boost device. On average, the boost pressure for λ = 1.3 was approximately 0.129 MPa abs. whereas for λ = 1.6 intake pressure was 0.159 MPa abs.

At the selected valvetrain parameters (shown in tab. 1) and injection durations (shown in tab. 3) an average IMEP was 0.39 MPa for λ =1.3 and approximately 0.41 MPa for λ = 1.6. The difference in IMEP values at nearly constant fuelling resulted from slightly higher thermal efficiency at leaner mixture and influence of external boost device. The rate of internal EGR was approximately 0.36 for λ = 1.3 and 0.37 MPa for λ = 1.6.

Table 3. Engine control parameters

λ (–)	1.3	1.6
Intake pressure, [MPa]	0.129	0.159
SOI_1, [°CA]	–40	–40
m_{F1}, [mg]	2.6	2.6
Inj. time 1, [ms]	0.27	0.27
SOI_2, [°CA]	180	180
m_{F2}, [mg]	2.44-12.55	2.43-12.35
Inj. time 2, [ms]	0.25-1.3	0.25-1.28
SOI_3, [°CA]	320	320
m_{F3}, [mg]	0-9.76	0-9.7
Inj. time 3, [ms]	0-1	0-1
Total m_F, [mg]	15.1	14.9

Data analysis

The data analysis was performed with the use of AVL BOOST software [29], where the measured pressure traces (in-cylinder, intake and exhaust manifold) were utilized as boundary conditions for models of gas exchange and combustion. The aim of modeling was to provide detailed information on heat and mass transfer (through intake and exhaust valves) in the analysed cylinder. This enabled accurate estimation of IVC conditions relevant for further in-cylinder pressure analysis.

Net heat release rate (HRR) was computed using the first law of thermodynamics in the following form eq. (1):

$$\mathrm{HRR} = \frac{\gamma}{\gamma - 1} p\mathrm{d}V + \frac{1}{\gamma - 1} V\mathrm{d}p \tag{1}$$

where p was in-cylinder pressure, V was volume above the piston, and the ratio of specific heats γ was computed according to instantaneous temperature and mixture composition in the cylinder. Mass fraction burned (MFB) was computed as standardized cumulative HRR. To avoid errors originating from measurement noise, pressure rise rate was calculated as a slope of linear function fitted to a single-degree-range 11 experimental points of measured pressure.

The raw emission results (concentrations) were recalculated to the unit of mass flow rate, using the exhaust mass flow and molar masses of the individual species. These were further transformed to (net) indicated specific values by dividing by indicated power generated from full engine cycle. Consequently, engine efficiency was assessed by means of indicated specific fuel consumption (ISFC). Bearing in mind constant fuelling used in all presented experiments, this value is to be interpreted as the measure of how the given amount of fuel is transformed to useful piston work.

Experimental results and discussion

In-cylinder pressure curves for investigated parameter sweeps, shown in fig. 3, reveal combustion sensitivity to mixture stratification. Without any detailed analysis the pressure curves show that the most favourable conditions for early combustion onset appear for the highest mass of fuel delivered with the third injection (m_{F3}). At the same time peak pressure values are the smallest and increase with reduction of m_{F3}. Moreover, pressure curves indicate differences in the combustion response to injection strategy at different overall λ values. In spite of difficulty in estimation of stratification degree it is a reasonable assumption that at leaner

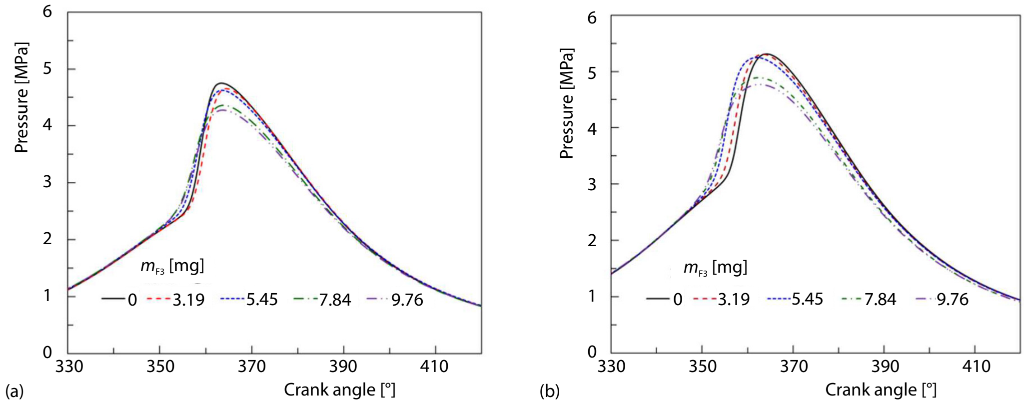

Figure 3. In-cylinder pressure at variable m_{F3}; (a) λ = 1.3, (b) λ = 1.6
(*for color image see journal web site*)

overall mixture, λ value of premixed charge, marked with yellow color in fig. 2(b), prior to third injection is higher.

More detailed data on combustion evolution is provided by the HRR curves, shown in fig. 4. It can be noted that variable third injection quantities affect not only combustion onset, but also combustion speed. In all cases, the highest peak HRR values are observed for only two first injections. When the degree of fuel stratification increases, the thermal stratification increases as well, creating regions with higher temperature where combustion starts. However, increase of stratification reduces combustion rate. The plausible explanation of this trend can be as follows. Increase of stratification creates regions of oxygen deficiency, thus after oxygen consumption from these regions, combustion is controlled by diffusion processes. Additionally, the premixed charge is leaner for higher values of m_{F3}, which reduce reaction rates. It can be noted that there is no meaningful differences between peak HRR for applied overall mixture strengths. However, it results from differences in combustion variability between measurement series.

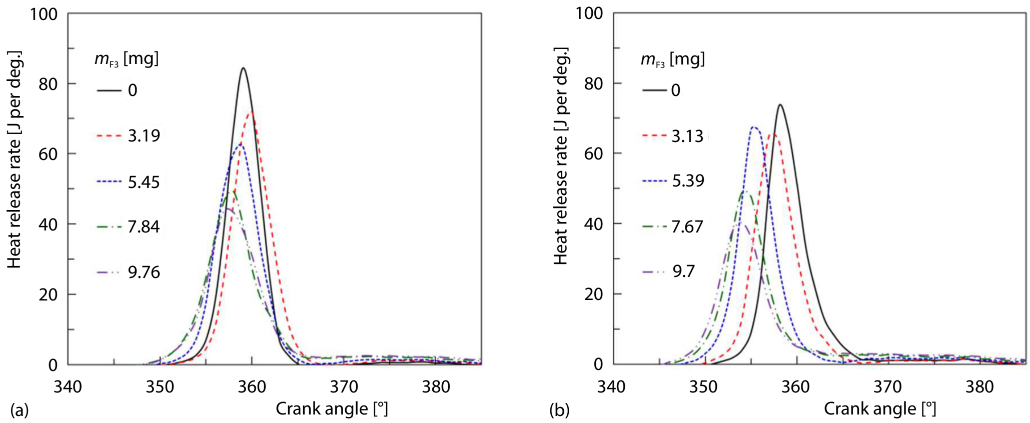

Figure 4. Calculated heat release rate at variable m_{F3}; a) λ = 1.3, b) λ = 1.6

To avoid the effect of cycle averaging, the values of peak PRR were calculated for each operating point from 100 analysed cycles separately and then averaged. These quantities, shown in fig. 5(a), reveal that at leaner mixture PRR is reduced by approximately 0.1 MPa/°CA for all injection strategies. This observation justifies the previous considerations

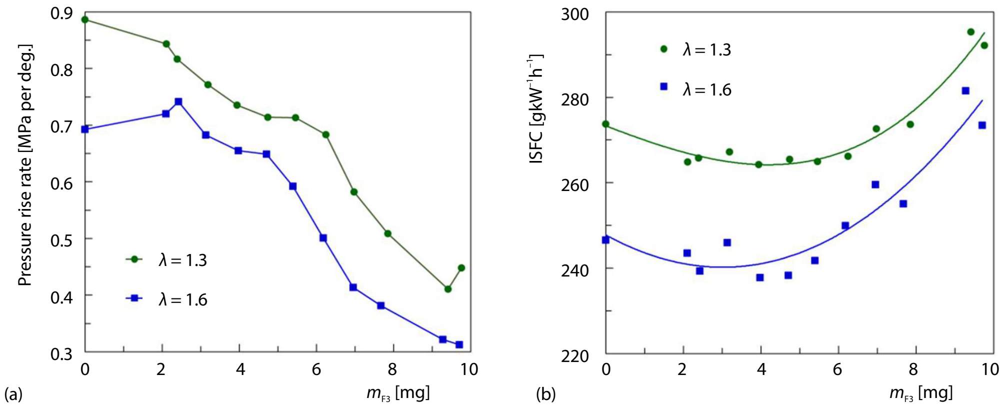

Figure 5. (a) peak pressure rise rates, (b) indicated specific fuel consumption for all investigated conditions

of the effect of λ and stratification on combustion rates. Nevertheless, at small or zero m_{F3} the limit of 0.5 MPa/°CA is exceeded. The minimum m_{F3} amounts which ensure acceptable PRR are approximately 6 mg for $\lambda = 1.6$ and 8 mg for $\lambda = 1.3$.

Although variation in mixture stratification using direct fuel injection shows potential of combustion phasing control, the regulation boundaries can be limited by acceptable levels of exhaust emissions and engine efficiency. The changes in the latter can be deduced from indicated specific fuel consumption, shown in fig. 5(b). The ISFC reaches minimum values for m_{F3} values around 4-6 mg. This is consistent for both tested lambda conditions. With constant fuelling (15.1 mg lambda of 1.3 and 14.9 mg for lambda of 1.6), this means that moderate stratification supports better engine efficiency, while over stratifying significantly worsens the engine performance. Generally more favourable ISFC values for the leaner mixture are attributed to higher γ. Note that in real-world engine operation the efficiency advantage of the lean case might be lost due to required, increased supercharger effort, which is not taken into account for indicated power calculation.

Figure 6 shows indicated specific emissions of toxic gaseous compounds and PM. It can be noted that the trends for CO and UHC are nearly the same, despite the mixture strength. Moreover, decrease of UHC emissions with increase of m_{F3} indicates that there was no piston wetting during the experiment. The observed trade-off between CO and UHC emissions plausibly results from two factors: in-cylinder temperature and wall-side effect. Usually too low combustion temperature increases CO emissions by breaking the CO oxidation process at later

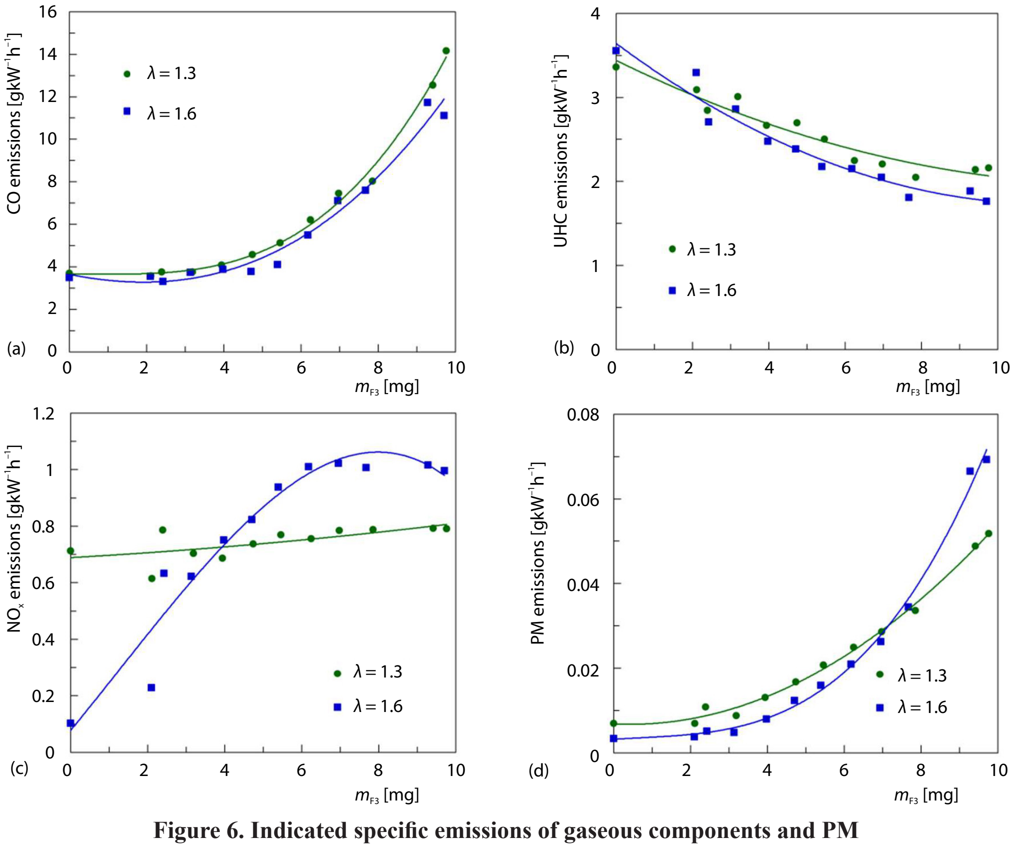

Figure 6. Indicated specific emissions of gaseous components and PM

stage of combustion. The peak in-cylinder temperature obviously results from fuel dilution and combustion timing. Under the conditions of current experiments the peak temperature without third injection, *i. e.* with the most premixed charge, was 1540 K for leaner mixture, and 1660 for richer mixture.

At maximum quantity of m_{F3} the average temperature decreased to approximately 1520 K and 1570 K, respectively. Additionally, thermal stratification reduced temperature in some regions of combustion chamber, which affected increase of CO content. The effect is further supported by lower oxygen availability in the fuel rich regions associated to increased stratification. At the same time emissions of UHC decreased, plausibly because of leaner premixed mixture, which had contact with combustion chamber walls.

Note that for both mixture strengths the total chemical energy associated to the products of incomplete combustion (CO and UHC) was fairly constant, across the whole m_{F3} sweep. This suggests that the ISFC changes observed in fig. 5(b) result from combustion phasing rather than from combustion efficiency.

The trends shown in fig. 6(c), reveal complex mechanisms which affect factors enhancing NO_X production, which are predominantly local temperatures and air excess ratios. For $\lambda = 1.3$ there is only slight increase in NO_x emissions with increase of m_{F3}. In contrast, for $\lambda = 1.6$ NO_x emissions increases much more rapidly, however it stabilizes at larger amounts of fuel injected for stratification. It should be noted how large is the effect of excess air on NO_X emissions at nearly homogeneous mixture, *i. e.* with only two injections. Increase of λ from 1.3 to 1.6 results in drop in the emissions from 0.7-0.1 g/kWh.

Figure 6(d) shows that trends in emissions of PM are likewise CO. The more fuel injected for stratification the higher PM emissions for both mixture strengths. For small quantity of fuel injected late, at $\lambda = 1.3$ the PM emissions are considerably higher, however for the maximum amount of fuel injected for stratification higher emissions appear for leaner mixture.

Conclusions

Increasing the amount of fuel injected during the main compression event was proven to be an effective measure for reducing pressure rise rates in the CAI combustion mode. This provides load extension potential which was identified as one of the main development challenges for the concept.

- For both mixture strengths, the maximum PRR reduction was around 50%, from the case when no post injection was used to the case with 65% of fuel injected in the late post event (maximum tested). The regulation capability however, can be limited with certain trade-offs identified in this study.
- Combustion phasing showed minor sensitivity on variation in mixture stratification, using direct fuel injection, for mixtures with λ below 1.3. The sensitivity increased with leaning the mixture. For $\lambda = 1.6$, ignition delay significantly reduced with higher post injection ratios. The effect of prolonged combustion duration was observed for both tested mixture strengths.
- In terms of emissions, the effect of increasing the post injection duration from 0 to maximum case resulted in: around 40% reduced UHC emissions, large increase of CO and PM emissions (around 250% and 60%, respectively). The change was similar for those compounds for both mixture strengths, which was not the case for NO_x. The NO_X emissions, recorded for homogenous operation, were significantly higher for $\lambda = 1.3$, yet exhibited only slight increase with increased fuel stratification. For leaner mixture, despite the initial emission was ultra-low, it increased drastically with increased stratification.

- The potential for PRR reduction with increased mixture stratification is reduced by deteriorating efficiency attributed to early combustion. With the trend in decreasing PRR, across whole range of tested stratification cases, appearing independent of efficiency, this poses the opportunity for relatively easy combustion process optimization.

Acknowledgment

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