MOBILITY & VEHICLE MECHANICS

https://doi.org/10.24874/mvm.2025.51.03.02

UDC: 621.434:621.43.032.26:621.43.018.7:621.43.019.2

GASOLINE DIRECT INJECTION STRATEGY ANALYSIS FOR IMPROVED COMBUSTION STABILITY DURING SIDI ENGINE WARM-UP OPERATION

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Received in August 2024 *Accepted in October 2024*

RESEARCH ARTICLE

ABSTRACT: Energy efficiency drives the ICE development for more than a century, while ever-increasing ecology awareness in the last decades and requests for the reduction of toxic and CO2 emissions, lead to complex and demanding driving cycles and put severe challenges in the optimization of critical operation regimes to secure ICE application. The Spark Ignition Direct Injection Engine (SIDIE) featuring high boost and high compression ratio, Variable Valve Timing (VVT), complex injection strategies and combustion systems (HCCI/SACI), seems to be an effective response to high demands. Cold start/warm-up, regardless of combustion system applied, is a critical part of emission certification procedure in all standard driving cycles. Catalyst efficiency depends on temperature and engine capability to produce the exhaust gas heat flux necessary for its warm-up. A specific test procedure was designed and performed on Spark Ignition Single Cylinder Research Engine (SI-SCRE) to investigate and evaluate the influence of DI strategy set-up on the capability to provide fast catalyst warm-up during low-temperature/low-load operation. Double injection strategies in terms off Start of Injection (SOI), Split Factor (SF) and Rail Pressure (RP) were tested and analysed against combustion dynamics and combustion stability parameters obtained from in-cylinder pressure indication and complex thermodynamic post-processing.

KEY WORDS: *Spark ignition direct injection, warm-up operation, split injection, combustion dynamics, combustion stability*

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ANALIZA STRATEGIJE DIREKTNOG UBRIŽANJA BENZINA ZA POBOLJŠANU STABILNOST SAGOREVANJA TOKOM ZAGREVANJA SIDI MOTORA

REZIME: Energetska efikasnost pokreće razvoj motora sa unutrašnjim sagorevanjem više od jednog veka, dok sve veća ekološka svest u poslednjim decenijama i zahtevi za smanjenjem emisije toksičnih materija i CO_2 dovode do složenih i zahtevnih ciklusa vožnje i postavljaju ozbiljne izazove u optimizaciji kritičnih režima rada kako bi se osigurala primena motora sa unutrašnjim sagorevanjem. Motor sa direktnim ubrizgavanjem (SIDIE), koji se odlikuje visokim pritiskom i visokim stepenom kompresije, promenljivim vremenom rada ventila (VVT), složenim strategijama ubrizgavanja i sistemima sagorevanja (HCCI/SACI), čini se da je efikasan odgovor na visoke zahteve. Hladni start/zagrevanje, bez obzira na primenjeni sistem sagorevanja, ključni je deo postupka sertifikacije emisija u svim standardnim ciklusima vožnje. Efikasnost katalizatora zavisi od temperature i sposobnosti motora da proizvede toplotni fluks izduvnih gasova neophodan za njegovo zagrevanje. Specifična procedura ispitivanja je dizajnirana i izvršena na jednom cilindričnom istraživačkom motoru sa unutrašnjim sagorevanjem (SI-SCRE) kako bi se istražio i procenio uticaj podešavanja DI strategije na sposobnost brzog zagrevanja katalizatora tokom rada na niskoj temperaturi/niskom opterećenju. Strategije dvostrukog ubrizgavanja u smislu početka ubrizgavanja (SOI), faktora podele (SF) i pritiska u šini (RP) testirane su i analizirane u odnosu na dinamiku sagorevanja i parametre stabilnosti sagorevanja dobijene iz indikacije pritiska u cilindru i složene termodinamičke postprocesne obrade.

KLJUČNE REČI: *Direktno ubrizgavanje sa paljenjem svećicom, rad zagrevanja, podeljeno ubrizgavanje, dinamika sagorevanja, stabilnost sagorevanja*

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INTRODUCTION

The low fuel consumption, and corresponding CO_2 emission, are two of the main objectives of the optimization during testing on reference driving cycles. European emissions regulations are not only about limiting CO_2 emissions, but also about limiting emissions of toxic exhaust gas components, such as carbon monoxide (CO), unburned hydrocarbons (HC) and nitrogen oxides (NOx), which are the consequence of combustion reactions dynamics and complexity. These harmful components of the exhaust gases can be eliminated to a considerable extent, but unfortunately not completely by subsequent exhaust gas treatment. One way to achieve this, in engines operating with a stoichiometric mixture, is to use a three-component catalyst.

Three-component catalytic converters (TWC or 3WC) have proven to successfully cope with the ever more stringent emission limits, being able to convert more than 95% of the relevant raw engine-out emissions under nominal operating conditions. However, as long as the temperature of the TWC is below the “light-off temperature” of around 250 °C, its conversion rate is significantly reduced. Consequently, a large portion of the raw engine-out emissions emitted into the atmosphere remains untreated [1].

Therefore, one of the main tasks, in addition to reducing fuel consumption, is to reach the catalytic converter optimal efficiency as soon as possible after the cold start. Strategies for the catalyst heating phase can be categorized as follows:

Strategies with additional hardware components enabling fast catalyst heating-up process, such as secondary air pumps [5–7] or electric heaters [8,9].

Strategies without any additional hardware components, based on dedicated combustion control techniques that in turn generate hot exhaust gas.

In this paper, special attention is paid to the second strategy, more specifically to the double injection strategy as a part of the entire engine control strategy. After an introductory analysis of the combustion problem in gasoline engines and a comparative review of the reference driving cycles, a detailed analysis of the engine injection control parameters that affect the enthalpy of exhaust gases was performed.

1 DRIVING CYCLES

Vehicle exhaust emissions tests incorporate specific test procedures based on driving cycles which in turn can be performed on test benches or in real road driving conditions, such as:

- NEDC (New European Driving Cycle), designed in the 1980s and became outdated today due to continuous evolutions in automotive technology, changes in driving conditions and driving styles.
- WLTC (Worldwide harmonized Light vehicle Test Cycle)
- RDE (Real Driving Emission test) step 1 (with a NOx conformity factor of 2.1) is applied since September 2017 to new car types, and It from September 2019 for all types.

- RDE step 2 (with a NOx conformity factor of 1.0 plus an error margin of 0.5) is applied from January 2020 for new types and from January 2021 for all types.

Quite opposite to the technology used for the WLTC, which is based on real-driving data, the old NEDC test was based on a theoretical driving profile. Driving conditions have also changed since the 1980s with increased traffic congestion, resulting in more inefficient driving. Additionally, driving styles have also changed. In some countries the new car sales are driven by the companies whose drivers use a company fuel card usually not considering the fuel economy as the highest priority. Therefore, NEDC isn't as representative of today's driving profiles as it was to be in the past.

Figure 1 depicts direct comparison of the vehicle speed profile and simulated engine coolant temperature during the NEDC and WLTC [2]. In Table 1 the main descriptive parameters of these two cycles are given. Operating cold engine increases harmful exhaust emissions and CO_2, due to higher mechanical friction (higher lubricant viscosity, parts tolerances), and mostly due to inefficient combustion process during warm-up (fuel atomization/evaporation, mixture strength, heal losses). The absolute contribution of a cold start to emissions is almost the same across all standardized driving cycles, but its impact on the total emissions during the reference cycles decreases with increased distance travelled by the vehicle during testing. Considering the simple correction for the test distance, which is shown by the expression (1), the effect of the warm-up sequence in the WLTC can be roughly approximated to roughly 50% lower values than in case of NEDC.

$$\Delta CO_{2\mathrm{wltc}} = \Delta CO_{2\mathrm{nedc}} \cdot \frac{11{,}03}{23{,}27}, \qquad (1)$$

The tested vehicle is conditioned before being tested at a constant ambient temperature, being specific to test procedure. For the NEDC, the temperature range during the start is between 20 °C and 30 °C, while for the WLTC the range is 23 ± 5 °C. As an example, the engine starting coolant temperature impact on CO_2 emissions (gCO_2/km) in NEDC, relative to standard starting condition is presented in Figure 2 [3]. Carbon dioxide emissions can be increased by up to 19% when starting from -7°C and decreased by roughly 12% when the engine is fully warmed.

Table 1 Comparative specification of NEDC and WLTC

	Units	NEDC	WLTC
Start conditions		cold	cold
Duration	s	1180	1800
Distance	km	11,03	23,27
Mean velocity	km/h	33,6	46,5
Max. velocity	km/h	120	131,3
Stop phases		14	9
Duration of stops	s	280	226
Duration of constant speed	s	475	66
Duration of acceleration	s	247	789
Duration of deceleration	s	178	719
Percentage of stops	%	23,7	12,6

Percentage of constant speed	%	40,3	3,7
Percentage of accelerations	%	20,9	43,8
Percentage of deceleration	%	15,1	39,9

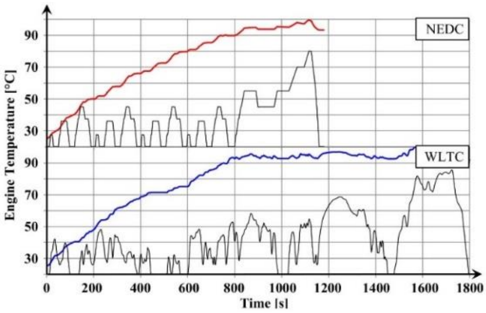


Figure 1 Simulated engine temperature along NEDC and WLTC for DISI-ICE powered vehicles [2]

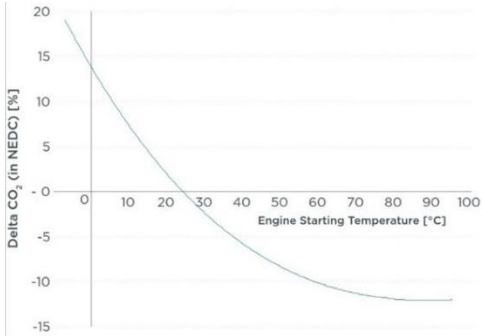


Figure 2 Engine starting temperature impact on CO2 emissions in NEDC (11.03 km) [3]

1.1 Combustion process stability in forced heating regimes of a 3WC

Emission test procedures usually incorporate specific phase during which the engine idles shortly before take-off. This stage is further divided into two parts with completely different requirements. The first one, lasting a few seconds, is the starting of the engine, which should enable reliable starting of the engine to idling. The next one is the TWC heating phase during which the catalyst is supposed to reach the temperature at which it may operate, taking few tens of seconds. The initial idle phase during the cold start period takes roughly 11 s in test procedures based both on both NEDC and WLTC. For example, even in the case of standard pre-conditioned ambient temperature of roughly 25 °C, this is far from sufficient to heat the TWC over 250 °C at which it can operate at efficiency of as low as 50%. Therefore, most of the emissions during the test cycle come from the initial critical part of the cycle.

Generally, forced TWC warm-up strategies through dedicated combustion control algorithms, rely on the combustion phase-out from optimal advanced to suboptimal (inefficient) retarded ignition (aTDC), yielding higher exhaust temperatures and enthalpy due to late combustion process that can even extend into the exhaust stroke.

The main issue in terms of control the combustion in the expansion stroke is that the mixture pressure and the temperature are continuously decreasing during the combustion delay phase which is critical and adversely affects the formation of the flame kernel and stable flame propagation in subsequent combustion stages. The heat losses are also critical due to continuously increasing combustion chamber surface during expansion superimposed to already low combustion chamber walls temperatures. Therefore, misfire represented statistically by largely increased deviation of all combustion dynamics parameters, is ever present, but certainly inadmissible from the aspect of exhaust emissions.

This strategy may benefit from Split Direct Injection (SDI) and produce larger combustion phase-out. Provided the optimized SDI combined with mixture turbulence, rich and ignitable mixture around the spark electrodes may be produced, enabling more stable combustion process even at retarded ignition. Balance between the late ignition and late combustion based on SDI from which TWC warm-up time surely benefits and affected efficiency and emission will depend to specific boundary conditions (specific application and test requirements) [4].

2 TESTBED SETUP AND PRODUCERS

The tests which are to simulate catalyst heating was performed on a Direct Injection Spark Ignition Single Cylinder Research Engine (DISI-SCRE). Test bed configuration with engine auxiliaries is presented in Figure 3. The engine is of modular type allowing engine components change according to the test specification (piston, cam shafts, injectors etc.). Engine systems are external (cooling, lubricating, charging, etc.), independent of the engine itself and independent and flexible in terms of control and adjustments according to the test specification. Charging is realized by a separate compressor, while the exhaust back-pressure (flow resistance and pressure drop over charger turbine) is controlled using external turbocharger turbine simulator. The engine technical specification is given in Table 2.

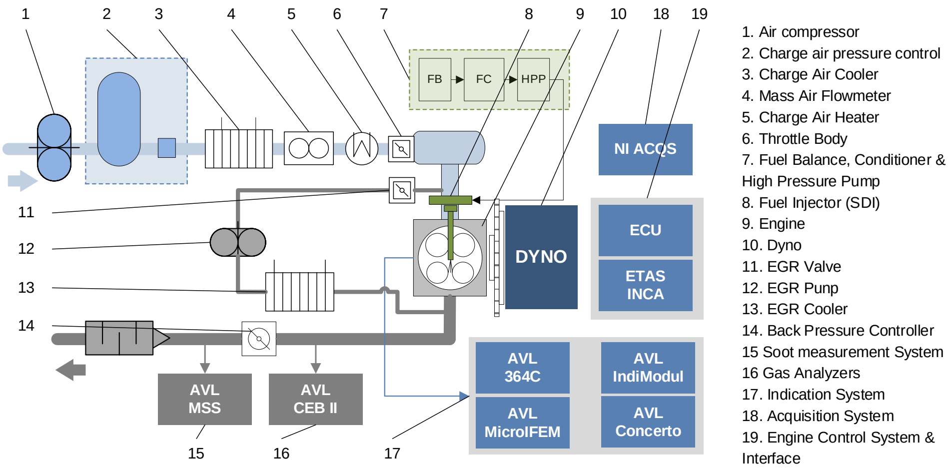


Figure 3 Schematic diagram of the test bench

Table 2 SCRE technical specification

Parameter	Nomenclature	Value	Unit
Bore	D	82	mm
Stroke	S	90	mm
Working displacement	V_h	475	cm^3
Conrod length	L	144	mm
Pin offset	b	-0.5	mm
Compression ratio	ε	11.5	-
Inlet valve opening	IVO	30	^{o}CA aTDC
Inlet valve closing	IVC	40	^{o}CA aBDC
Exhaust valve opening	EVO	44	^{o}CA bBDC
Exhaust valve closing	EVC	13	^{o}CA bTDC

3 TEST PROCEDURES

A specific catalyst heating test procedure – CatHeat – was designed as to investigate and evaluate the influence of injection and ignition strategy on the capability to provide higher exhaust temperature and in turn faster catalyst warm-up during the low-temperature/low-load operation specific to the starting sequence of a reference driving cycle. In this work, double injection strategies were tested and analyzed against combustion dynamics and combustion stability parameters obtained from in-cylinder pressure indication and thermodynamic post-processing. Setup parameters covered within separate group of variation tests are as follows:

- The start of the first injection sweep (SOI1);
- The start of the second injection sweep (SOI2);
- The ratio of the fuel mass injected during the first and second injection - Split Factor sweep (SF);
- Fuel injection pressure or fuel rail pressure sweep (RP).

Engine warm-up, under conditions in which the engine parts, coolant and lubricating oil temperatures are close to the ambient temperature is a critical part of the standard driving cycle. Therefore, the oil and coolant temperatures were maintained at 35 °C during the test. The engine crankshaft speed (n) and indicated mean effective pressure (IMEP) were set to the expected average values at engine idle. Maintaining the IMEP at a given level is achieved by means of the throttle position. Mixture strength was maintained stoichiometric during test campaign by controlling the injection timings (Δt_{INJ1}, Δt_{INJ2}). Being focused on injection strategies exclusively, spark advance was set constant at and shifted after TDC, as to enable the combustion shift further into expansion stroke and so increase the exhaust temperature. Each of the 4 tests yielded one optimal setup for the varied parameter, which was then adopted as a constant in subsequent tests. The initial values and ranges for given parameters in the tests are given in Table 3.

Table 3 The initial engine setup

Description	Designation	Unit	Initial Value	Variation range
Engine speed	n	min^{-1}	1200	-
Indicated Mean Effective Pressure	IMEP	bar	2	-
Spark angle	SA	°CA aTDC	10	-
Excess air coefficient	λ	-	1.00	-
Start of the first injection	SOI1	°CA bTDC	330	-330…(5)…-245
Start of the second injection	SOI2	°CA bTDC	235	-250…(5)…-70
Fuel rail pressure	p_RAIL	bar	100	50…(25)…200
Fuel mass split factor	SF	$\%m_{F,INJ1}$ vs. $\%m_{F,INJ2}$	[70_30]	[80_20]…[40_60]

3.1 Measurements and data analysis

Usual representation of combustion dynamics based on identification of angular positions where predefined fraction of the fuel/mixture is burned is used. Burn rates are derived and analyzed upon thermodynamic analysis of the in-cylinder indicated pressure traces, as follows:

Burn rate angles – the CA at which burn rate percentages of 5, 10, 50 and 90% are determined (MBF 05, MBF 10, MBF 50, MBF 90);

Combustion Duration (CD);

Statistical representation of combustion stability in form of coefficient of variance (CoV_05, CoV_10, CoV_50, CoV_90).

To eliminate the ever-present sudden changes and noise in indicated pressure measurements, fast data smoothing is applied using moving average on interval of ±2°CA. Considering low engine speed and stationary operation, the pressure transducer output was referenced to readings from intake manifold pressure transducer at BDC [10].

The analysis was performed using the AVL Concerto software package. Simplified method for the evaluation of the heat release rate is based on Rassweiler and Withrow method and improved lately by Hohenberg and Killman [11-13]. Assuming thermodynamic state change split in two quasi-stationary equilibrium steps – adiabatic compression (expansion) and isochoric heat supply:

$$p_{i+1,s} = p_i \cdot \left(\frac{V_i}{V_{i+1}}\right)^{\kappa_i}, \tag{2}$$

$$\Delta Q_i = m \cdot c_{v,i} \cdot \left(T_{i+1} - T_{i+1,s}\right). \tag{3}$$

This yields, upon simple transformation, expression for finite heat change:

$$\Delta Q_i = \frac{1}{\kappa_i - 1} \cdot V_{i+1} \cdot \left[p_{i+1} - p_i \cdot \left(\frac{V_i}{V_{i+1}}\right)^{\kappa_i}\right]. \tag{4}$$

Isentropic exponent depends on temperature and can be determined starting from the basic thermodynamic relation for specific heat capacities while specific heat capacity is approximated using empirical relation [12,13] and discrete adjustment for SI-ICE and homogenous charge (A=0,1):

$$\kappa_i = \frac{c_{pi}}{c_{vi}} = \frac{R + c_{vi}}{c_{vi}} \approx \frac{0{,}2888}{c_{vi}} + 1 \approx \frac{0{,}2888}{0{,}7 + \frac{p_i \cdot V_i}{m \cdot R} \cdot (0{,}155 + A) \cdot 10^{-3}} + 1. \tag{5}$$

This evaluation method is not featured by particularly high absolute accuracy, having in mind originally neglected heat transfer in combustion chamber. Short computing time and only a few external data required for processing make this method first choice for fast evaluation, on-line monitoring, and purposes of relative comparison. Accuracy of burn rate parameters determined in this way, however, is sufficient for comparative analysis [12,13].

The start and the end of combustion, based exclusively on theoretical approach are to be determined as the crankshaft positions where derivate of heat flux dQ/dα changes from zero to positive and positive to zero, respectively. In discrete methods, ignition delay will be usually represented by 1, 2 or 5% MFB. Combustion duration is calculated as the crank angle between the end of the ignition delay and typically 90, 95 or 99% MFB. Determining small or large percentages such as 1 or 99% can be difficult and insufficiently accurate due to the susceptibility of the calculation to the effects of noise with small pressure changes.

Considering constant spark advance for all tests, the start and the end of combustion are arbitrarily defined as MBF05 and MBF90, and consequently, the combustion duration is calculated as the difference of two:

$$CD = MFB90 - MFB05. \quad (6)$$

Considering combustion cycle during warm-up critical in terms of stability, statistical evaluation of variation of each parameter of interest (x) is performed based on the coefficient of variation (CoV) defined as the root-mean-square deviation (σ_x) normalized to mean value (μ_x):

$$CoV_x = \frac{\sigma_x}{\mu_x} = \frac{\sqrt{\frac{1}{n-1}\left(\sum x^2 - n\cdot {\mu_x}^2\right)}}{\mu_x} = \frac{\sqrt{\frac{1}{n-1}\left(\sum x^2 - \left(\sum x\right)^2\right)}}{\frac{1}{n}\sum x}. \quad (7)$$

4 RESULTS AND DISCUSSION

4.1 Start of injection – SOI1 test

The SOI1 sweep was performed in the range of -330…-245 °CA (ΔSOI1=5 °CA), while the other relevant parameters were set according to the starting values given in the Table 3. Figure 5 depicts the indicated mean effective pressure (IMEP) and the combustion duration (CD) vs. start of the first injection (SOI1), and additionally corresponding coefficients of variation for both parameters (CoV_IMEP and CoV_CD, respectively).

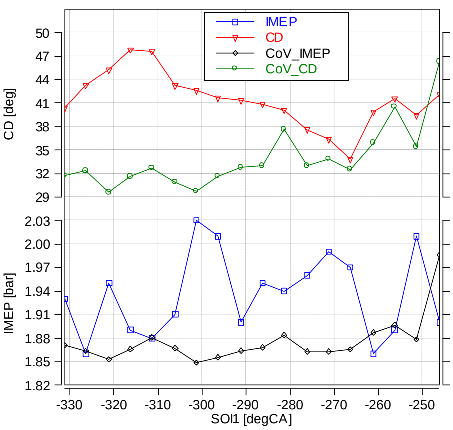


Figure 5 Indicated mean effective pressure (IMEP, CoV_IMEP), and combustion duration (CD, CoV_CD) vs. SOI1

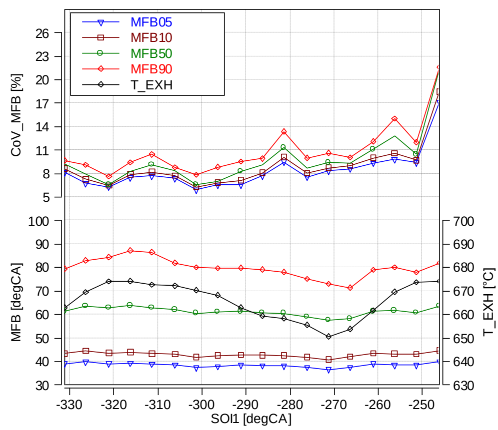


Figure 6 Mass fraction of burned fuel (MBF, CoV_MFB) and exhaust temp. vs. SOI1

The CoV_IMEP is considerably high ranging from 10 to 32% (15% for initial setup) which is up to 20 times higher value than usual for combustion process of the fully warmed-up SI engine. Significant change is observed for the case of the latest SOI1 (-245 °CA), indicating instable combustion due to compromised mixture formation caused by interaction of the fuel spray and intake valve. This correlates well with increased combustion duration covariance (CoV_CD). Combustion duration peak is observed in the range -320…-310 °CA, while the minimum is found around -265 °CA. Figure 6 provides information on combustion dynamics. The MFB angles remain unchanged in observed SOI1 interval, but slightly better combustion stability is documented around SOI1=300 °CA. The CoV traces are fairly similar to each other but indicate the most stable operation in the SOI1 range of -320 and -300 °CA. Also, instable operation and misfire indicated by CoV_CD at the end of the SOI1 interval is confirmed by increased CoV values for all four values of MFB. The highest exhaust temperature (~675 °C), if neglected the local maximum observed at the end of the SOI1 range featured by most instable combustion, is also spotted in the SOI1 range of -320…-300 °CA. Considering slightly improved overall combustion stability around SOI1=300 °CA, this setup was adopted for the next test step.

4.2 Start of injection – SOI2 test

Based on the results of the previous analysis, with fixed SOI1=300 °CA, SOI2 sweep was performed within the range of -250...-70 °CA (ΔSOI2=5 °CA), keeping constant split factor (SF), spark advance (SA) and fuel rail pressure (p_RAIL) as defined in Table 3.

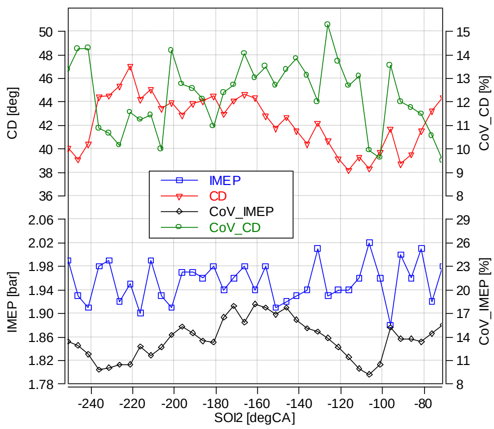


Figure 7 Indicated mean effective pressure (IMEP, CoV_IMEP), and combustion duration (CD, CoV_CD) vs. SOI2

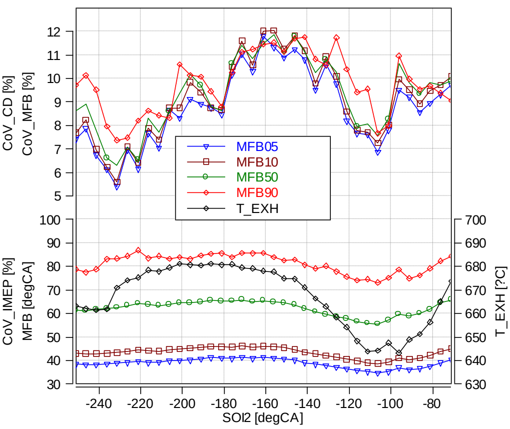


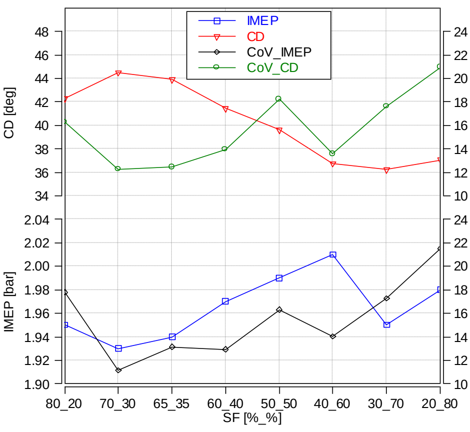
Figure 8 Mass fraction of burned fuel (MBF, CoV_MFB) and exhaust temp. vs. SOI2

In Figure 7, indicated mean effective pressure (IMEP), combustion duration (CD) and coefficients of variation for both parameters (CoV_IMEP and CoV_CD, respectively) are presented vs. start of the second injection (SOI2). The combustion stability indicated by the covariance of indicated pressure proved to be less prone to the influence of SOI2 change than it was in the case of SOI1 variation test. The CoV_IMEP was determined in the range of 10…18%. Considering split factor 70%-30%, results may point to the conclusion that variation of the SOI2 for 30% of injected fuel doesn't affect significantly the overall quality of mixture formation. The local minimums of the combustion duration (CD) were determined at the very start of the SOI2 range (-250…-245 °CA) and just before the piston reaches the mid-stroke (-110…-100 °CA) indicating most probably best results in mixture homogenization. Best results for combustion duration covariance are found close to the first minimum of combustion duration (-235…-200 °CA), but also at the very end of the SOI 2 range. Results of the combustion dynamics analysis presented in Figure 8 also indicate two narrow ranges in which MFB angles get local minimum corresponding well to the results for combustion duration. However, covariance traces point clearly at the start of the SOI2 range where covariances of all four MFB angles reach absolute minimums without exception. By analyzing the exhaust temperature measurements, it appears that this range doesn't provide the highest results (~660 °C), but reasonably close to the absolute maximum (~680 °C), while maintaining overall better combustion stability. Considering that, SOI2=-235 °CA was determined setup for further test step.

4.3 Split factor test

The split factor (SF) sweep was performed by retaining unchanged spark advance and fuel rail pressure as defined in Table 3 and applying SOI1 and SOI2 optimums from previous tests (-300 and -235 °CA). Eight discrete fuel split factors (%mF,INJ1 vs. %mF,INJ2) were tested, namely 80-20, 70-30, 65-35, 60-40, 50-50, 40-60, 30-70 and 20-80.

In Figure 9, indicated mean effective pressure (IMEP), combustion duration (CD) and coefficients of variation for both parameters (CoV_IMEP and CoV_CD, respectively) are presented vs. injected fuel split factor. Combustion stability was determined in the range of 11 to 22%, reaching two local maximums – one at the very start of the of the range (80-20, 18%) and the second at the very end (20-80, 22%). Combustion duration shows strong dependance on split factor, being determined in the range of 36…42 °CA, while covariance was found in the range of 12,5% to 21%. By increasing the fraction of the second injection, the combustion duration decreases which indicates that fuel concentration in the upper parts of the combustion chamber, close to the spark plug, was evidently higher. Increased covariances of both parameters indicate compromised homogenization. Also, increased covariances at the start of the SF range points to almost equally poor mixture homogenization, most probably caused by fuel spray collision with piston crown since SOI1 is close to TDC.



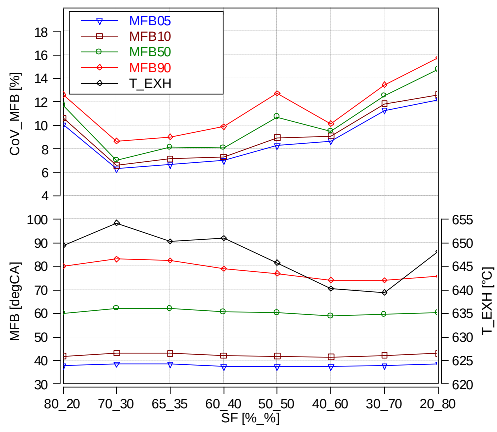


Figure 9 Indicated mean effective pressure (IMEP, CoV_IMEP), and combustion duration (CD, CoV_CD) vs. split factor (SF)

Figure 10 Mass fraction of burned fuel (MBF, CoV_MFB) and exhaust temp. vs. split factor (SF)

Heat release rate analysis presented in Figure 10 yields results similar to those gathered in previous SOI sweeps – MFB05 and MFB10 show no particular dependency to split factor, while MFB90 decreases with increased fraction of the second injection (local minimum MFB90=72 °CA at SF=40-60). Covariance traces follow the same dependence to SF reaching best values at SF=70-30 (7-8,5%) and indicating poor homogenization for increased portions of second injection (SF=40-60…20-80). The best covariance results for heat release rate parameters, expectedly correspond well to the highest exhaust temperature measured in this test (~654 °C). For final fuel rail pressure variation test, SF=70-30 setup was adopted.

4.4 Fuel rail pressure (p_RAIL) test

Introducing the direct fuel injection in SI-ICE range opened wide range of problems previously exclusively related to mixture formation in CI-ICE, namely jet cone development, jet penetration, primary and secondary break-up, droplet break-up, collision and coalescence, flash boiling and evaporation. All named phenomena are highly dependent on changing thermodynamic conditions during intake and compression strokes and even more on injection pressure [14,15].

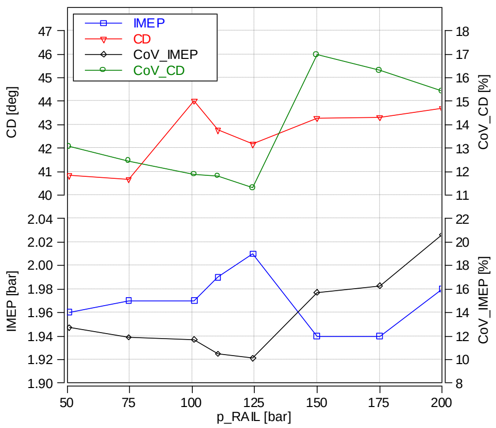


Figure 11 IMEP, IMEP_CoV, BDUR and BDUR_CoV vs. rail pressure(p_RAIL)

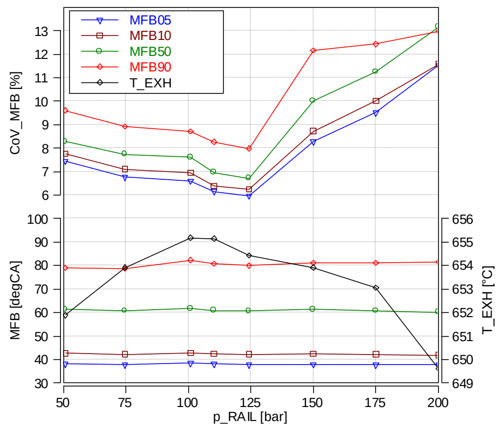


Figure 12 MBF, MBF covariance and exhaust temperature vs. rail pressure(p_RAIL)

Fuel rail pressure sweep was performed in the range of 50…200 bar (Δp_RAIL=25 bar). The low-to-mid pressure range is obviously favorable due to more energy efficient fuel injection process but also are more suitable for injection strategies performed during intake stroke and/or during the low-pressure range of the compression stroke. Setups gathered from previous tests meet these conditions and final setup for p_RAIL is expected to be found in the same pressure range.

In Figure 11, indicated mean effective pressure (IMEP), combustion duration (CD) and coefficients of variation for both parameters (CoV_IMEP and CoV_CD, respectively) are presented vs. rail pressure. The best overall process stability was determined between 100 and 125 bar. Combustion duration covariance minimum was also localized in that range. As for combustion heat release rate all MFB values proved to be less dependent to rail pressure variation, but covariances shown improvement in the range of 100…125 bar. Optimal rail pressure setup was identified by introducing additional test point at 110 bar giving further improvement in combustion stability. Covariances for MFB parameters were all found in the range of 6-8%, IMEP covariance was decreased to 10,5%, while retaining exhaust temperature at the same initial level (655 °C, compared to 660 °C).

5 CONCLUSIONS

Double injection strategies in terms of injection control parameters were tested and analysed against combustion dynamics and stability in idle operating conditions during the starting phase of the standardized driving cycle. Stationary test designed around the constant low engine thermal load was used to analyse possibilities to improve stability in conditions of late combustion used to increase exhaust temperature during engine and catalyst warm-up.

The research results and analysis were limited to gain information on capabilities of double injection strategies to improve combustion stability in idling warm-up conditions. The influence of valve timings, spark advance and emission results were not considered at this early stage and are part of the further optimization.

The sequence of variation tests was performed on the research single cylinder direct injection spark ignition engine on a constant engine load and speed corresponding to the idle operation. Full sweeps of four basic injection control parameters – start of injection (1^{st} and 2^{nd}), injected fuel mass split factor and fuel rail pressure were performed while maintaining constant engine load, speed, mixture strength, valve timings and spark advance. Combustion heat release rate was analysed using fast, simplified evaluation method based on Hohenberg approach. Combustion stability was evaluated upon covariance of indicated mean effective pressure, combustion duration and fuel mass fraction burned values (5,10,50 and 90%).

Significant improvement in terms of IMEP covariance was seen in case of SOI1 sweep. Shifting injection from -330 to -300 °CA improved homogenization and gave CoV_IMEP decrease from 15 to 10%, and a bit smaller decrease was also seen in case of CoV_MFB parameters. The exhaust temperature increased from 662 to 670 °C. Variation test of SOI2 provided small step in setup from initial -250 to optimal -235 °CA. The heat release rate hasn't changed significantly (MFB05-MFB50 remained unchanged), but combustion stability gained some improvement, particularly at the very start of combustion process (Cov_MFB05 and CoV_MFB10). Small decrease in exhaust gas flux was reported.

Fuel mass split factor variation test gained no particular improvement compared to initial setup (70-30). Increasing the fuel mass fraction of the second injection proved to be inefficient mainly due to compromised homogenization. Like the SOI2 sweep results, this is partially caused by the fact that spark advance remained constant throughout all tests and evaporation time was decreased gradually for each step-up in fuel mass introduced in second injection.

Fuel rail pressure sweep provided new setup which was expected upon theory of mixture formation. Compared to initial pressure of 50 bar, new setup at 110 bar gained improvement in terms of CoV_MFB parameters, again particularly for MFB05 and MFB10 while keeping exhaust temperature and CoV_IMEP unchanged.

REFERENCES

[1] Shelef, M. and Mccabe, R. W.: "Twenty-five years after introduction of automotive catalysts: what next?" Catal. Today 2000, 62, 35–50; doi:10.1016/S0920-5861(00)00407-7.

[2] Morra, E.P., Ellinger, R., Jones, S., Huss, A., Albrecht, R.: "Tank-To-Wheel CO2 Emissions Of Future C- Segment Vehicles." In: Liebl, J. (eds) Der Antrieb von morgen 2014. Proceedings. Springer Vieweg, Wiesbaden. https://doi.org/10.1007/978-3-658-23785-1_8

[3] Mock, P., Kühlwein, J., Tietge, U., Franco, V., Bandivadekar, A. and German, J.: “The WLTP: How a new test procedure for cars will affect fuel consumption values in the EU,” 2014, doi: 10.1016/j.enpol.2013.12.013i.

[4] Hedinger, R., Elbert, P. and Onder, C.: “Optimal cold-start control of a gasoline engine,” Oct. 01, 2017, MDPI. doi: 10.3390/en10101548.

[5] Lee, D.: “Effects of Secondary Air Injection During Cold Start of SI Engines”, Ph.D. Thesis, Massachusetts Institute of Technology, Cambridge, MA, USA, 2010.

[6] Hallgren, B.E.: “Impact of Retarded Spark Timing on Engine Combustion, Hydrocarbon Emissions, and Fast Catalyst Light-Off”, Ph.D. Thesis, Massachusetts Institute of Technology, Cambridge, MA, USA, 2005.

[7] Koehlen, C., Holder, E., Vent, G.: “Investigation of Post Oxidation and Its Dependency on Engine Combustion and Exhaust Manifold Design”, SAE Tech. Pap. 2002, 7, 1–14.

[8] Schatz, A., Knorr, T., Ellmer, D.: “Elektrischer Heizkatalysator zur Optimierung der Emissionen von Mildhybridsystemen”. MTZ-Motortech. Z. 2016, 77, 146–153.

[9] Pfahl, U., Schatz, A., Konieczny, R.: !Advanced Exhaust Gas Thermal Management for Lowest Tailpipe Emissions-Combining Low Emission Engine and Electrically Heated Catalyst”, SAE Tech. Pap. 2012, 24,2924–2931.

[10] Randolph, A. L.: “Methods of Processing Cylinder-Pressure Transducer Signals to Maximize Data Accuracy,” SAE Paper 900170, 1990.

[11] Brunt, M. F. J., Pond, C. R.: “Evaluation of Techniques for Absolute Cylinder Pressure Correction,” SAE Paper 970036, 1997

[12] Hohenberg, G., Killman, I.: “Basic Findings from Measurement of the Combustion Process”, XIX Fisita Congress, Melbourne, Australia, 1982, paper 82126

[13] Tomić, M. V., Popović, S. J., Miljić, N. L., Petrović, S. V., Cvetić, M. R., Knežević, D. M., & Jovanović, Z. S.: “A quick, simplified approach to the evaluation of combustion rate from an internal combustion engine indicator diagram”, Thermal Science, Vol. 12, No. 1, 2008, pp. 85-102, (ISSN 0354-9836), https://doi.org/10.2298/TSCI0801085T

[14] Baumgarten, C.: “Mixture Formation in Internal Combustion Engines” (Ed. Heat and Mass Transfer), Springer Verlag 2006, ISBN-10 3540308350

[15] Stiesch, G.: “Modeling Engine Spray and Combustion Processes” (Ed. Heat and Mass Transfer), Springer Verlag 2003, ISBN 978-3-642-05629-1