Injection strategies for low HC raw emissions in SI engines with CNG direct injection

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Abstract The energy source natural gas represents an attractive alternative fuel for SI engines due to its suitable chemical properties. But the lower performance yield and the insufficient infrastructure lead to a rejection by the customer. One approach to gain acceptance is the improvement of the driveability of the natural gas-powered vehicles. Direct injection of CNG is a technology with potential to achieve this objective. Nevertheless, the combustion concept has high requirements on injection system and mixture formation. Among other challenges the methane slip raises major issue. The compact molecular structure of the CNG's main component, methane, leads to an extremely slow reactivity, which expresses itself in a higher catalytic converter light-off temperature and thus aggravated exhaust aftertreatment. Especially, since methane's global warming potential is 21 times higher than CO₂, the greenhouse gas footprint is getting worse again. Thus, for the future implementation of such a combustion system concept it will be necessary to compensate the limited conversion rates of the catalytic converter with low raw emissions. For this, the comprehension of the mixture formation is necessary. This paper deals with different injection strategies at part load and charged operation to

achieve HC raw emissions as low as possible. The test-bench experiments are executed with a two-cylinder SI engine under homogeneous stoichiometric conditions. The DI injector used for this study is a prototype model fabricated by Robert Bosch GmbH. In summary, the need of adaption of the injection strategy to the respective engine map area is becoming evident. Furthermore, various injection timings and injection splitting strategy turned out as an appropriate measure to make the most of CNG-DI's potential and to achieve low HC raw emissions.

Keywords CNG-DI · Start of injection · Multiple injections · Injection pressure

1 Introduction

The sustained need for research and development work to optimize internal combustion engines is promoted by the increasing energy consumption worldwide [1] as well as by the more and more stringent emission standards [2]. Hence, to comply with EU6, automobile manufacturers have to tap their full potential to meet the required fleet fuel consumption. Nevertheless, regarding this further aggravation of emission limits this target is getting more and more difficult and expensive to achieve using the conventional crude oil based fuels [3, 4]. That is why there is no way to go past diversification of energy sources in the transport sector.

In the short and medium term, natural gas represents the most promising alternative fuel to gasoline. Depending on well natural gas is a mixture of gases consisting mostly of methane (CH₄). There are two types to store natural gas. Either as LNG—liquefied natural gas—or CNG—compressed natural gas [5]. For passenger cars' only CNG is a

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reasonable choice due to its convenient handling. The specific properties of CNG like reduced exhaust emissions, 25 % lower CO₂ emissions, knock resistance, availability and also the price advantage make this fuel attractive for engine application [1, 6–9]. However, despite increasing sales of CNG powered cars in Germany, this environmentally friendly propulsion still carves out a niche existence [3].

The main reasons for this weak customer demand include both, the insufficient infrastructure of refueling stations and the ca. 10 % lower performance yield of the offered engines compared to gasoline [3, 8, 10]. This lower power output is attributed to the current use of port fuel injection systems. Due to the fuels' gaseous state CNG displaces fresh air in the order of ca. 10 %. If this displacement effect cannot be compensated by turbocharging, a lack of torque emerges. Especially, at low engine speed this deficit of the so called low-end torque has an adverse effect on the driveability.

One approach to increase the driveability and by association the customers' acceptance is to switch to an internal fuel injection with CNG-DI. This technology avoids the fresh air displacement by direct fuel injection into cylinder after intake valve closing, thus increasing the volumetric efficiency [10–16].

But to realize such a combustion system there is still a major need for research. Among the high requirements on the injection system like large flow cross-section, high reliability and also leak tightness, the minimization of the methane slip poses a big challenge. Methane slip is defined as the emission of unburned hydrocarbons (UHC) through incomplete combustion, which mostly consist of methane, the main CNG component. Its stable molecular structure leads to the knock resistance but also to extremely slow reaction rates challenging the exhaust gas aftertreatment. Especially, this manifests itself in higher light-off temperature for hydrocarbons and reduced catalytic long-run activity [10, 11, 17]. Since methane's global warming potential is 21 times higher than of CO₂ [17], the greenhouse gas footprint is getting worse again. Thus, for the future implementation of this combustion system it will be necessary to meet the limited conversion rates with low raw emissions. For this, the knowledge of influences on HC sources in combustion chamber is unalterable to take engine internal measures for reduction.

Figure 1 represents schematically the main sources for UHC. The primary reason for these emissions can be attributed to quenching effects. Those occur at the combustion chamber walls (1) due to increased heat loss from flame to wall or in crevices like the top land (2) through restrained flame propagation [18]. But also misfire and bulk quenching (3) through inhomogeneous charge distribution have to be taken into account. Also the storage of HC in

Fig. 1 UHC sources in combustion chamber according to [19]

intake port with subsequent scavenging into the exhaust port (4) or possible storage in piston deposits or lubricant film (5) can be mentioned as reasons.

The manifestation of the individual causes can be influenced by design and calibration measures. The design ones are piston shape including top land crevice and compression ratio as well as the injection system. The calibration parameters contain the valve timings, center of combustion, mixture composition (λ , EGR), coolant and lubricant temperature and not least the injection strategy.

In particular, this paper focuses on the injection strategy, which is influencing several HC sources simultaneously. This includes the start and splitting of injection as well as the injection pressure. In recent past some studies were already conducted to investigate these parameters. The scientific works in [14, 16, 19] varied the start of injection using CNG-DI. It was determined that the optimal injection timing regarding best fuel consumption, lowest emissions and shortest burn duration is around the bottom dead center. As soon as the injection was shifted towards ignition timing, the engine performance got worse. A similar behavior was also observed in [5, 20–26]. The general explanatory approach was the ever shorter mixture formation time until the ignition due to the later injection timings, which led to an inhomogeneous mixture. Additionally, associated CFD simulations proved an increase of local turbulence originating from the jet during the injection event. However, this induced turbulence decreased rapidly after the end of injection and couldn't be maintained until the ignition, if the start of injection was very early. Furthermore, an important role of the cylinder internal flow was identified. This determines significantly the mixture distribution in the combustion chamber due to the low gas density. All the investigations mentioned above were carried out with a side mounted injector. But the study [23] showed a comparable behavior also with a central mounting position. In [27] the influence of injection pressure at different engine speeds was examined. It became evident that at lower engine speed an increased injection pressure showed better performance due to higher

volumetric efficiency as well as providing sufficient mixing time for the air–fuel charge. At high engine speed, the dominant mechanism was the mixing rate where the lower injection pressure resulted in better mixture formation since the injection duration was longer. In addition to the injection pressure and timing [28] was concerned with a double injection. However, it was found that the emission levels of a single injection could not been neither reached nor undercut.

Most of the recent investigations have in common that their main attention was on the achievable power output with CNG-DI while the emissions were considered marginally. If, nonetheless, emissions were taken into account most of the studies used side mounted position, an atypical swirl concept for SI engines or even examined several partial aspects. Also uncertainties due to failing hardware emerged in some works impeding clear statements of the influence of injection strategy.

The present study mainly concentrates on a holistic view of the influence of start of injection, multiple injection and injection pressure on the emissions with central mounted injector. This approach contributes to an extended gain of knowledge for the combustion system with CNG-DI.

2 Experimental setup and methodology

The test engine is a two-cylinder SI engine manufactured by Weber Motor [29]. The engine is originally designed for gasoline application and is used for CNG for this study. Due to this fact the peak firing pressure is limited to 100 bar. Nevertheless, the authors are aware that for MBT operation CNG demands higher cylinder pressures. Therefore the investigated points are chosen with respect to this restriction. Further specifications of this engine are given in Table 1. The engine is equipped with water-cooled piezoelectric cylinder pressure sensor (type 6041B) as well as with piezoresistive sensors (type 4005B and type 4049A) for low pressure indication, made by Kistler [30–32]. The data acquisition runs with the Indimaster Advanced from AVL [33] including 32 channels with a resolution from 0.1 to

0.5 °ca. For each operation point 200 cycles are measured and evaluated with the internal gas exchange- and combustion cycle analysis software BeCAT [34].

The experiments are exclusively executed with the gas G20, consisting of 99.5 % methane. A prototype from Bosch serves as injector for the gaseous fuel. It is a modified multi hole inward opening solenoid injector for gasoline. Taking the distinct lower density of methane (0.72 kg/m³) into account the nozzles' flow rate is increased by larger hole diameter. The mass flow of a compressible medium through a nozzle is defined by St. Venant and Wantzel as follows (Eq. 1):

$$\dot{m} = A_{\text{eff}} \cdot \sqrt{2 \cdot p_{\text{inj}} \cdot \rho} \cdot \sqrt{\frac{\gamma}{\gamma - 1} \cdot \left[\left(\frac{p_{\text{cyl}}}{p_{\text{inj}}} \right)^{\frac{2}{\gamma}} - \left(\frac{p_{\text{cyl}}}{p_{\text{inj}}} \right)^{\frac{\gamma+1}{\gamma}} \right]} \quad (1)$$

Thus, it can be found that the mass flow rate $\dot{m}$ depends on effective cross-section A_{eff} , current pressure ratio $p_{\text{inj}}/p_{\text{cyl}}$ and as well as on substance properties ρ and γ . Since the critical pressure ratio (2) is reached, the jet speed is increased to the speed of sound in the smallest cross-section. As a result, the mass flow rate becomes maximized, regardless of the current cylinder pressure. A further increase is only possible with higher injection pressure or larger cross-section (3).

$$p_{\text{crit}} = \frac{p_{\text{cyl}}}{p_{\text{inj}}} \leq \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}} \approx 0.54 (\text{mit } \gamma_{\text{CH}_4} = 1.31) \quad (2)$$

$$\dot{m}_{\text{max}} = A_{\text{eff}} \cdot \sqrt{2 \cdot p_{\text{inj}} \cdot \rho} \cdot \sqrt{\frac{\gamma}{\gamma + 1}} \quad (3)$$

In the present investigations the injection pressure is at least 15 bar, so the critical pressure ratio can be exceeded only in case of a late injection during the compression stroke or at higher loads. In all other cases the injection flow is supersonic and reaches a constant rate of 3 kg/h at $p_{\text{inj}} = 20$ bar for the used injector.

The determination of raw emissions is made by the exhaust analysis system MEXA-7100D from HORIBA [35]. This device measures the concentration of HC, CO, CO₂, NO_x and also O₂. Additionally, HC emissions in the intake manifold are determined with the Fast Response FID 123ISS from Testa [36].

Two operation points are specified for these experiments. A typical NEDC operation point with $n = 2000$ rpm, $\text{bmep} = 2$ bar is chosen for part load operation. Even if the relevance of NEDC will decrease in future this operation point still represents the issue of low exhaust temperatures for the aftertreatment system in part load. The second point $n = 2000$ rpm, $\text{bmep} = 16$ bar covers the charged conditions with scavenging, which are relevant for CNG-DI.

Table 1 Specifications of Weber MPE 850 DOHC [29]

Cylinder number	2
Bore × stroke	89 mm × 68 mm
Displacement volume	846 cm ³
Compression ratio	10.5:1
Peak firing pressure	100 bar
Valve train	DOHC
Charging system	TC, eWG
Injection system	DI, c/m

The injection pressure varies from 15 to 150 bar whereby the higher load is only feasible with pressure of $p_{inj} \geq 70$ bar. The symmetrical valve overlap (VOL) remains constantly at $VOL = 30^\circ \text{ca}$. For both points a variation of start of injection (SOI) is performed. For this purpose the sweep starts with $SOI = 360^\circ \text{ca b. ITDC}$, which is retarded continuously by 20°ca until misfires occur. The multiple injection is implemented with $p_{inj} = 20$ bar and by splitting the injection into two parts. The splitting ratio (mass 1st inj./mass 2nd inj.) varies from 100/0 to 5/95 %. For each ratio the end of injection (EOI_2) of the second one is varied while the first start of injection remains fixed.

The global equivalence ratio is always $\phi = 1.0$. The results presented below are averaged over three repetitions with a respective measuring time of 45 s.

3 Engine results: influence of injection strategy

3.1 Influence of start of injection and injection pressure

Figure 2 shows the emissions of HC, CO and the engine running smoothness (COV_{imep}) over the end of injection (EOI) for different pressure values at part load. Due to various injection durations respective to the pressure the end of injection suites better for a comparison. The first value corresponds always to the $SOI = 360^\circ \text{ca b. ITDC}$, thus the injection duration for each pressure can be derived from the graphs.

As it is apparent from the HC emissions the influence of injection pressure on both, qualitative and quantitative curve progression, is of minor importance. The lowest raw emissions can be found at EOI around the bottom dead center. Retarded EOI's closer to the ignition timing supply insufficient fuel–air mixing, incomplete combustion and thus disproportionally raised HC emissions and also raised CO and COV_{imep} . Furthermore the range for HC minimum is wider at higher pressures and the level slightly lower.

But from $p_{inj} = 70$ bar no difference is existing anymore. This wider operation window with higher pressure levels can be traced to the injection duration. Concerning all examined injection pressures the optimal injection window is 120°ca long, starting with $EOI = 240^\circ \text{ca b. ITDC}$ ($p_{inj} \geq 70$ bar) and ending with $EOI = 120^\circ \text{ca b. ITDC}$ ($p_{inj} < 70$ bar). The longer the injection at lower pressure lasts the less injection timings fit in this window. But for $p_{inj} \geq 70$ bar injection duration remains almost equally (Fig. 3), thus the graphs show a comparable progression.

Figure 4 explains the slightly lower HC level at higher injection pressure. It visualizes the jet shape by absorption of infrared radiation in a pressure chamber. It is clearly recognizable that the impulse and thus the penetration raise with the injection pressure until $p_{inj} = 70$ bar. Afterwards it remains constantly. Instead, the suction of the middle beam becomes larger forcing a collapse of the surrounding beams. However, the stagnated axial penetration with higher pressure seems to convert to radial penetration causing a wider jet at all. In conjunction with deeper penetration this expansion enhances the air entrainment and therefore an improved mixture formation with higher pressure.

An explanation for the time dependent sharp minimum is given by the graphs of ignition timing, ignition phase and burn duration in Fig. 5.

To maintain the center of combustion ($mbf_{50\%}$) at $8^\circ \text{ca a. ITDC}$ an advanced ignition timing is necessary between $EOI = 240$ and $120^\circ \text{ca b. ITDC}$, causing also an increased ignition phase and burn duration. Interestingly a correlation between these graphs and those of HC emissions is noticeable. At the time of HC minimum the longest burn durations occur with the same trend of injection pressure influence. Until $p_{inj} = 70$ bar the burn duration is raising, afterwards it remains constantly. However, concerning the running smoothness such longer burn durations lead simultaneously to higher values which extend the limit of $COV_{imep} = 3\%$. Therefore to use these injection timings further measures are necessary, see Sect. 3.2.

Fig. 2 Engine results for variation of EOI and p_{inj} at $n = 2000$ rpm, $bmp = 2$ bar

Fig. 3 Injection duration t_i for different injection pressure at $n = 2000$ rpm, $b_{mep} = 2$ bar

Nevertheless, such a behavior of the ignition timing and combustion implies a strong interaction between injection and in-cylinder flow, which is getting enhanced by the growing jet impulse with higher injection pressure. The raise in ignition phase and burn duration indicate that at ignition timing the turbulence level is very low, thus the flame propagation is retarded and burn duration extended. This prolonged combustion into the expansion phase is one reason for the lower HC emissions. Another one is the mixture distribution caused by the interaction between tumble flow and jet. This process is already described in [37] and is summarized in Fig. 6.

The left-hand side of Fig. 6 presents the case for injection timing outside the optimal timeslot for low HC. The jet is not able to influence the tumble significantly resulting in a maintained charge rotation and a fuel distribution along the cylinder walls and in the top land crevice. As already explained, these areas are the main sources for HC emissions due to flame quenching, leading to higher emissions in this case. On the other hand, the right-hand

side of Fig. 6 shows a favourable mixture distribution. The injection impulse weakens the tumble flow, resulting in concentrated charge accumulation in center of the cylinder and therefore minimizing fuel mass in wall proximity and crevices.

This explanatory approach is also supported by the heat release rate for three significant EOI with $p_{inj} = 20$ bar in Fig. 7.

To enable a better comparison of the results the start of combustion is equaled in the diagram. The points outside the optimal HC window at $p_{inj} = 20$ bar (EOI = 210 and 70 °ca b. ITDC) show a faster kernel growth and higher burn rate due to increased turbulence but have a shorter burn duration and lower net heat release compared to the best point EOI = 130 °ca b. ITDC, caused by the different fuel distribution as it has been shown above.

Furthermore Fig. 8 shows the time dependent tumble flow and the cause for the EOI dependence. As already mentioned the area between EOI = 240 and 120 °ca b. ITDC contains the favourable injection timings. If the two limiting EOI's are traced back to the start of injection it becomes obvious that both have a common injection start. This allows the conclusion that all injection timings before SOI = 260 °ca b. ITDC have low influence on the tumble. Due to the raising valve lift and the suction process the tumble generation is still in progress, thus the cylinder flow reacts insensitively to disturbance from the gas jet. Together with the long duration until the ignition the maintained tumble spreads the mixture as shown in Fig. 6 (left-hand side). The picture becomes different when the injection takes place after SOI = 260 °ca b. ITDC. In

Fig. 4 Jet visualization of CH_4 in N_2 by absorption of infrared radiation in a pressure chamber for different injection pressure

Fig. 5 Ignition and burn data for variation of EOI and p_{inj} at $n = 2000$ rpm, $b_{mep} = 2$ bar

Fig. 6 Influence of tumble flow on fuel distribution in combustion chamber

Fig. 7 Heat release rate dQ_b and net heat release Q_b for selected EOI at $n = 2000$ rpm, $bmep = 2$ bar, $p_{inj} = 20$ bar

Fig. 8 Time-dependent Tumble-Number determined by CFD at $n = 2000$ rpm for motored operation ($p_{in} = 550$ mbar) and HC emissions at $n = 2000$ rpm, $bmep = 2$ bar, $p_{inj} = 15$ and 150 bar

this time period the tumble generation is almost completed so the gas jet has much more impact to weaken the flow and thus concentrates itself in the center, surrounded by an air cushion (Fig. 6, right-hand side). The remaining late injection timings after $EOI = 120$ °ca b. ITDC take place in the compression stroke when the tumble decays into turbulent kinetic energy (TKE). However, regarding mixture formation these timings are very late for sufficient homogenization, resulting in high HC emissions.

The results shown previously refer to part load operation. Besides the investigation of part load, further investigations on the emissions at higher load with scavenging are done and presented in Fig. 9. Due to insufficient mass flow engine operation with injection pressure levels lower than $p_{inj} = 70$ bar could not be ensured and therefore were omitted.

In contrast to the increased emissions at retarded start of injection at part load, a decreasing trend of the HC emissions at higher load can be observed. The earliest $SOI = 360$ °ca b. ITDC emits first the highest HC and NO_x and the lowest CO. The reason for this can be reduced to the phenomena of scavenging. Scavenging is the process of pushing fresh air through the cylinder during the valve overlap phase due to a positive pressure drop between in and outlet caused by forced induction. This reduces the exhaust gas rate in the combustion chamber and also increases the mass flow through the turbine, which leads to a better response of the turbocharging system. If fuel is injected simultaneously during scavenging, it gets with the fresh air into the outlet manifold and is detected as unburned hydrocarbon emissions. In general the local equivalence ratio in the combustion chamber is $\phi > 1.0$ at a global $\phi = 1.0$ scavenging operation. Nevertheless, in this case the mixture in cylinder becomes leaner due to the missing fuel mass and causes the higher NO_x and low CO emissions. Thereby, the higher the injection pressure is the more fuel is scavenged due to higher mass flow. This leads to the high HC peak at $p_{inj} \geq 110$ bar. When the injection is shifting progressively from the valve overlap phase to later timings the emissions decrease permanently and remain at low level after intake valve closing (IVC). However, a further small HC peak around the bottom dead center is detectable.

This abnormality can be explained with a back flow into the intake manifold, see Fig. 10. A part of the injected mass is stored in the intake manifold and subsequently scavenged in the next cycle during the valve overlap phase. Simultaneously, the storage leads to a reduction of fuel mass in the cylinder and thus a lean mixture for the combustion which causes slightly higher NO_x and lower CO again. Therefore, to reduce this HC sources it is helpful to inject the fuel after IVC.

Additionally, it is recommendable to use such late injection timings because of the already mentioned charge displacement which is demonstrated with the intake manifold pressure in Fig. 11. It becomes obvious that at early injection timings at opened intake valves more boost pressure is necessary to maintain the load. Retarding the SOI the boost pressure demand decreases more and more. Reversely, this means that a higher load can be reached at a given boost pressure if the injection takes place after the intake valve closing. This helps to enhance the low-end torque and improves the driveability.

Fig. 9 Engine results for variation of EOI and p_{inj} at $n = 2000$ rpm, bmep = 16 bar

Fig. 10 HC measurement in intake manifold with FFID at $n = 2000$ rpm, bmep = 16 bar

Fig. 11 Intake manifold pressure for various injection timings $n = 2000$ rpm, bmep = 16 bar

Furthermore, it is remarkable that even if the injection ends close to the ignition, no negative influence on HC emissions can be detected (Fig. 9). Also the acceptable engine running smoothness (COV_{imep}) indicates a stable combustion without misfires. However, aiming low injection pressure around $p_{inj} < 20$ bar for higher mileage in the future series application such timings will become irrelevant due to the critical pressure ratio caused by increased cylinder pressure during the compression stroke, which will be exceeded far before the ignition timing. The

Fig. 12 Restrictions for injection timing in low-end torque range

high pressure values used in this study are an exceptional case to ensure the flow rate and investigate basic effects.

Summarizing the mentioned aspects above, following restrictions for the injection at charged operation point with scavenging can be made (Fig. 12).

Basically, the major part of the 360 °ca long injection window is taken by the intake valve lift and thus cannot be used for the injection due to charge displacement and HC emissions at very early injection timings. The rest of the timeslot is further limited by the cylinder pressure. From the moment of exceeding the critical pressure ratio, no injection should occur due to sub-critical flow conditions. All in all, there remains a small injection window between IVC and supersonic conditions. However, beyond the low-end torque area the injection window is no longer subjected to the restrictions of the scavenging conditions due to the available boost pressure and the start of injection can be extended to the earliest SOI = 360 °ca b. ITDC.

3.2 Influence of multiple injection

Figure 13 shows the HC and PN emissions as well as the indicated specific fuel consumption with a double injection

Fig. 13 Comparison of single and double injection at $n = 2000$ rpm, $b_{mep} = 2$ bar, $p_{inj} = 20$ bar

at various splitting ratios and end of injection. The solid graph represents the single injection, whereas the symbols on the abscissa stand for the first end of injection (EOI_1) of the respective splitting ratio. The broken lines show the variation of the second end of injection (EOI_2). The bold dashed line refers to the best point of CNG-PFI.

The results demonstrate a clear advantage of the double injection reducing emissions as well as improving fuel consumption regardless of the splitting ratio. Taking a closer look at the influence of the EOI_2 and its quantity, it becomes clear that with smaller fuel amounts injected close to the ignition timing the emissions decrease whereas using a single injection a disproportionate increase is observable. This progression is already recognizable with a split ratio of 40/60 % and is continuing to 95/5 %, reaching the absolute HC minimum with 90/10 %. Thereby, the corresponding first end of injection is located around IVC (best point of single injection). Compared to best results with single injection a HC reduction of approximately 25 % and compared to the equal EOI of 60 % can be achieved.

Noticeable are the PN emissions remaining at scarcely detectable values even if the injection ends shortly before ignition. A further advantage of CNG-DI, especially with double injection, can be determined over the CNG-PFI. This combustion concept doesn't permit useful multiple injection as well as injection close to the ignition timing, thus such a strategy for HC emission reduction cannot be applied. Instead, the emissions are in the region of higher values with CNG-DI single injection due to the adverse mixture distribution as shown in Fig. 6, left-hand side.

The mentioned advantages of a double injection are attributable to two major effects. First, the split up to 50/50 % and end of injection around IVC helps to mix the fuel with air by dividing the total amount into several parts. This reduces inhomogeneous zones and thus the HC emissions from the bulk. However, later EOI 's with this splitting ratio increase the emissions. To use the potential of retarded EOI 's the amount of the second injection has to be minimized. Thereby 10 % of the total mass seems to be the optimum. Figure 14 illustrates the related mechanism for this splitting ratio.

The figure describes the so called homogeneous-split injection (HSP). The first injection is placed in the early intake stroke to provide a homogeneous lean basic mixture. As can be derived from Fig. 13 the optimal injection timing for this one is the same as for the single injection, using the already mentioned positive effects on mixture formation for this timing. The second small injection takes place close to the ignition producing both, slightly rich mixture in spark plugs' vicinity and turbulence kinetic energy which ensures a stable ignition of the lean mixture. As it can be extracted from the right-hand side of Fig. 14, there is a HC minimum at $\lambda = 1.1$. This is the reason for best results with 90/10 % causing such a lean basic mixture. In combination with a stable ignition due to the second amount the combustion with low HC emissions of the lean charge can be ensured. The crucial factor among the splitting ratio is the linked EOI of the second injection to the ignition timing. Too early EOI provides the dissipation of the induced TKE as well as the advanced mixing of the second injection aggravating the ignition. In contrast, too late EOI would lead to reverse effects like too rich and turbulent mixture around spark plug to be ignited.

An evidence for the mentioned approach delivers the data of ignition phase and burn duration in Fig. 15. For the right interpretation of the graphs the EOI of the single injection has to be used as reference and be compared with the EOI_1 of the double injection to point out the influence of the second injection.

The splitting ratios below 60/40 % show no significant difference in burn duration compared to the single injection. For this, the second amount is still too high resulting in comparable conditions to single injection. It becomes more interesting when the smaller amounts beyond 60/40 % are taken into account. As already mentioned above, it is importantly to consider the position of the EOI_1 . This one is placed in the optimum HC window around IVC. Since the most part of the fuel is injected at this time its influence on the in-cylinder flow is the same as from the single injection. Retarding the EOI_2 closer to the ignition timing the reduction in ignition time and burn duration compared to the reference becomes evident.

Fig. 14 HSP— injection strategy for double injection with splitting ratio 90/10 %

Fig. 15 Ignition phase, burn duration and COV_{imep} for single and double injection at $n = 2000$ rpm, $b_{mep} = 2$ bar, $p_{inj} = 20$ bar

Furthermore due to improved ignition conditions and accelerated combustion the running smoothness gets better and is below the limit. Referring to the issue of high COV at best HC values with single injection this injection strategy could be applied as a measure.

Section 3.1 included among the part load operation also a higher load. But due to long injection durations for the high load and thus short dwell timings a double injection is not considered in this case since it is not achieving its objectives.

4 Conclusion

The worldwide increasing energy demand and ever stricter emission legislations will require future diversification of the used energy sources in the transport sector. In this respect, especially the importance of CNG due to its CO_2 advantage compared to crude oil based fuels as well as its specific suitability for SI engines will grow. Already today, an increasing offer of CNG powered cars is detectable. Nevertheless, the customer demand remains weak despite the environmentally friendly properties. Reasons for this are an insufficient infrastructure and also a lower performance yield due to CNG port fuel injection currently used. An approach to improve the drivability is the change to CNG direct injection. But the realization of such a combustion concept has still to overcome several obstacles. Among the highly-stressed injector, the exhaust gas aftertreatment has to be mastered. Methane, main component of CNG and thus of the UHC, aggravates the

aftertreatment due to its high chemical stability causing high light-off temperature and low conversion rates. Therefore, reducing the HC emissions with engine internal measures is the most effective approach to relieve the catalytic converter. For this, a basic understanding of mixture formation with CNG is indispensable.

In this paper different injection strategies are employed to get the lowest HC raw emissions. The used test engine is a two-cylinder SI engine manufactured by Weber Motor [29]. The experiments are executed under homogeneous stoichiometric conditions with various injection timings, pressure and multiple injections. Following conclusions can be drawn:

The injection pressure has a minor impact on HC emissions. Even if the penetration and widening of the jet is increasing with pressure, scarce improvement in HC emissions is detectable. Concerning the mileage this low influence can be seen as an advantage, preferring low pressure for serial application.

The injection timing is an important parameter for HC emissions determining the available time for mixture formation as well as the interaction between jet and in-cylinder flow. Importantly in this context is the distinction between part load and higher load with scavenging.

In part load, late injection timings provide insufficient homogenization and thus incomplete combustion with high emissions. The optimal timing for minimized HC is independently from injection pressure and is located between BDC and IVC. In this timeslot an intensified interaction between jet and tumble flow takes place supporting a favourable mixture distribution for low HC emissions.

Charged operation with scavenging shows an opposite behavior. The early SOI's during opened intake valves produce highest emissions promoting fuel storage in intake manifold as well as pushing fuel through the exhaust valves. Therefore, recommended injection timings are after IVC avoiding the mentioned effects above and further reducing the charge displacement. The limit for the end of injection is determined by the cylinder pressure. To ensure supersonic flow conditions the critical pressure ratio should not be exceeded. Subjecting to these restrictions the injection window is very short. Thus, to inject the desired fuel mass a large cross-section is needed. However, beyond scavenging operation the injection window is extended up to the opened intake valves due to the available boost pressure compensating the charge displacement.

Multiple injection carries a promising potential for the reduction of HC emissions in part load. Particular advantage has a double injection with an early first injection containing 90 % of the total fuel amount followed by a second injection close to the ignition with 10 % fuel amount. This so-called homogeneous-split injection (HSP) produces a homogeneous lean basic mixture with the first injection and ensures its ignition by the second small injection close to the ignition timing. Thus, a stable combustion of the lean mixture occurs reducing the HC emissions by up to 25 % compared to the best point of the single injection.

The present study shows that to achieve lowest HC emissions it is necessary to take care in choosing the injection timing and furthermore to adapt the injection strategy to the respective engine map area. The double injection indicates that especially in part load operation different strategies can be applied to reduce the emissions. According to previous studies found in literature, an extension in knowledge with a central mounted injector could be made as well as a proof for an advantage of multiple injection could be delivered.

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