



Performance of pure OME and various HVO–OME fuel blends as alternative fuels for a diesel engine

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Abstract

Since the potential for reducing CO₂ emissions from fossil fuels is limited, suitable CO₂-neutral fuels are required for applications which cannot reasonably be electrified, and therefore still rely on internal combustion engines in the future. Potential fuel candidates for CI engines are either paraffinic diesel fuels or new fuels like POMDME (polyoxymethylene dimethyl ether, short “OME”). Besides, also blends of these two types of fuels might be of interest. While many studies have been conducted on OME blends with fossil diesel fuel, the research on HVO–OME blends has been less extensive to date.

In the current work, pure OME and HVO–OME blends are investigated in a single-cylinder research engine. The test results of the various fuel blend formulations are compared and evaluated, particularly with regard to soot-NO_x trade-off behavior. The primary objective of the study is to examine whether the major potential of blending these two fuels is already largely exploited at low OME content, or if significant additional emission reduction potential can still be found with higher content blends, but still without the need to switch to pure OME operation. Furthermore, the fuel blend which is best suited for the realization of an ultra-low emission concept under the current technical conditions should be identified. In addition, three different injector designs were tested for operation on pure OME3-5, differing both in hydraulic flow and in the number of injection holes as well as their layout. The optimum configuration is evaluated with regard to emissions, normalized heat release and indicated efficiency.

Keywords Diesel engine · CI engine · Alternative fuels · Oxygenated fuel · Synthetic fuel · HVO · Hydrogenated vegetable oil · OME · Oxymethylene ether · Emissions · Soot-free combustion

Abbreviations

| | | | |
|-----------------|---|-----------------|---|
| CA | Crank angle | IMEP | Indicated mean effective pressure |
| CI | Compression-ignited | NO _x | Nitrogen oxides |
| CO | Carbon monoxide | OME | “Oxymethylene ether” = polyoxymethylene dimethyl ether (POMDME) |
| CO ₂ | Carbon dioxide | PL | Part load point |
| DF | Diesel fuel | SOI | Start of injection |
| ECU | Electronic control unit | TDC | Top dead center |
| EGR | Exhaust gas recirculation. | WtW | Well-to-wheel |
| FAME | Fatty acid methyl ester | | |
| HC | Hydrocarbon | | |
| HVO | Hydrogenated/hydrotreated vegetable oil | | |

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1 Introduction

To reduce global warming due to greenhouse gas emissions, a shift away from fossil fuels is inevitable. Today, the transport sector is responsible for a considerable share of worldwide CO₂ emissions through the combustion of fossil fuels. In a joint effort, all energy sectors (industry, commerce housing and transport) therefore need to drastically reduce and finally completely avoid greenhouse gas emissions. Since vehicles with internal combustion engines (ICEs)

are likely to continue to account for a significant share of the global vehicle stock for considerable time to come, the mobility sector will continue to depend on fuels for a longer period of time. However, these fuels should no longer be derived from fossil sources, but from renewable energy or biomass. Despite the very high level of ICE exhaust gas quality which has already been reached to date, the requirements in this regard are still increasing as pollutant emission standards are continuing to be tightened worldwide. As the potential to further reduce emissions from the combustion of fossil fuels is limited, the use of such alternative fuels might also be a promising solution in this regard.

Potential alternative fuels for use in CI engines include the so-called paraffinic diesel fuels of biogenic origin (such as hydrogenated vegetable oil HVO) or from fully synthetic production (so-called “e-fuels”, e.g., Fischer–Tropsch fuels) as well as other synthetic fuels with properties deviating more strongly from fossil diesel, such as oxygenates like POMDME (polyoxymethylene dimethyl ether, abbreviated: OME), and also blends of different biogenic and/or synthetic fuels.

While numerous investigations have already been carried out on OME blends with fossil diesel fuel, considerably less is known on the behavior of HVO–OME blends. In a substantial number of studies, the blends investigated contained additional blend components, with the quantitative share of these additional components exceeding the OME share in several cases.

As a part of a publicly funded renewable fuels research project, extensive tests have been carried out on a single-cylinder research engine using OME and various HVO/OME blends. These investigations aimed at the definition of the optimum blend for future vehicle applications as well as the specific combustion system development for pure OME and HVO/OME blends.

Additionally, the results of a small-scale test using an HVO/OME blend with an elevated OME content (30%_{vol}) and investigations with pure OME are presented and compared to previous results for HVO/OME blends with a lower OME content and for two reference fuels (according to EN590 and EN15940). In the discussion, a special focus will be put on the soot–NO_x trade-off. Finally, some results of a parameter study aiming at optimizing the injector design for pure OME operation will also be discussed.

2 Theoretical basics

Paraffinic diesel fuels as substitutes to diesel are defined according to the EN1594 standard in Europe (in the US and Japan according to the same standards as for conventional diesel fuel, i.e., ASTM D975 and JIS K 2204). These types of fuels consist almost entirely of alkanes and are therefore

free from aromatics. This results in a lower tendency to form soot compared to conventional diesel fuel (abbreviated by “DF” in the following). The potential of such fuels for soot reduction, but also for the reduction of CO and HC emissions, has already been demonstrated under a wide range of operating conditions in many studies (e.g., [1–7]). However, some studies also showed that under certain operating conditions, soot emissions may increase compared to DF when operating the engine with the paraffinic fuel HVO [2, 6, 8]. NO_x emissions for HVO and DF applications are generally reported to be at comparable levels [1–6]. The group of paraffinic diesel fuels can be subdivided according to the method of fuel production. HVO is produced by the hydrogenation of vegetable oil [9]. Since animal fats or other raw materials may also be used in the production process, the term HDRD (Hydrogenation-Derived Renewable Diesel [10]) might be more appropriate as a generic term. Another possibility for obtaining a paraffinic diesel fuel is the so-called Fischer–Tropsch process [9]. In this case, the fuel is gained purely from a synthesis process instead of modifying oil from biogenic sources. Both variants of paraffinic diesel fuels offer the possibility of significantly reducing Well-to-Wheel (WtW) CO₂ emissions when produced using renewable energy (thus considering the upstream processes of fuel production). Ultimately, there is even the potential to reach CO₂-neutral operation. In the current study, HVO was used instead of a fully synthetic paraffinic diesel fuel due to its better availability.

Other candidates for alternative diesel fuels are oxygenated fuels such as OME. A distinctive feature of these fuels is their molecular structure [CH₃O(-CH₂O-)_nCH₃] [8]. In these types of molecules, there are no direct carbon-to-carbon bonds due to oxygen atoms bonded in-between [7]. Except for negligible soot formation from the combustion of lubrication oil, this results in soot-free combustion when OME is used, provided that there are no soot-forming blend components [7, 8, 11–13, 15, 16]. This opens up the possibility of resolving the so-called soot–NO_x trade-off (thus the mutual dependence of soot and NO_x emissions usually observed in diesel fuel combustion), which also allows to realize ultra-low NO_x levels [8, 11–13, 15, 16]. Even in a blend with DF or HVO, a significant reduction of soot emissions can be observed at many operating conditions [14, 17–19]. However, at elevated load points, studies of DF–OME blends indicate at least a slight increase in NO_x emissions [17–19]. Comparative studies of different OMEs with varying chain length (OME1, OME2, OME3–4 and more) show only small differences in CO or HC emissions compared to HVO [13, 15]. According to investigations using DF–OME blends, such fuels provide at least a small potential for reducing CO and HC emissions compared to DF operation [17–19]. The physical properties of OME are strongly determined by the number of oxymethylene groups

Table 1 Characteristic values of OME fuels with differing chain length [13, 15]

| | Unit | OME3 | OME4 | OME5 |
|-------------------------|--------------------|------|------|------|
| Heating value | MJ/kg | 19.4 | 18.7 | 18.1 |
| Density (@15 °C) | Kg/m ³ | 1030 | 1070 | 1110 |
| Kin. viscosity (@40 °C) | mm ² /s | 1.08 | 1.72 | 2.63 |
| Cetane number | – | 67 | 76 | 90 |
| Boiling point | °C | 156 | 202 | 242 |
| HFRR (@60 °C) | | 534 | 465 | 437 |
| O2 concentration | % (m/m) | 47.0 | 48.1 | 48.9 |

[(-CH₂O-)n] present in the chain and may even differ considerably in some cases (cf. Table 1) [13–15].

OMEs are synthetic fuels which can also be classified as an “e-fuel” (or PtL = power-to-liquid). Similar to paraffinic diesel fuels, there is a high potential for realizing ultra-low WtW CO₂ emissions up to complete CO₂ neutrality. OME is generally produced via a methanol route. The production of methanol requires hydrogen and CO₂. For a CO₂-neutral fuel, the hydrogen used needs to be produced, e.g., by water electrolysis using renewable electricity, and the required CO₂ must be extracted from the atmosphere. Since the production of CO₂ from ambient air is not yet economically viable considering the currently quite low price of CO₂, the required carbon dioxide might still need to be extracted from (inevitable) industrial sources for a certain period of time [15].

Since paraffinic diesel fuels can also be produced by hydrogenating fats or oils, it will initially be easier to provide such a fuel in larger quantities than for a fuel which can only be produced in a pure synthesis process (such as OME). Therefore, a realistic drop-in scenario for alternative fuels might be to first replace conventional fossil diesel fuel with a paraffinic diesel fuel like HVO. In a subsequent step, OME could then be used as a blending ingredient to provide further pollutant reduction benefits. As the availability of OME increases, a full switch to this type of fuel could be realized in the medium to long term.

3 Engine test bench setup

The investigations described below focused on determining the influence of OME and HVO–OME mixtures on the combustion process of a diesel engine, with a particular focus on emissions. The investigations were carried out using a single-cylinder research engine derived from a passenger car diesel engine. The most relevant technical data of the engine can be found in Table 2.

The fuel system components were adapted to the use of OME in terms of the choice of materials used. For the

Table 2 Technical details of single-cylinder engine

| Engine specification | |
|----------------------------|-----------------------------|
| Bore diameter | 83 mm |
| Stroke | 91.4 mm |
| Displacement | 494.5 cm ³ |
| No. of valves per cylinder | 4 |
| Injection system | Common rail (max. 2500 bar) |

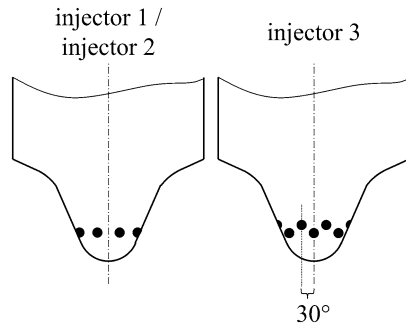
investigations of the different fuel blends which will be discussed first (Sect. 4), the fuel injection was realized by an 8-hole solenoid injector (*injector 1*; cf. Table 3) which—apart from the sealing materials used—corresponds to a conventional diesel injector, particularly with respect to the hydraulic flow rate. The flow rate of this injector is designed for the calorific value of conventional diesel fuel, which is significantly higher than that of OME. In an additional series of experiments carried out exclusively with pure OME3–5 (Sect. 5), two other injector designs were used in addition to this injector, both with an adapted hydraulic flow rate to account for the lower calorific value of OME (*injector 2* and *injector 3*; cf. Table 3).

In the case of *injector 2*, the increase in hydraulic flow was achieved by increasing the opening cross-section of the injection holes. Apart from this adjustment, the design of the injector (e.g., the spray targeting) corresponds to *injector 1*. In the case of *injector 3*, the elevated hydraulic flow is a result of a larger number of injection holes and not an increase in the opening cross-section of the individual injection holes. Since an arrangement of 12 injection holes on a single hole circle would have caused an unacceptable reduction of the material thickness, the injection holes were arranged in two separate hole series offset by 30°, each with 6 injection holes (see the schematic representation of the injection hole arrangement in Fig. 1).

All operating fluids of the test bench (i.e., charge air, cooling water, engine oil, and fuel) are conditioned regarding temperature and pressure. An IAV FI2RE prototype ECU with integrated pressure indication is used to control the test engine. The ECU can be used to set all the necessary operating parameters (e.g., injection parameters including rail pressure, EGR = exhaust gas recirculation rate, swirl flap position, boost pressure, and exhaust backpressure). Supercharging is realized by an external compressor unit, providing intake pressures of up to 4.5 bar_{abs}. The engine control unit and the associated engine control software also allow a fast online calculation of the heat release based on the measured indicated pressures. However, the calculation assumes a constant polytropic

Table 3 Injector configurations

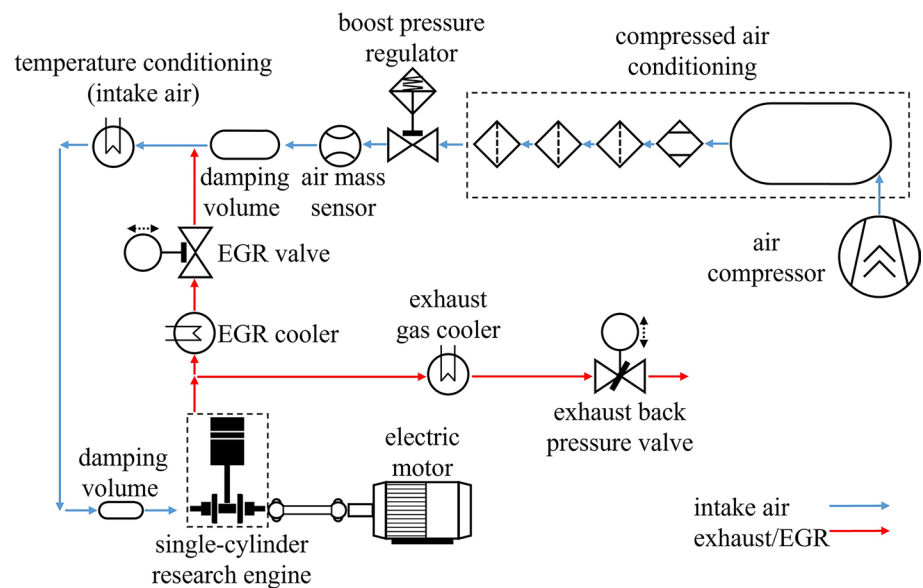
| | Injector 1 | Injector 2 | Injector 3 |
|----------------|----------------------|------------------|----------------------|
| Nozzle | 8 Holes | 8 Holes | 12 Holes (2×6) |
| Hydraulic flow | Diesel configuration | Increased by 50% | Increased by 50% |
| Hole diameter | Diesel configuration | Increased | Diesel configuration |

**Fig. 1** Schematic illustration of the injection hole layout for injector 3 compared with injector 1 and injector 2

as the basis for comparison. The following blends were investigated:

- blend A: HVO/OME3-5 (90/10%_{vol})
- blend B: HVO/OME3-5/2-ethyl-1-hexanol (80/15/5%_{vol})
- blend C: HVO/OME3-5 (70/30%_{vol}) (only small-scale test).

Some of the results of the tests with blend A and blend B have already been reported in a previous publication [20]. Since then, the investigations have been extended to include a small-scale test with blend C and tests with pure OME as well as a variation of the injector design for pure OME

Fig. 2 Test bench setup

exponent. Soot concentrations are determined by an AVL Micro Soot Sensor system, and an AVL FTIR analyzer is used to measure the gaseous exhaust components.

The setup of the test bench is displayed in Fig. 2. For data acquisition and processing, several software systems were used (NI LabVIEW, Matlab, AVL CONCERTO 5).

Within the scope of the related project, various fuel blends of HVO and OME3-5 as well as pure OME3-5 were investigated, with conventional diesel fuel (service station grade with 7 vol% FAME, according to EN590) and HVO as paraffinic diesel fuel (according to EN15940) serving

operation. In this work, some of these new findings will be discussed and compared to the previous results.

Since it has been found that HVO–OME blends exhibit a blending gap at low temperatures, suitable blending promoters are required beyond a certain OME content for practical use. In blend B, therefore, 2-ethyl-1-hexanol is used as a mixing promoter, since otherwise phase separation effects may occur at low temperatures with OME fractions of this magnitude. To investigate the potential of higher OME contents to reduce soot emissions, measurements were also carried out with such a blend (blend C) in a dedicated

Table 4 Characteristics of the test fuels

| | Unit | Diesel | HVO | OME3-5 |
|------------------------------|--------------------|---------|---------|---------|
| Heating value | MJ/kg | 41.4 | 43.4 | 19.8 |
| Density (@15 °C) | kg/m ³ | 833.0 | 779.9 | 1055.5 |
| Cetane number | – | 54.2 | 78.5 | 80.5 |
| Kin. viscosity (@40 °C) | mm ² /s | 2.724 | 2.862 | 1.188 |
| O ₂ concentration | % (m/m) | <0.5 | <0.5 | 45.3 |
| C/H ratio | – | 6.43 | 5.54 | 5.07 |
| | Unit | Blend A | Blend B | Blend C |
| Heating value | MJ/kg | 40.3 | 38.7 | 34.7 |
| Density (@15 °C) | kg/m ³ | 804.7 | 820.0 | 858.5 |
| Cetane number | – | 74.6 | 78.1 | 77.5 |
| Kin. viscosity (@40 °C) | mm ² /s | 2.412 | 2.155 | 1.855 |
| O ₂ concentration | % (m/m) | 7.2 | 9.4 | 16.5 |
| C/H ratio | – | 5.46 | 5.51 | 5.38 |

experiment. As suitable blending agents are not yet defined for this blend ratio, the blend had to be used without blending agents. Since phase separation may occur already at temperatures in the range of 15–20 °C with the OME content of blend C, extensive adjustments to the fuel system (including a heated fuel tank with a stirrer) were necessary for the use of this fuel to exclude segregation at least for the duration of a small-scale test (including an EGR variation at a part load operating point). The most important characteristic values for the fuels used can be found in Table 4.

To ensure reproducible measurement results, the following procedure was applied. On each measuring day, the measuring instruments for determining the gaseous constituents and the soot concentration were calibrated before the start of the measuring operation. To verify the reproducibility of previous measurements, a dedicated reference measuring point was set at the beginning of each measurement campaign and checked against previous results. After setting the operating point to be investigated, the engine was operated until stable conditions were established. The actual measurement was then carried out over a period of 60 s. The results shown below were determined from the average of these 60 s. Table 5 shows the measurement tolerances of the test bench equipment used.

4 Potential for soot and NO_x emission reduction of the different fuel specifications

A primary objective of the current experimental investigations is to characterize the different fuel blends as well as pure OME with respect to their emission reduction potential. The investigations cover several characteristic operating

points of the corresponding passenger car diesel engine and include variations of several operating parameters (injection parameters, swirl level, and EGR rate).

The starting point of the fuel study was a conventional diesel parameter set with multiple pre-injections (max. 3 events) and an optional single post-injection. Based on the original parameter set for diesel fuel, the gravimetric injection quantities of the individual pre-injections and the post-injection were corrected according to the calorific values of the fuels used for the corresponding study.

The duration of the main injection is specified by the control unit using a load controller (based on IMEP—indicated mean effective pressure). Another controller sets the desired center of combustion via the SOI (start of injection) of the main injection. The pre- and post-injections are coupled to the main injection via constant offset values, so that a change in the SOI of the main injection also leads to a shift of the respective timing of a pre- or post-injection by the same value. Except for this correction of injection quantity based on the calorific value, no further fuel-specific adjustment of the combustion process was carried out at this step. Starting

Table 5 Measurement accuracies of the test equipment

| | Tolerance (%) | Resolution |
|---------------------|---------------|------------------------|
| Engine speed | 0.5 | 1 rpm |
| Engine load | 0.5 | 0.01 bar |
| Crank angle | 0.1 | 0.01°CA |
| Intake pressure | 0.1 | 1 kPa |
| Intake temperature | 1 | 0.1 K |
| Fuel flow meter | 1 | 0.01 kg/h |
| Mass air flow meter | 1 | 0.1 kg/h |
| Gaseous analyzer | 1.5 | 1 ppm |
| Soot mass analyzer | 3 | 0.01 mg/m ³ |

Table 6 Operating points

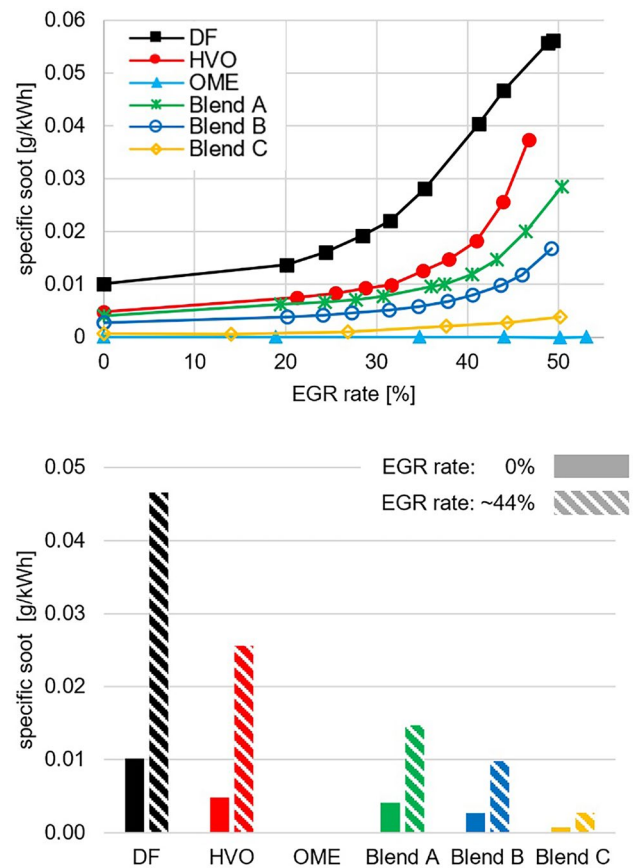
| | Engine speed | IMEP | Rail pressure | Number of pre-injections | Number of post-injections |
|-----|--------------|-------|---------------|--------------------------|---------------------------|
| | [rpm] | [bar] | [bar] | [–] | [–] |
| PL1 | 1250 | 3.70 | 755 | 3 | 0 |
| PL2 | 1750 | 15.6 | 1665 | 1 | 1 |

from this initial data set, the optimum operating strategy was defined by variations of the operating parameters. In the following, exemplary results of an EGR variation focusing on the soot-NO_x trade-off are discussed for two part-load (PL) operating points. The characteristic parameters of these operating points are listed in Table 6.

In the following, the results for the low-load point PL1 will be discussed in detail first. As obvious from Table 6, this point is operated with three pilot injections and a rail pressure of 755 bar. The amount of pre-injected fuel was adapted to the calorific value of the individual fuel. In total, approx. 50% of the total fuel quantity is injected via the three pre-injections at this operating point. The center of combustion was adjusted to the same value for all fuel variants and for all EGR rates. It should be noted that the small-scale test with blend C was also carried out at this operating point, and the corresponding results are therefore included in the evaluation. The aim of this small-scale test was to determine whether a further increase in the OME content would lead to a substantial additional reduction in soot emissions, to assess whether the effort required to identify a suitable blending agent for blends with higher OME contents (to avoid segregation) is justified.

The upper part of Fig. 3 shows the soot emission curves as a function of the EGR rate. The lower part of the figure contains a point-by-point comparison of the operating cases without EGR and with an EGR rate of approx. 44%. The conventional diesel fuel (DF) shows the characteristic and well-known significant rise in soot emissions when the EGR rate is increased. The soot formation is likely to be additionally favored by the poorer autoignition capability compared to the other fuels (DF exhibiting the lowest cetane number of all fuels tested, cf. Table 4) at higher EGR rates. Qualitatively, HVO shows a similar tendency. However, due to the lack of aromatics, HVO already results in significantly lower soot values (approx. 44% less at an EGR rate of 44% compared to DF, see lower part of Fig. 3), so that the soot limit shifts toward higher EGR rates. The potential to reduce soot emissions with HVO compared to DK has already been reported in several studies [2–4].

In contrast, pure OME exhibits a completely different behavior. As has already been shown in various publications [7, 8, 11–13, 15–16], virtually, no soot emissions occur

**Fig. 3** Specific soot emissions depending on the EGR rate (PL1)

when OME is used, provided that no soot-forming blend components are present. Accordingly, operation with pure OME results in extremely low-soot emissions also in this case. Throughout the entire parameter range investigated, soot emissions are either non-existent or below the detection limit of the measurement technology, even at the highest EGR rates applied during the investigation. As explained before, this can primarily be attributed to the absence of carbon-to-carbon bonds and to the oxygen bound in the molecule, which significantly reduces or even completely prevents the formation of soot precursors during combustion.

A comparison of the soot production of the three HVO–OME blends indicates that the already quite moderate soot emission of HVO is further strongly reduced by the addition of OME. Thus, the combination of the lack of aromatics in the HVO fraction with the quasi-zero soot formation of the OME fraction results in ultra-low soot, especially at higher EGR rates. The soot-reducing effect of adding OME to conventional diesel fuel [14, 18, 19] or HVO [21] has already been documented in other studies. It is also possible that the potential for soot formation of the blending agent 2-ethyl-1-hexanol (which is also an oxygenated fuel) is lower than that of HVO, which might additionally contribute

to the lower soot formation of blend B. Another factor resulting in reduced emissions could be increased in-cylinder soot oxidation. The soot emissions in the raw exhaust gas of a diesel engine generally only represent a small fraction of the total amount of soot formed during combustion. Most of the soot formed is already oxidized in oxygen-rich zones during combustion. Compared to pure HVO operation, the addition of OME should lead to improved mixture formation, which should also have a positive effect on the post-oxidation of soot. The EGR compatibility thus increases significantly in these cases.

The direct comparison of the blends underlines that compared to HVO, the soot emissions of an HVO–OME blend can be successively reduced by increasing the OME content. It should be kept in mind that blend B contains a non-negligible proportion of 5% by volume of 2-ethyl-1-hexanol as a blending agent, which might also influence the soot emissions. However, since blend C (with the highest OME content) does not contain an additional blending agent, but still clearly shows the same trend, the OME content is likely to dominate in this regard.

Blend C produces extremely low-soot emissions over the entire range of EGR rates, which are even hardly distinguishable from the results of pure OME at low-to-medium EGR rates. Only at higher EGR rates, a noticeable increase as a result of the soot-forming properties of the HVO fraction can be recorded. However, the resulting soot level is still well below the values of the other fuels (except OME) or blends due to the strongly reduced or absent soot-forming tendency of the OME and probably further improved post-oxidation of the soot. Thus, at an EGR rate of 44%, the soot emissions are merely at the level of the other two blends in operation without any EGR. Starting from DF, the soot emissions are reduced by approx. 68% for blend A, 78% for blend B and 94% for blend C with increasing OME content at this EGR rate.

The upper part of Fig. 4 shows the specific NO_x emissions as a function of the EGR rate. In principle, it can be assumed that the higher the EGR rate, the lower the peak temperatures will be during combustion due to the increase in heat capacity as well as the reduction in oxygen content of the cylinder charge and the resulting reduced combustion speed. This mechanism leads to a significant reduction in NO_x emissions for all fuel variants. DF, HVO, and blend B are at a similar level, with HVO and blend B producing slightly higher nitrogen oxide values at the same EGR rate. The observation that, at low load, HVO operation results in a comparable or at most slightly elevated NO_x level compared to DF has already been reported in various studies, e.g., [2, 4].

The NO_x values are most striking in operation with OME. In this case, except for the highest EGR rates studied, significantly lower NO_x values are achieved in comparison to

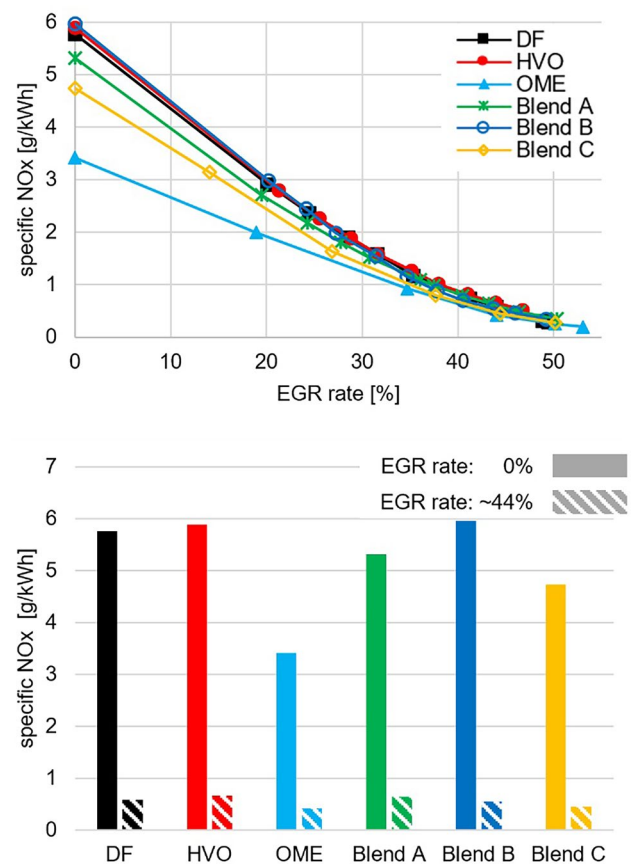


Fig. 4 Indicated specific NO_x emissions of the investigated fuels as a function of EGR rate (PL1)

the other fuels investigated. It could be assumed that this is, among other factors, due to the design of the injector used, whose hydraulic flow is designed for conventional diesel fuel. Because of the significantly lower calorific value of OME, the required fuel quantity increases, so that with the same injector design and rail pressure, the injection rate related to the fuel-specific calorific value is reduced with respect to the other fuels. Due to the resulting slower combustion, reduced combustion peak temperatures will occur, resulting in lower NO_x emissions. This is indeed an important influencing factor, especially at higher loads and in combination with elevated engine speeds. However, as will be demonstrated in the following section (OME operation with injector variation), the low NO_x level for PL1 compared to HVO and DF can only partly be attributed to the injector layout. Since PL1 is a low-load operating point, only a relatively small amount of fuel is injected, which is distributed over a total of four injections. In PL1, only about 50% of the fuel quantity is injected via the main injection, while the rest is divided among the pilot injections. In relation to the relatively low engine speed in PL1, the required injection time increases only moderately with OME. At this low-load

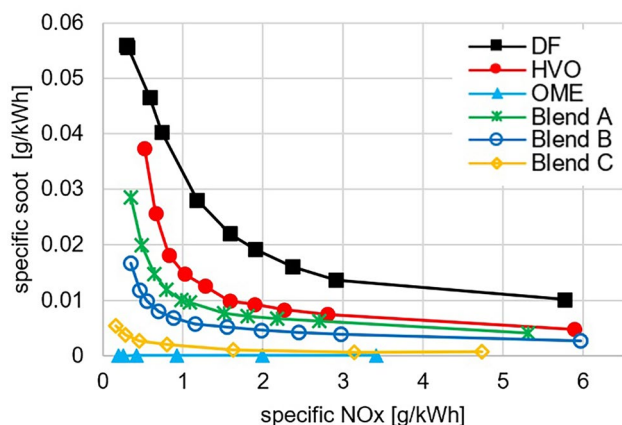


Fig. 5 Indicated specific soot emissions as a function of indicated specific NO_x emissions of the investigated fuels (PL1)

operating point, the disadvantage of the lower injection rate is at least partly compensated for by the good mixture-forming properties of OME. Together with combustion temperature, the oxygen concentration is one of the most important promoters of NO_x formation. Since there is a high air/fuel ratio at this low-load point, the additional oxygen content contained in the OME fuel molecules does not seem to have any NO_x-increasing effect. This leaves the combustion temperature as the critical parameter at this point [19]. Therefore, the low nitrogen oxide emissions in PL1 might rather be explained by a significantly improved homogenization of the fuel–air mixture with OME compared to the other fuels. This should lead to a further leaning-out of the already oxygen-rich combustion zones, resulting in a decrease of the local combustion temperatures and thus a reduction of NO_x emissions. However, further investigations (especially CFD calculations) are required to confirm this hypothesis.

At low EGR rates, blend A results in up to 8% lower NO_x emissions than DF, while for blend B, these are roughly on a par with DF and HVO. However, the additional blending component in the form of 2-ethyl-1-hexanol included in blend B could again also have an influence on emissions. In a publication covering experiments on commercial vehicle engines with HVO–OME blends containing 10% 2-ethyl-hexanol and 7% FAME, an increase in NO_x emissions was reported [21]. Compared to DF, Blend C (without a third component) again produces approx. 18% lower nitrogen oxide emissions in operation with moderate EGR rates. This may be due to better homogenization of the fuel–air mixture by the OME content, similar to the effect observed with pure OME. At a high EGR rate (cf. lower part of Fig. 4; EGR rate: ~44%), the emissions of all fuels are at a comparably low level, with only slightly better values for pure OME and blend C.

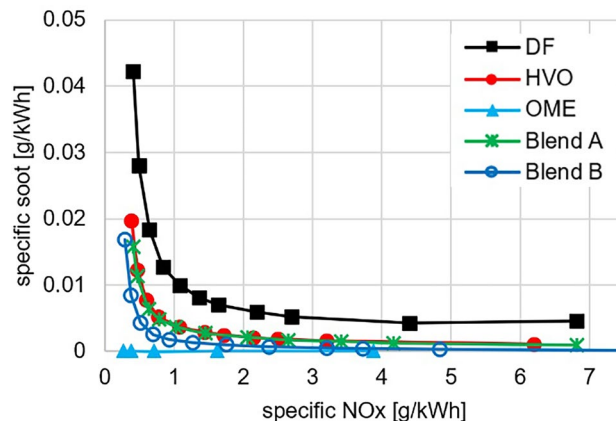


Fig. 6 Indicated specific soot emissions as a function of indicated specific NO_x emissions (PL2)

To illustrate the soot–NO_x trade-off, the soot emissions are plotted as a function of the NO_x emissions in Fig. 5. This once again clearly illustrates the advantage of low-soot fuels over conventional diesel fuel. At the same nitrogen oxide level, a significant reduction in soot emissions can already be achieved with HVO, resulting in about 50% reduced soot at NO_x raw emission levels down to 1 g/kWh₁. Only below this level, the soot values start to rise progressively to almost DF level. The same effect is still relevant when adding small amounts of OME, and particularly noticeable at extremely low NO_x values due to the higher EGR compatibility of the fuel. With blend C, soot emissions remain at an extremely low level even for very high EGR rates, so that the soot–NO_x trade-off is already almost negligible, at least at this low-load operating point. In operation with pure OME, no soot at all is detectable in the entire range investigated, resulting in a complete elimination of the soot–NO_x trade-off. The results of blend C indicate a high soot reduction potential which already comes quite close to operation with pure OME. Further investigations at other operating points are required to confirm this behavior. For this purpose, it is, however, necessary to identify a suitable mixing agent first, to ensure stable conditions for real engine operation. It is also necessary that a potential mixing agent does not negatively influence or even eliminate the soot reduction potential.

Similar investigations as described above have been carried out for an operating point at elevated part load (PL2; engine speed: 1750 rpm; IMEP: 15.6 bar). As obvious from Table 6, this operating point is operated with one pre-injection and one post-injection at a rail pressure of 1665 bar. The center of combustion is once again adjusted to the same value for all fuel variants and EGR rates via the SOI controller of the main injection.

For PL2, soot emissions are plotted against NO_x emissions in Fig. 6 for all variants which have been tested, again

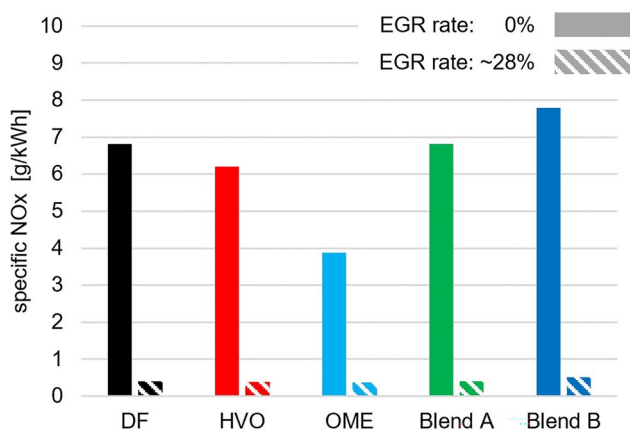


Fig. 7 Indicated specific NO_x emissions of the fuels in operation with and without EGR (PL2)

using indicated specific values. As expected, DF again exhibits the highest soot emissions, associated with a clearly pronounced soot-NO_x trade-off. Due to the lack of aromatics in the fuel, HVO produces lower soot emissions than DF also at this operating point. However, the effect is less pronounced than in PL1. This might partly be attributed to the significantly higher combustion temperatures at such higher load operating points, favoring the post-oxidation of soot particles during combustion already in DF operation. The post-injection applied for PL2 also has a soot-reducing effect. In addition, it can be assumed that at this operating point, the reduced autoignition capability of diesel is less relevant than at PL1 due to the higher end-of-compression pressure level (as a result of increased boost pressure), and also higher combustion temperature. When considering the two blends A and B, it can be seen that at this operating point, blend A results in only very slight reductions in soot emission compared to HVO. In a recent study on HVO-OME blends [21]—however with ethyl hexanol and FAME as additional blend components—similar soot emission levels were noted at some higher load operating points. Also there, the soot level of the investigated blend with the highest HVO content was almost at the level of pure HVO operation at certain operating points, while the emissions could be reduced further with increasing the OME content.

With Blend B, soot emissions can be reduced at least a bit more significantly. This may be attributed to the further increase in the OME content—burning almost soot-free—and to the 2-ethyl-1-hexanol content, which will also have with a lower tendency to form soot than HVO. As already observed for PL1, operation on pure OME produces no detectable soot emissions due to properties of the fuel as described before; if soot should be present at all, the values are below the detection limit of the measurement technology. Consequently, a soot-NO_x trade-off behavior cannot be detected at this operating point, either.

Fig. 7 shows the nitrogen oxide emissions for operation without EGR compared to the results for operation with an EGR rate of approx. 28% at PL2. While the NO_x emissions are at almost the same level for all fuels at high EGR rates, there are significant differences in some cases at lower EGR rates, as already observed for PL1. In relation to DF, the NO_x emissions for HVO are about 9% lower. A similar slight reduction in the NO_x value in HVO operation at medium load has also been observed in the other studies cited before [2, 4].

The most striking feature at this operating point is again the NO_x level in operation with pure OME, which is significantly lower than for the other fuels. Compared to DF, the reduction amounts to up to 43%. At this operating point, this can very probably be attributed to the significantly lower calorific value of the OME combined with the limited hydraulic flow of the injector. In PL2, a significantly higher quantity of fuel needs to be injected compared to PL1. Especially in operation with pure OME, this increases the required injection duration substantially, and thus also the relative proportion of mixture-controlled combustion. The slower combustion results in lower peak temperatures and thus a reduction in nitrogen oxide emissions. As already explained above, the good mixing properties of OME might additionally favor a lower nitrogen oxide level through better homogenization of the air-fuel mixture.

When considering the two blends, it can be seen that in PL2 for the case without EGR, the NO_x emissions in operation with blend A correspond to the level of DF operation and are thus higher than those of HVO. As in PL1, NO_x emissions rise further in operation with blend B compared to blend A. The increase amounts to approx. 14% compared to blend A and DF. Looking at the NO_x levels of HVO and pure OME, lower NO_x emissions would have been expected for the blends, also in view of the results in PL1. This could be due to the relatively faster combustion of the blend fuels, which—in contrary to operation with pure OME—overcompensates the longer injection duration required as a result of the lower calorific value. Because the main factors for NO_x formation are high temperatures and a sufficiently high oxygen content, the increase in NO_x emissions in blend operation in comparison to DF and HVO is probably due to the high oxygen content of the OME component and its high volatility. Both factors increase oxygen availability during combustion. Due to the increased fuel injection quantity compared to PL1, oxygen-deficient zones are expected to occur more intensively in PL2 in operation with DF or HVO as a result of insufficient mixture formation, which is counteracted by the two before-mentioned factors of blend operation [19].

Because of the higher heat capacity of the cylinder charge and the reduction of the oxygen content in the combustion chamber by the recirculated exhaust gas, NO_x formation is

inhibited during operation with EGR, so that nitrogen oxides are significantly reduced with increasing EGR rate. At an EGR rate of about 28%, the NO_x level is almost the same for all fuels (cf. Figure 7).

5 Influence of injector layout on OME operation

In addition to the studies of HVO–OME blends discussed in the previous section, an additional focus of the project is on the requirements and the potential of pure OME as an engine fuel. These investigations also include an adjustment of the hardware, in particular with respect to the injector layout. As a starting point, the injector already applied in the previous investigations was used (which is identical to the standard diesel injector layout, except for modifications with respect to material choice which are required for OME compatibility). This injector will be designated as “*injector 1*” in the following. In addition, two injector variants designed specifically for OME were investigated: an 8-hole injector with increased hydraulic flow (“*injector 2*”) and a 12-hole design with two offset series of 6 holes, also with increased hydraulic flow (“*injector 3*”, cf. Table 3). The investigation of *injector 1* serves primarily as a reference here; due to its layout for the higher calorific value of conventional diesel fuel, this injector would imply severe restrictions at high-load operation, and the map in which the engine could be operated might probably be significantly reduced.

For the pilot injections, the injection strategy applied for diesel operation was carried over in this test series. Also, the pre-injection control duration was kept constant for all injectors. With this same application, it could generally be expected that with *injector 2* or *injector 3*, the amount of fuel injected via the pre-injections should increase significantly due to their higher hydraulic flow rate. However, in the context of optical injection tests,¹ it was found that these two injectors also exhibit a significantly faster hydraulic closing behavior than *injector 1*, especially when operating with pure OME. For the same hydraulic opening duration, the required electric energizing duration thus increases compared to *injector 1*. As a result of this behavior, the two effects of increased hydraulic flow on the one hand and accelerated hydraulic closing behavior on the other almost compensate each other for short electric energizing durations. However, it needs to be kept in mind that the pre-injection quantities might differ slightly between the injectors.

¹ Beutler et al., “Numerical Modeling of Diesel and Polyoxymethylene Dimethyl Ether Spray in a High Pressure Chamber Using the Fischer Primary Breakup Model, submitted for Automotive and Engine Technology” (currently in review).”

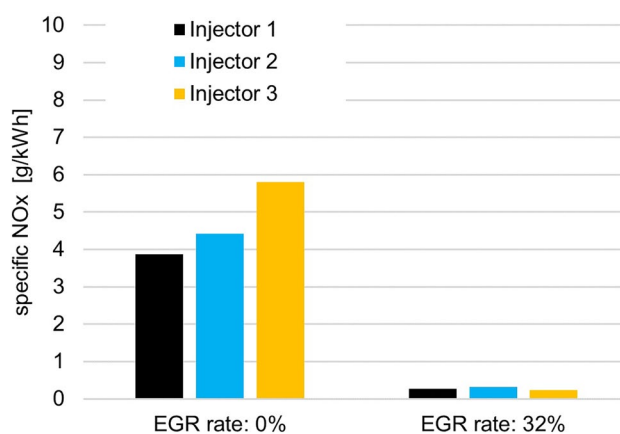


Fig. 8 Indicated specific NO_x emissions in operation with and without EGR (PL2)

In the following, some results of these investigations in engine operation with pure OME are discussed, using the example of PL2 as a representative operating point for operation at elevated load. This is followed by a short discussion of the corresponding results for the low-load point PL1.

As a part of the standard data set, the PL2 operating point applies an early post-injection in diesel operation, which is intended to reduce soot emissions. Since almost no soot emissions occur in pure OME operation, this post-injection was not used in the test series discussed here. The operation in PL2 was examined both without EGR and with an EGR rate of 32%. The rail pressure was set to 1665 bar, and the center of combustion was adjusted to the same value for all variations via the SOI of the main injection.

Figure 8 shows the specific NO_x emissions. In operation without EGR, *injector 1* produces the lowest emissions; when using the 8 hole injector with increased flow (*injector 2*), NO_x emissions increase slightly, while with *injector 3* (12-hole with increased flow), emissions rise even more sharply. Especially, the low NO_x values observed for *injector 1* can be attributed directly to the injector design. Due to the lower hydraulic flow, the injection duration for this injector is significantly longer than for the other two injectors.

Figure 9 shows the normalized heat release curves, for which the results of all injectors were normalized to the maximum value of the curve of *injector 1*. The result of the reduced injection rate of *injector 1* compared to the other two injectors is clearly visible. Due to the lower injection rate, injection starts significantly earlier than for the other two injectors, while the center of combustion remains the same. Furthermore, injection also ends significantly later, resulting in the relatively flat heat release curve obvious from in Fig. 9 for *injector 1*. This results in slower combustion and thus lower peak temperatures. Since nitrogen oxide formation requires high local combustion temperatures in

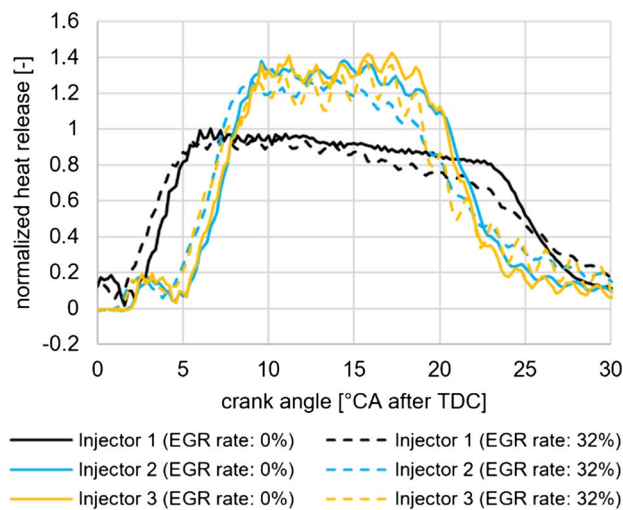


Fig. 9 Normalized heat release for operation with and without EGR (PL2)

addition to a local excess of oxygen, a reduction in peak temperatures accordingly leads to a reduction in nitrogen oxide emissions.

Due to the increased hydraulic flow rate, the injection rates for *injector 2* and *injector 3* are higher. Accordingly, the combustion rate is not limited that much by the injection rate as in the case of *injector 1*. Due to the faster energy conversion, an increase in local peak temperatures can be expected, which in turn favors NO_x formation. This is reflected by the increase in the NO_x level with respect to *injector 1*. However, operation with *injector 2* obviously results in lower nitrogen oxide emissions than observed for *injector 3*, in spite of the hydraulic flow being identical for both injectors. The increased hydraulic flow of *injector 3* is achieved by a higher number of injection holes, while for *injector 2*, it is due to an increase in the cross-section of the injection holes. As a consequence, *injector 3* is expected to improve mixture formation particularly at higher injection pressures, where it can be assumed that the vaporization of the fuel is improved due to the reduced droplet size combined with a larger number of droplets. This might result in a larger NO formation zone compared to *injector 2* and is probably the reason for the higher NO_x values of *injector 3*.

As expected, operation with EGR results in a significant reduction in nitrogen oxide emissions for all three injectors due to the reduction in peak combustion temperature as a result of a higher heat capacity and the reduced oxygen content of the cylinder charge. Emissions are roughly at the same level for all of the injector variants.

The indicated specific CO emissions detected in this investigation are shown in Fig. 10. As expected, the CO values are low in operation without EGR, with *injector 3* showing a slight tendency toward lower emissions compared

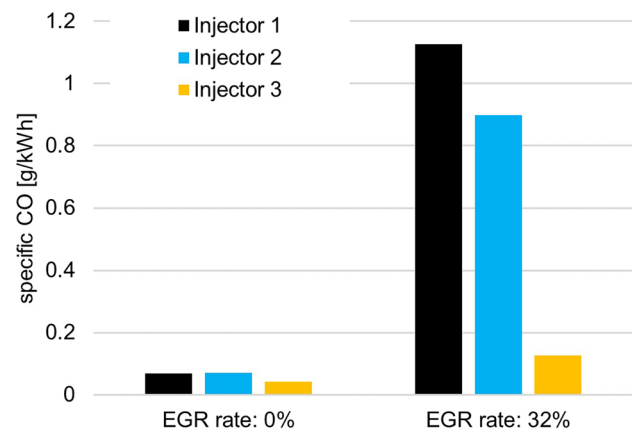


Fig. 10 Indicated specific CO emissions in operation with and without EGR (PL2)

to the other two injectors. In operation with EGR, CO emissions increase significantly, especially when using *injector 1* and *injector 2*. *Injector 3* also shows a slight increase, but much less pronounced and at a level which is still relatively close to that of operation without EGR.

In the case of *injector 1*, the lower injection rate results in a reduction of combustion speed. Especially in lean areas, slower combustion and thus lower combustion temperatures lead to an increase in CO emissions [22, 23]. With *injector 2*, on the other hand, the injection rate is much higher, so that combustion occurs much faster and with higher local peak temperatures. Since *injector 2* is an 8-hole injector like *injector 1*, the increased hydraulic flow is achieved via larger hole diameters here. This results in larger injection jets, which have an unfavorable effect on mixture formation and might lead to increasing local zones with a lack of oxygen. In combination with high combustion temperatures, such conditions typically favor the formation of CO [22, 23].

In contrast to *injector 2*, the 12-hole layout of *injector 3* should significantly improve mixture formation and air utilization. The higher injection rate with respect to *injector 1* results in increased combustion speed and higher local peak temperatures. As a result of better mixing with air, there are fewer oxygen-deficient zones to be expected, especially when compared to *injector 2*, thus avoiding the increase in carbon monoxide emissions at high combustion temperatures. It is also expected that higher combustion temperatures will occur in the lean combustion zones than with *injector 1* due to the higher combustion rate, which will also reduce the risk of CO emissions.

The indicated efficiency calculated from these tests is shown in Fig. 11 for the three injector variants. The lowest efficiency is obtained when using *injector 1*, mainly due to the lower injection rate and thus slower conversion speed compared to the other two injectors. As a result of the

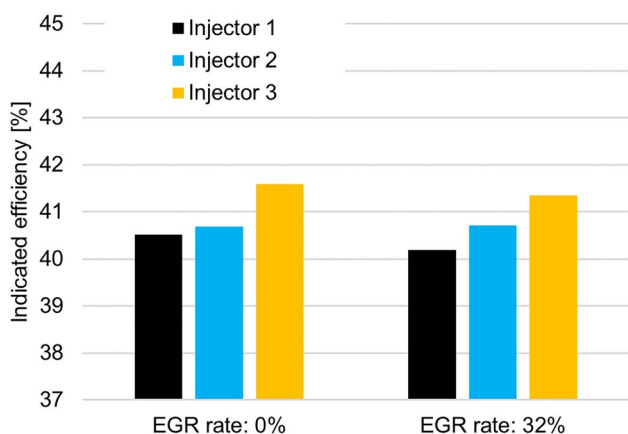


Fig. 11 Indicated efficiency—operation with/without EGR (PL2)

significantly longer injection duration, the end of combustion shifts further into the expansion stroke when *injector 1* is used due to the center of combustion held constant during the investigation. This generally has an unfavorable effect on efficiency, also due to less complete combustion.

Because of its higher injection rate and faster energy conversion, a slightly higher efficiency is achieved with *injector 2*. The highest efficiency is obtained with the 12-hole injector (*injector 3*). This is not only due to the high conversion rate, but probably also to improved mixing and combustion chamber utilization. As a result, combustion conditions are expected to be more favorable compared to *injector 2*, promoting more complete combustion and thus improving efficiency.

As combustion (or the conversion rate) is slower in operation with EGR, the end of combustion is delayed into the expansion cycle due to the center of combustion kept constant during the variation. This results in a corresponding decrease in efficiency. However, *injector 3* still has the highest efficiency potential, and *injector 1* the lowest.

In the following, the resulting nitrogen oxide and carbon monoxide emissions at low load are briefly discussed using the example of operating point PL1 (engine speed: 1250 rpm; IMEP: 3.7 bar). For the investigation, the engine was operated with three pre-injections using the injection application data (electric energizing duration of the injector) of conventional diesel operation. All measurements with EGR were made at an EGR rate of approx. 44%.

Figure 12 shows the NO_x emissions for measurements with pure OME in operation in PL1. In contrast to PL2, there are hardly any differences between the injectors both in case of operation without EGR and with EGR.

As mentioned previously (Sect. 4), the low emission level at this operating point compared to the other fuels can only partly be attributed to the lower hydraulic flow of *injector 1*. This is confirmed by the results shown in Fig. 12,

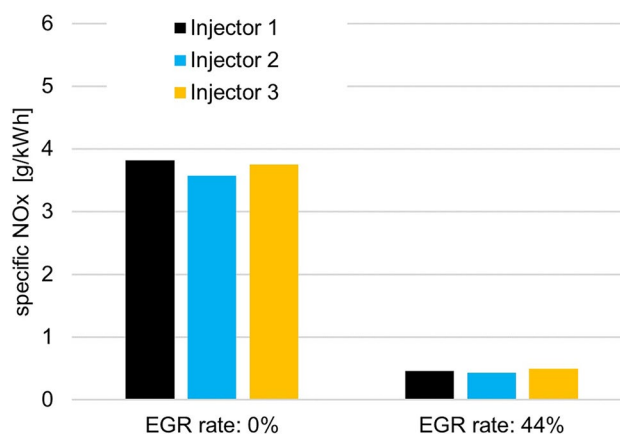


Fig. 12 Indicated specific NO_x emissions in operation with and without EGR (PL1)

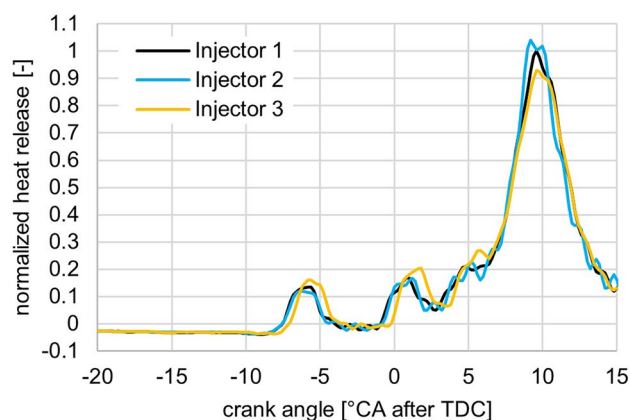


Fig. 13 Normalized heat release for operation without EGR (PL1)

as otherwise a significant increase in nitrogen oxide emissions would have been expected for the other two injectors. It must be taken into account that at this operating point, approx. 40–50% of the fuel is injected via the pilot injections. Accordingly, only 50–60% of the fuel is introduced through the main injection. As a result, the influence of the lower hydraulic flow from *injector 1* on the main injection duration is much less significant than at higher load operating points.

Figure 13 shows the heat release curve, again normalized to the maximum value of *injector 1*. The negative heat release in the compression phase is due, in part, to the assumption of constant material properties and heat transfer. Apart from the fact that the graph of *injector 3* is shifted slightly later due to its minimally delayed SOI, there are almost no differences between the heat curves of the individual injectors, in contrast to the previously considered high-load operating point. An influence of the lower hydraulic

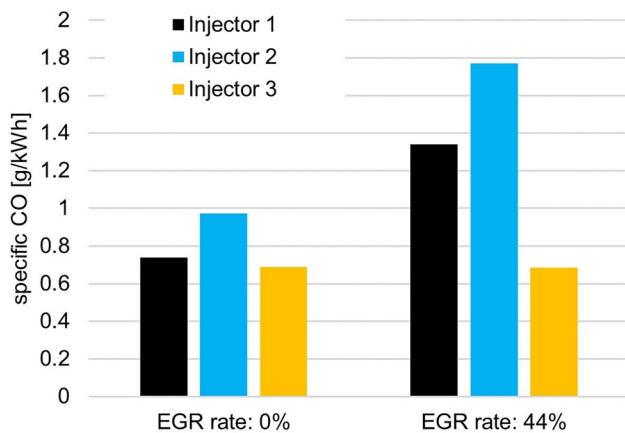


Fig. 14 Specific CO emissions—operation with/without EGR (PL1)

flow of *injector 1* cannot be observed here. The slightly lower NO_x emissions for *injector 2* could be attributed to poorer mixture formation due to the injector design. As a consequence, the occurrence of combustion zones with conditions favorable for NO_x formation is expected to decrease.

The results for specific CO emissions are shown in Fig. 14. The highest CO emissions are obtained with *injector 2*, both in operation with EGR and without. Again, due to the injector design, it can be expected that *injector 2* tends to produce more oxygen-deficient combustion zones at simultaneously high combustion temperatures than the other two injector variants, which helps to explain the slightly higher CO emissions. When using the 12-hole injector (*injector 3*), CO emissions are still produced in lean combustion zones occurring due to the expected wider spatial distribution of the injected fuel. As a result of the relatively slow combustion, low combustion temperatures are likely to occur especially in these zones, which will favor CO emissions.

In general, CO emissions again tend to increase in operation with EGR due to the reduction in combustion temperature and oxygen content. The fact that for *injector 3*, on the contrary, again no increase in CO emissions is observed for this EGR rate compared to operation without EGR could also be due to the improved spatial distribution of the injected fuel mentioned above. Compared to the other two injectors, this might in this case at least counteract the occurrence of oxygen-depleted combustion zones. With *injector 1* and *injector 2*, on the other hand, a significant increase in carbon monoxide emissions is observed. In the case of *injector 1*, CO emissions are probably attributable—at least to some extent—to rather lean combustion zones combined with low local combustion temperatures. As the combustion rate decreases during operation with EGR, the local combustion temperatures also decrease even more, promoting incomplete combustion and thus CO emission. With

increased EGR content, more oxygen-deficient zones will occur, which also has an unfavorable effect on CO oxidation. This is particularly true for *injector 2* due to the lower quality of mixture formation as a result of the injector design. As the oxygen-deficient zones continue to increase in EGR operation, this also results in a corresponding rise of CO emissions.

6 Summary and conclusions

A major focus of this work was the investigation of HVO–OME blends with particular emphasis on soot and NO_x emissions. While OME blends with fossil diesel fuel have already been extensively investigated, much less was known about the behavior of HVO–OME blends and in many cases was limited to blends containing additional components whose proportion even exceeded that of the OME in part. The objective of this work was therefore to improve the knowledge of HVO–OME blends and, in doing so, to minimize as far as possible the proportion of additional blend components that serve as blending agents. The fuel comparison studies discussed here have shown that HVO–OME blends bring about a considerable potential for reducing soot emissions compared to conventional diesel fuel, and in some cases also in comparison to HVO. In particular, Blend C (HVO/OME3-5: 70/30 vol.-%) demonstrates a very high soot reduction potential, particularly at low loads. However, it is still necessary to identify a suitable blending agent which does not significantly deteriorate the emission behavior. The other two blends (containing less OME) also offer significantly improved EGR compatibility and thus the possibility of achieving strongly reduced nitrogen oxide emissions with EGR, combined with ultra-low-soot emissions. Blend B should be mentioned in particular here, as this blend still lies within the diesel standard DIN EN590 in terms of its fuel properties (including density), which would significantly simplify the practical use of such a fuel blend from the point of view of the fuel standardization required for commercial applications. The almost soot-free combustion resulting from operation with pure OME is a characteristic which is already well known for OME and could also be confirmed in this investigation.

Another focus of the current work was the investigation of the influence of different injector designs on operation with pure OME. During the investigation of pure OME using three different injector variants (cf. Table 3), the results indicate that the design of *injector 3* (12-hole design with increased flow) appears to be more suitable for operation with OME than *injector 2* (8-hole design with increased flow). In comparison with *injector 1*, both *injector 2* and *injector 3* were able to operate at higher conversion rates and faster combustion due to the hydraulic flow adapted to

the calorific value of OME, particularly in high-load operation. This resulted in an increase in efficiency, but also an increase in nitrogen oxide emissions. Since with *injector 3*, the adjustment of the hydraulic flow rate was achieved by a larger number of injection holes, there were advantages in mixture formation and combustion chamber utilization compared with *injector 2* and its enlarged injection hole cross-sections. This resulted in lower carbon monoxide emissions, higher efficiency, but also higher nitrogen oxide emissions when operating with *injector 3*. However, the increase in NO_x levels is still manageable and can be considerably reduced in operation with EGR, as expected. Future studies to optimize OME operation will therefore be based primarily on the *injector 3* variant.

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Declarations

Conflict of interest All authors declare that there is no conflict of interest.

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