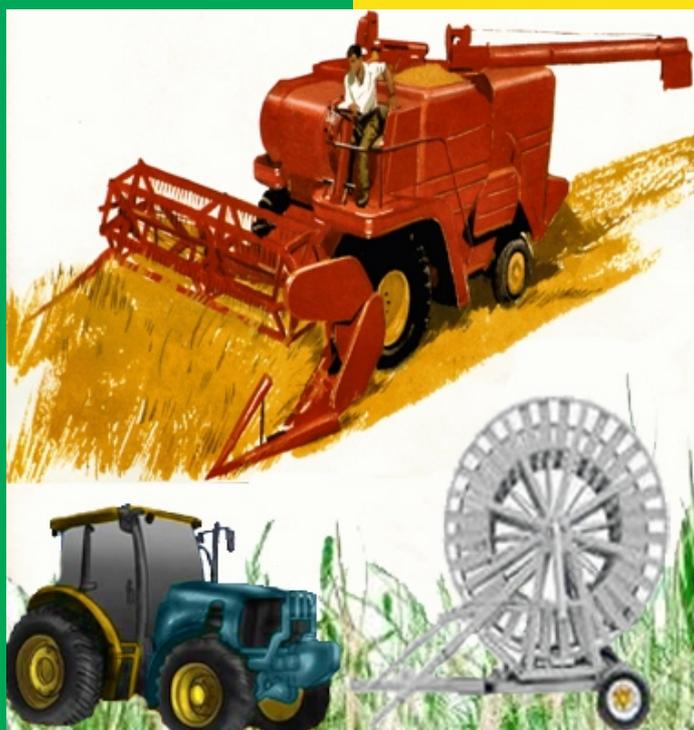


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Utilisation of sustainable Ethanol in a Dual-Fuel Diesel Engine

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Abstract: *The dual-fuel combustion of ethanol and two different mixtures of ethanol and gasoline (E85 and E65) in a modified Diesel engine was investigated. With rising alcohol amount, a significant reduction of soot mass and particle count was observed at all operating points. At some load conditions, substituting diesel with ethanol, E65 or E85 led to a reduction of the NO_x emissions, however, the real benefit concerning the nitrogen oxides was introduced by the mitigation of the soot-NO_x trade-off, allowing higher EGR rates. With regard to the engine efficiency aspect, the results showed a bidirectional behaviour: at low load regimes engine efficiency degraded, whereas combustion became remarkably more efficient at higher engine loads. The measurements showed a high reduction of engine CO₂ emissions in all cases, with the reduction being proportional to the alcoholic fuel energy share at the combustion. Ethanol sustainability is discussed based on the case study of an Austrian bio-ethanol producer.*

Keywords: DUAL FUEL ENGINE, DIESEL, SUSTAINABLE ETHANOL, EMISSIONS, EFFICIENCY ANALYSIS

1. Introduction

The transport sector is a major contributor to the greenhouse gas emissions in the European Union with a share of about 23 % in 2019 (excl. international aviation and international maritime transport) [1]. Reducing CO₂ from traffic is crucial for achieving the desired climate stabilizing objectives and lowering the dependence on fossil fuels. The efforts in this field should be supported by rational as well as technically and economically feasible sustainable solutions. Increasing the usage of biofuels is one of these solutions. Ethanol, in particular, is a very well-known fuel produced from different sources and by various methods.

Despite the present alternative powertrain development, the diesel engine is predicted to remain crucial for mobility of people and goods in the long term due to its unmatched efficiency, robustness and operation cost and flexibility. Therefore, the search for diesel substitute fuels that are sustainable and of wide availability is of high priority. The usage of ethanol is normally assumed with spark ignition engines, but not with a compression ignition engine, because of the very low cetane number of this fuel. However, there are several different possibilities to utilise this alcohol in a diesel engine and it is expected to defuse the classical soot-NO_x trade-off due to its high oxygen content, advantageous molecule structure and better mixture formation and combustion process. Ethanol can be used as a single fuel in a dedicated ethanol compression ignition engine with a higher compression ratio. Such engines are available on the market. To ensure ethanol ignition, however, fuel additives are necessary [2]. Adding the alcohol to the conventional diesel fuel in the form of a so-called “blend” is another way of using it in a diesel engine. Though it is the simplest method, the ethanol quantity is very limited due to separation issues, poor ignitability and current diesel norm EN590 violation in terms of cetane number and flash point. Perhaps the most promising technique for ethanol usage in a diesel engine is the unconventional application in the form of a dual-fuel combustion process. In this case, the alcoholic fuel is fed into the engine as a separate fuel and the ignition of the directly injected diesel fuel starts the combustion of the alcohol-air mixture. This approach provides high substitution ratios and flexible operation based on the fuel availability – diesel only or dual-fuel.

The dual-fuel combustion process itself is not a novelty, it has been used in certain applications since the very early age of the internal combustion engine with different fuels, mostly natural gas [3, 4], but also liquefied petroleum gas [3], gasoline [5], hydrogen [3], reformed gases [6, 7] and other alternative fuels [8]. It was in 1901 that Rudolf Diesel obtained a U.S. patent covering the concept of the dual-fuel engine [9]. Similarly, to the other low reactivity fuels mentioned, alcohols like ethanol or methanol [10, 11] can also be very successfully utilised in a dual-fuel designed diesel engine. There are several possibilities to configure the dual-fuel system depending on the mixture formation method – external or internal. On the one hand, the alternative fuel can be added to the intake air upstream via manifold injection [12], carburetor or evaporator – a

method widely known as “fumigation”. On the other hand, the fuel may be directly injected into the combustion chamber through a dedicated injector or a special dual-fuel injector for diesel and alternative fuel.

This study handles the operation of a diesel engine in a dual-fuel mode with an intake manifold ethanol injection. Several researchers have published articles on investigations of the dual-fuel combustion process with alcohols. The findings, however, are not always consistent and differ at some points, as can be concluded from a comprehensive review of many publications on this topic in [13]. In terms of NO_x emissions, most articles report a reduction with increasing ethanol percentage [13, 14] but statements about NO_x increase can also be found like in [15] or [16]. Regarding engine efficiency, it is found to increase with higher ethanol percentage in [17], while other authors report that a positive change in engine efficiency occurs only at higher engine loads and the rise of ethanol share at lower loads even degrades engine efficiency [10] [18, 19]. The literature review did not offer satisfactory explanations for the observed efficiency changes with rising amount of intake manifold alcohol injection; therefore, an engine-process model for the test engine was generated and selected operating points from the engine experiments were analysed with it. The conducted efficiency loss analysis supports the understanding of the measured results.

Furthermore, the role of bioethanol as a biogenic energy carrier is introduced. Special attention is given to the production of bioethanol and the greenhouse gas saving potentials of this fuel. A case study of AGRANA’s bioethanol plant in Austria is presented.

2. Bioethanol-fuel from socio-economic perspective

Currently, bioethanol is mainly obtained as a so-called first-generation biofuel and blended to fossil gasoline. That includes every conventional fermentation procedure for agricultural raw materials containing sugar and starch in which the fruit of the respective raw material plant is exploited. The key advantage of second-generation biofuel production is that, in future, any kind of biomass such as timber, cellulose or vegetable waste could be utilised. Though being very promising due to the reduced competitiveness to food production, these fuels are still produced only in insignificant amounts. According to the World Bank [20], more than 50 % of food price increases derive from crude oil prices, not from the usage for production of biofuels. In 2008, just 4 % of the world’s grain was used for bioethanol production, taking the combined production of animal feed into consideration. Since 2008 EU Biofuels production increased by 68 % [21], while global food prices dropped by 20 % [22]. Moreover, the Food and Agriculture Organization of the United Nations (FAO) and the International Food Policy Research Institute (IFPRI) recognize the chances sustainable and ethically maintainable biofuel production could offer for both food security and modern bioenergy development [23, 24]. Only around 2 % of EU produced grain and only 6 % of EU

sugar substrate are utilised for bioethanol production. The most important byproduct of ethanol production is a high-quality protein animal feed DDGS (Distiller's Dried Grain with Solubles), which helps reduce the soya imports from abroad, meaning that soya-growing areas in export countries can be used to grow foodstuffs.

In Europe, but especially in Austria, a stringent set of rules and environmental regulations apply. All economically useful crops, including those destined for energy production, are cultivated in line with strict environmental criteria. It is illegal to cut down forests to produce bioethanol in Europe. Energy crops to be used for bioethanol production are grown on existing areas of land or on land which had previously been forcibly laid fallow in order to limit exports and which was once again made available for agricultural production by the EU Commission in 2009.

Despite a perceptible trend towards drive train electrification in the road transport, liquid fuels are expected to play a crucial role in defossilising mobility in the next decades, especially in hard-to-electrify sectors. Organizations like the International Renewable Energy Agency (IRENA) and the International Energy Agency (IEA) often call for more biofuels in EU's policy and stress on the importance of bioethanol [25, 26] for achieving the Union's energy and climate targets. The United Nations Framework Convention on Climate Change (UNFCCC) also addresses the strengths of bioethanol in reducing CO₂ emissions from transport [27].

The European Union Directive 2009/28/EC on the promotion of the use of energy from renewable sources set the following targets to be achieved by 2020: a 20 % cut in greenhouse gas emissions and 20 % renewable energy, with 10 % share of energy from renewable sources in transport [28]. The new Renewable Energy Directive 2018/2001 (known as RED II) [29] prescribes 14 % renewable energy share in transport by 2030 and confirms the importance of sustainably produced crop-based biofuels such as European ethanol for achieving EU climate goals. However, it still caps at 7 % energy share the contribution these low-carbon fuels can make. So-called "advanced" biofuels (e.g. from straw or non-food cellulosic material) should account for at least 3.5 % energy share and can be double-counted. In accordance to this directive, the greenhouse gas emission savings for biofuels produced in installations starting operation from 1 January 2021 shall be at least 65 % compared to fossil fuels.

3. Bioethanol by AGRANA

Production

In Pischelsdorf (Lower Austria), AGRANA Stärke GmbH operates a bioethanol fuel plant and a starch factory, collectively referred to as "biorefinery". The residual starch slurry produced in the starch factory is processed into bioethanol and animal feed DDGS and wheat gluten feed at the bioethanol plant. Up to 650 000 tons of grain per year can be converted to 260.000 m³ or 210.000 tons of bioethanol. In addition to the fuel, up to 160.0000 DDGS and 110.000 of wheat gluten feed (mixture between bran and syrup) can be produced each year. This high-quality, certified GMO-free protein-rich animal feed helps to make the production of bioethanol commercially viable, with the quantities produced replacing round about a quarter of Austria's soya imports from countries that can no longer guarantee GMO-free production. As a by-product of bioethanol production, up to 90.000 t/a of biogenic carbon dioxide are produced for the food industry and various technical applications.

The raw material sources for the bioethanol production are surplus cereals such as wheat, rye, triticale, barley and maize, all in animal feed quality. Food quality crops are not used. A new additional facility started operation in 2019 and today, the starch-containing residuals from the wheat starch processing at the starch factory (second flour, small grains and residual starch slurry) make approx. 45 % of the bioethanol raw material. Especially the residual slurry from wheat starch production is converted into approx. 80.000 m³ of bioethanol with this compound plant.

The delivered raw material is fed via a silo intermediate storage to the grinding, suspended by addition of water, then enzymatically and thermally liquefied and introduced into a simultaneous saccharification and fermentation. In the course of the saccharification, the dextrans contained are decomposed by enzymes into monosaccharides. This raw material is then converted by yeast fermentation into an alcoholic mash, which is subsequently processed in a multi-stage distillation to raw alcohol. Via molecular sieves, a further dehydration of the raw alcohol to fuel alcohol is carried out.

The protein-rich distillation residuum (distiller's wash) is centrifuged. The thin slurry (centrate) is concentrated in the evaporation to syrup or occasionally recirculated to the liquefaction process. The thick slurry (wetcake), which are the centrifugalized fiber residuals, is mixed with the syrup and then carefully dried and pelletized to storable DDGS.

The bioethanol plant is equipped with an energy supply system: a thermal waste incineration plant owned by the EVN group supplying high-pressure steam. The carbon-based calorific power plant located nearby was shut down. The supply of steam is coming from the waste incineration plant and a backup steam boiler. The total efficiency of the industrial compound (electricity and steam) is hence higher than for a stand-alone facility. The energy sources for steam generation are 100 % waste, coal is no longer used. For power generation AGRANA purchases green electricity.

Greenhouse gas emissions calculation

In order to assess the sustainability of bioethanol production at the AGRANA facilities, life-cycle analyses are carried out. A life-cycle analysis (as defined by ISO 14040) calculates emission levels and the cumulated primary energy demand which are associated with providing a certain transport service using a bioethanol or petrol-driven car. Each individual material and process involved in both transport services is recorded ("from the cradle to the grave"), ranging from the way in which the raw material is extracted from the environment to how material and energy is discharged. This analysis includes every process related to emissions and energy, both domestically and abroad, which is necessary to run vehicles on bioethanol. Finally, the results are compared with the assessments of the supply and application of petrol. When producing bioethanol and DDGS, for the biogenic carbon used it is assumed that the balance of net carbon fixing during photosynthesis, the amount of carbon stored and the combustion of bioethanol and DDGS use, is zero, as set out in the guidelines for the energy industry produced by the Intergovernmental Panel on Climate Change (IPCC).

As a biofuel producer, AGRANA has to provide the results of the greenhouse gas savings through external certifications in accordance to the International Sustainability and Carbon Certification (ISCC); Fig. 1 and Fig. 2.

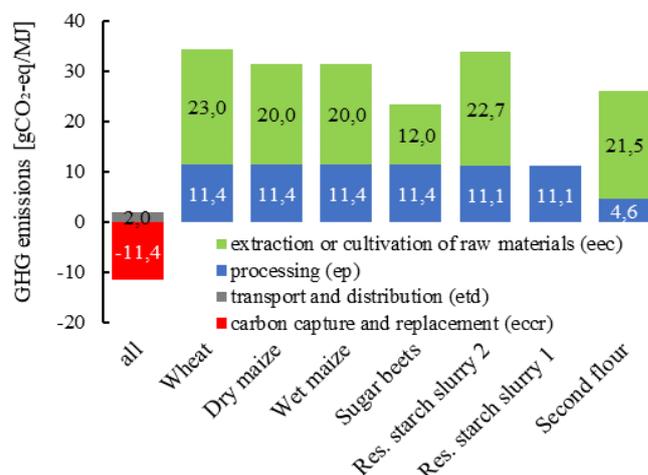


Fig. 1 GHG shares calculated in accordance to EU Directive 2009/28/EC for bioethanol from different raw materials; starch slurry 1 calculation

Some countries, like England, accept that no cultivation emissions are to be accounted for the raw material residual starch slurry (calculation method ISCC EU). The greenhouse gas savings from bioethanol produced by AGRANA are at least 68 and are certified by ISCC.

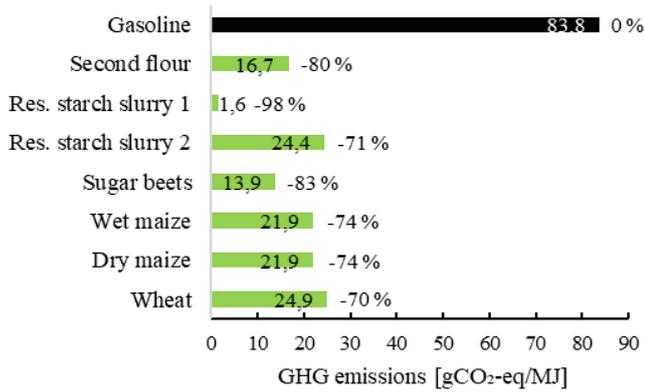


Fig. 2 GHG emissions and reductions calculated in accordance to EU Directive 2009/28/EC for bioethanol from different raw materials; starch slurry 1 calculation method ISCC EU, starch slurry 2 – ISCC DE

4. Investigated Fuels

Table 1: Properties of the investigated fuels

Fuel notation	Diesel	EtOH	E85	E65
Density [kg/m ³]	831	789	782	773
Lower heating value [MJ/kg]	42.68	26.68	29	31.46
Cetane number [-]	54	8	-	-
Research octane number [-]	-	111	105	101
C-fraction [wt %]	85.4	51.5	56.5	62.8
H-fraction [wt %]	14.2	13.7	12.8	13.8
O-fraction [wt %]	0.4	34.1	29.8	23.3
H/C atomic ratio [-]	1.99	3.2	2.72	2.65
O/C atomic ratio [-]	0.004	0.5	0.4	0.28
gCO ₂ /MJ	73.3	71.22	72.02	73.12
gH ₂ O/MJ	29.94	46.54	40.08	39.69

Table 1 gives an overview of some of the properties of the tested fuels.

In this study, a certified CEC test diesel fuel was used as a reference for comparison. Technically pure ethanol was the first representative of the investigated oxygenated fuels. However, the intake manifold injected fuel does not necessarily have to be of high purity – a great advantage lies within the possibility of using lower-grade ethanol, since even higher water contents had no harmful impact on the engine. Additionally, E65 and E85 gasoline-ethanol mixtures were supplied, because their availability at filling stations is more presumable. E85 (RF-01-08) was delivered by a supplier and E65 was self-mixed (35 vol% winter quality gasoline RF-04-03).

5. Test engine and test methodology

Test engine and measurement equipment

A modern in-line four-cylinder diesel engine was modified for single cylinder operation and used as a test engine. Only the first cylinder was fired, while the other three were deactivated and their gas exchange was separated from the gas exchange of the fired cylinder. The cylinder had a displacement of 537 cm³ and a compression ratio of 17.5. Bore and stroke were 88 and 88.34 mm, respectively. The original high-pressure common rail diesel injection system was equipped with seven-hole nozzle solenoid injectors and could handle a pressure of up to 2000 bar. Fig. 3 depicts a schematic layout of the test bench. Charging was performed with an external electrically driven supercharger. An electrically adjustable flap valve was used to adjust a turbocharger

comparable exhaust gas back pressure. Cooled high-pressure exhaust gas recirculation (EGR) was applied. The standard exhaust gas aftertreatment system was removed and emission sampling took place in the raw exhaust gas. The temperatures of the charge air and the fuel were conditioned. The alcoholic fuel was injected with a pressure of 4.5 bar (abs.) into the intake air through a conventional gasoline injector, which was installed in a specially designed intake manifold.

The test engine was equipped with high- and low-pressure indication and standard exhaust gas measurement, as well as AVL Micro Soot Sensor 483 and AVL Particle Counter 489.

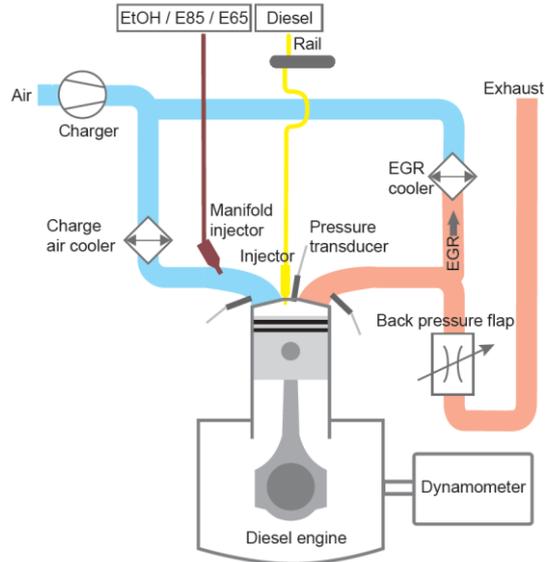


Fig. 3 Schematic layout of the test bench

Test methodology

The investigations were conducted at five operating points, which cover a typical commercial vehicle engine operating map. The most important engine operating parameters are depicted in Table 2. Variations of the exhaust gas recirculation (EGR) rate, of the combustion center position MFB50% (by adjusting the start of injection) and of the proportion of diesel fuel to alcoholic fuel (as percent energy share of the alcoholic fuel in relation to the total fuel energy, [%e.]) were carried out. At the dual-fuel investigations, the manifold fuel injection started at 0 °CAaTDC (firing). The pure diesel operation was the reference for the comparison. All specific emission values in this article are with respect to the indicated engine work. The results at OP-1500/5 may be seen as representative for OP-2000/5, therefore OP-2000/5 will not be discussed in further detail in the following sections. The diesel fuel was always injected through a single injection event, multiple injection strategies were not pursued within the scope of the investigations. Given the fact that the combustion calibration of modern diesel engines advances towards multiple injection systems with enhanced pressure, it should be borne in mind that a diesel engine with an optimised injection hardware and strategy would deliver better results regarding combustion shaping, efficiency and emissions [30]. The dual-fuel combustion process shown in this paper, however, also offers significant optimisation potentials, since neither the diesel combustion chamber has been modified, nor has the manifold injection been designed to fulfill an optimum.

Table 2: Investigated operating points with engine parameters

Operating point OP-	1500/5	1500/15	2000/5 ^a	2000/20	3000/20
Speed [rpm]	1500	1500	2000	2000	3000
IMEP [bar]	5	15	5	20	20
Power Output [kW]	3.3	10	4.5	17.9	26.8
Rail pressure [bar]	~670	~1040	~800	~1240	~1400
Int. manifold pressure [mbar]	1130	2250	1230	2500	2600
Δp exhaust-intake [mbar]	270	500	330	300	300

^a Will not be discussed in further detail due to similarity to OP-1500/5

The reproducibility of the measurements was proved by conducting the experiments three times. The results were carefully analysed and the middle value was built for the representations.

Simulation model for efficiency loss analysis

Besides the experimental investigations at the engine test bed, a 0D/1D-simulation model of the combustion process was additionally set up and selected operating points were analysed with it. The software used was GT-POWER. An important feature of the simulation is the optional efficiency loss calculation, which contributes to the understanding of eventual efficiency differences between the discussed operating modes. The approach used for burn rate calculation was three pressure analysis (TPA) and required three measured pressure curves: intake, cylinder and exhaust. The test bed measurements delivered the input and calibration data for the simulation.

6. Results of the investigations

In the course of the investigations, at low load, degrading combustion quality, poor controllability and high cycle variation determined the maximum possible alternative fuel amount. In this project, a maximum substitution rate of 60-70 % energy share was possible with the fuels used. At high loads it was the occurrence of knock and auto-ignition that prohibited further increase of the substitution ratio. Nevertheless, at OP-1500/15 still 50-60 % e. (% energetic) substitution rate was achieved; the corresponding ranges for OP-2000/20 and OP-3000/20 were measured to be approximately 40 % e. and 25 % e., respectively. In accordance with its higher octane number, pure ethanol ensured slightly higher substitution rates at high load testing, whereas at OP-1500/5 it failed to provide stable combustion before E65 and E85 did.

Polluting emissions

NO_x , soot and particle emissions

The results of the NO_x and soot emissions measurements over the substitution rate for the tested operating points are depicted in Fig. 4 and Fig. 5. The soot mass in the exhaust gas was very effectively reduced at all operating points even when a relatively small diesel amount was replaced by ethanol, E85 or E65. Several factors play an important role in minimizing the soot emissions at dual-fuel operation. Ethanol per se is a substance known for its almost soot-free combustion. The molecule contains only two carbon atoms, which are single bonded, the H/C ratio is higher than for many other fuels and the oxygen atom is available for the combustion directly within the reaction zone. Furthermore, the low boiling point of 78°C promotes faster homogenisation. Distinctive for the dual-fuel combustion process with external alcohol supply is the fact that enough time is available for the air-fuel mixture to be premixed and homogenised before the diesel jet induces the ignition. Thus, the local fuel-rich zones are less and the formation of soot is limited. The lower aromatics amount also reduces the tendency to form soot precursors [31].

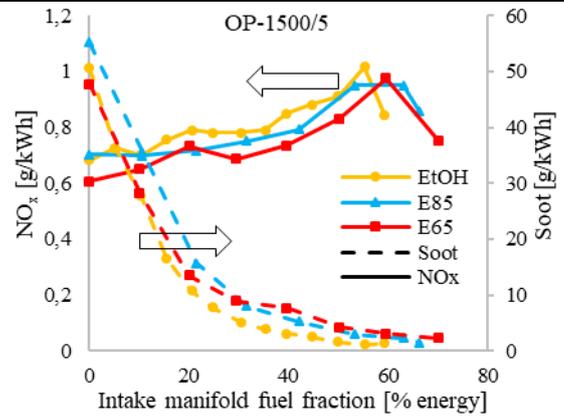


Fig. 4 NO_x and soot emissions at OP-1500/5, 30 % EGR, $MFB50\% = 7.5^\circ C_{aaTDC}$

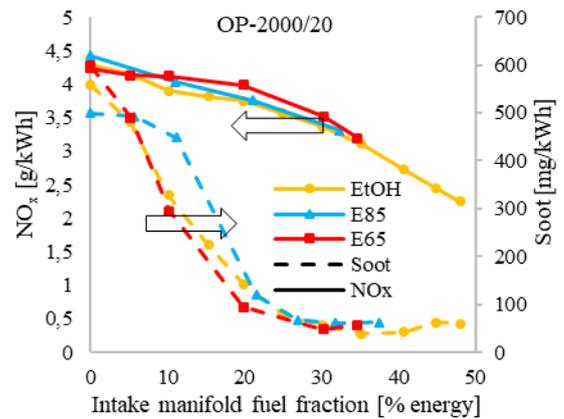


Fig. 5 NO_x and soot emissions at OP-2000/20, 0 % EGR, $MFB50\% = 17.5^\circ C_{aaTDC}$

By replacing diesel, the appearance of diffusive diesel combustion is reduced and therefore less soot is produced. Ethanol has also a very high enthalpy of evaporation, which causes a strong cooling effect. With the cylinder charge temperature sinking, the injected diesel fuel needs more time to auto-ignite, its ignition delay is prolonged and more time remains available for diesel spray homogenisation.

The behavior of the NO_x emissions over the substitution rate depends on the load conditions: at OP-1500/5 an increase was measured, while a strong reduction occurred at the tests with 20 bar IMEP. The most important factors for NO_x formation are temperature, reactant availability and time. On the one hand, the extended ignition delay results in a faster and more compact combustion with higher heat release and higher peak temperatures. This aspect and the higher oxygen availability, which ethanol provides, increase the NO_x production. On the other hand, the charge cooling effect of the ethanol evaporation decelerates the pollutant onset. The combustion of the premixed mixture reduces the local peak temperatures because the heat is released in the whole combustion chamber and not only in a small area around the diesel jet. Higher thermodynamic combustion efficiency means that less fuel is burnt and therefore less heat is released, which also may lower the NO_x output. Additionally, ethanol burns with lower flame temperature than diesel due to the composition of the combustion products [2, 32]. Whether the NO_x emissions rise or sink when substituting diesel through manifold injected alcoholic fuel depends on the weight and occurrence of the factors listed above.

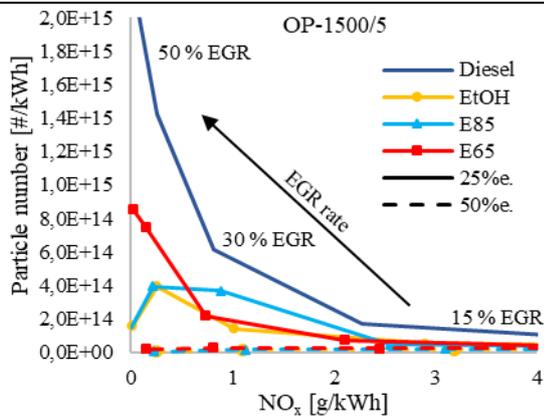


Fig. 6 Particle number-NO_x trade-off at OP-1500/5, MFB50% = 7.5°CaaTDC

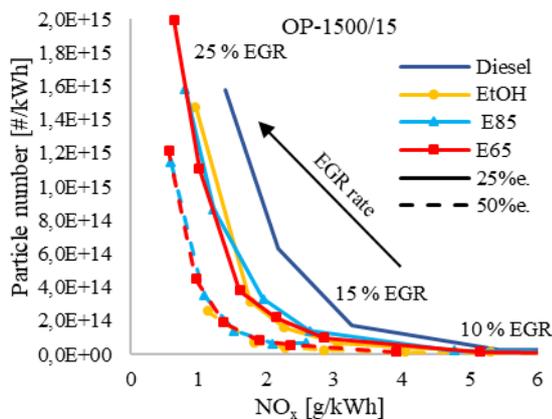


Fig. 7 Particle number-NO_x trade-off at OP-1500/15, MFB50% = 7.5°CaaTDC

Together with the soot mass, the particle number also dropped as diesel was replaced by the alternative fuel. By mitigating the smoke problem of the diesel engine, more EGR could be used as an in-cylinder measure for further NO_x emission reduction – the diesel typical soot-NO_x trade-off was weakened, Fig. 6 and Fig. 7. An outstanding advantage is visible for dual-fuel operation mode, in particular for OP-1500/5. In the case of 50 %e. EtOH, at 40 % EGR and a NO_x level of 0.25 g/kWh 3.5 mg/kWh of soot and an FSN of 0.043 was measured.

HC and CO emissions

Compared to the diesel reference operation, the output of unburnt or partly burnt fuel was found to be considerable at dual-fuel mode, especially at low loads. The amount of unburnt hydrocarbons (HC) and carbon monoxide (CO) steeply rose with the substitution rate. At OP-1500/5 the HC emissions increased by approx. 2 g/kWh per additional 10 %e. alcoholic fuel. This severe disadvantage resulted in lower engine efficiency; Fig. 8. The higher load points provided higher in-cylinder temperatures and pressures, hence ignition and combustion of the cylinder charge were much better supported and the incomplete combustion influence on the engine efficiency diminished.

Efficiency and CO₂ emissions

Increasing the substitution rate at OP-1500/5 resulted in a considerable drop of engine efficiency until approximately 30 % energy share and affected no significant further changes up to the maximum possible alcoholic fuel amount; Fig. 8. In contradiction to this case, a very pronounced efficiency gain was measured at the medium and high load operating points, as can be seen in Fig. 8 and Fig. 9. Due to prolonged combustion duration and associated efficiency loss, the highest efficiency at “diesel only” operation was

measured at a MFB50% of 17.5°CaaTDC for OP-2000/20 and at a MFB50% of 20°CaaTDC for OP-3000/20. Substituting diesel by an intake manifold injected alcoholic fuel moved the highest efficiency combustion center to more advanced MFB50% but, as could be expected, cylinder peak pressure, pressure rise rate and nitrogen oxide emissions were very high. Fig. 8 and Fig. 9 also include an error analysis of the experiments. In order to just outline the repeatability bandwidth of the experiments regarding the measured efficiency without overloading the diagrams, not all measured points are presented with error indicators. As can be seen, the results at OP-1500/5 show the highest deviations due to the higher uncertainties when measuring very small fuel flow rates at the single-cylinder engine.

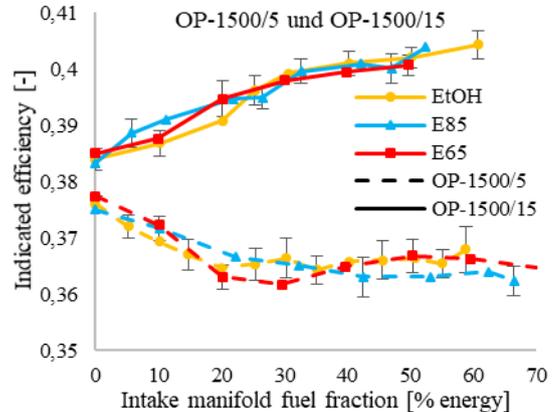


Fig. 8 Indicated efficiency at OP-1500/5 with 30 % EGR and OP-1500/15 with 15 % EGR, MFB50% = 7.5°CaaTDC

Operating the engine in dual-fuel mode with high alcoholic fuel energy shares also introduced a much higher tolerance against high EGR rates; Fig. 10 and Fig. 11. This benefit allowed the application of higher EGR rates without degrading the efficiency the same way as in pure diesel operation mode. In Fig. 11, the already mentioned additional engine efficiency enhancement, which results from advancing MFB50% when operating in dual-fuel mode at high loads, is shown together with the measured NO_x values for some interesting points.

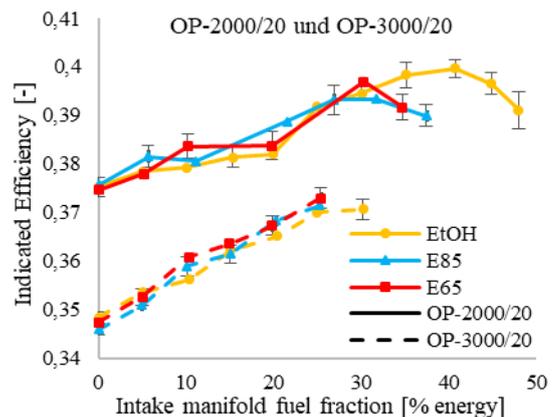


Fig. 9 Indicated efficiency at OP-2000/20 with MFB50% = 17.5°CaaTDC and OP-3000/20 with MFB50% = 20°CaaTDC, 0 % EGR

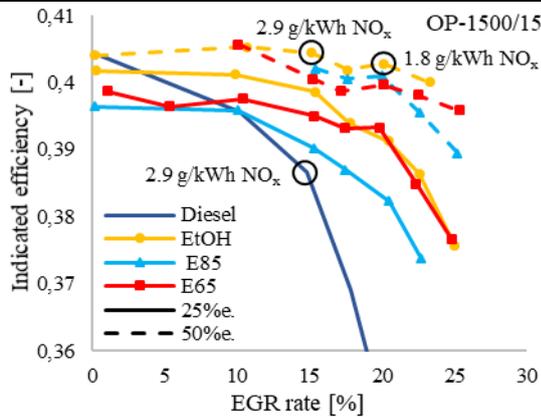


Fig. 10 Indicated efficiency at OP-1500/15, MFB50% = 7.5°C AaTDC

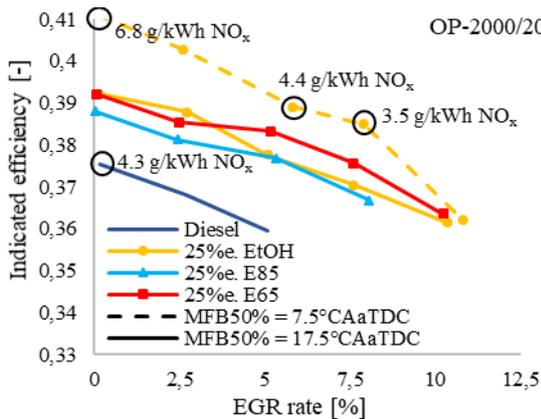


Fig. 11 Indicated efficiency at OP-2000/20

A significant reduction of the raw exhaust gas carbon dioxide emissions (tank-to-wheel) was measured with increasing substitution rate at all operating points, e.g. Fig. 12 and Fig. 13. Apart from OP-1500/5 (and OP-2000/5), the CO₂ reduction reflects the efficiency increase at dual-fuel operation mode, which is illustrated by a comparison between Fig. 13 and Fig. 9. As will be proved later, the poor dual-fuel efficiency at OP-1500/5 was the consequence of high amounts of unburnt fuel, which do not react to CO₂. At dual-fuel operation mode, the part of the fuel that gets ignited combusts thermodynamically more efficient than diesel at “diesel only” operation; therefore, less energy input (= less fuel that ignites and burns) is needed to deliver the same engine power output and less CO₂ is produced. The part of the fuel, which does not ignite and combust, is emitted as HC emissions and reduces the net engine efficiency. Additionally, the CO emissions increased with higher substitution rate. Unlike unburnt fuel (HC emissions), CO is a (incomplete) combustion product and its formation reduces the amount of the complete combustion product CO₂. However, knowing this, the operation at load points with high HC and CO emissions cannot be rated as CO₂ advantageous, since in a real-world application these pollutants would be converted to CO₂ in an oxidation catalyst. For this reason, within a simplified approach the full conversion of HC and CO to CO₂ was calculated for the case with pure ethanol operation following the presumption that all measured HC emissions in fact are unburnt ethanol molecules. The resulting CO₂ emissions are added to the measured CO₂ emissions in Fig. 12 and Fig. 13. As can be seen, at low load operation with ethanol, the theoretical CO₂ emissions are even slightly higher than in “diesel only”. Nevertheless, at medium and high load operation with significantly less HC and CO pollutants, the CO₂ savings are still considerable.

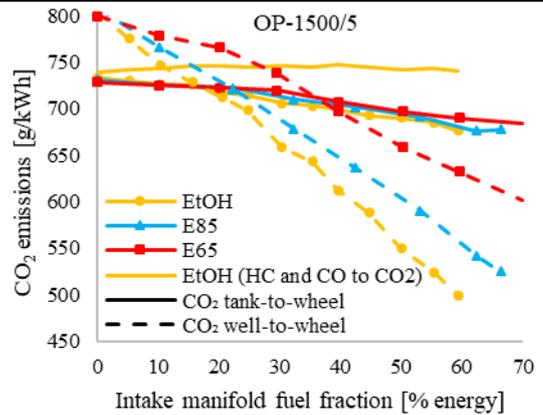


Fig. 12 CO₂ emissions at OP-1500/5, 30% EGR, MFB50% = 7.5°C AaTDC

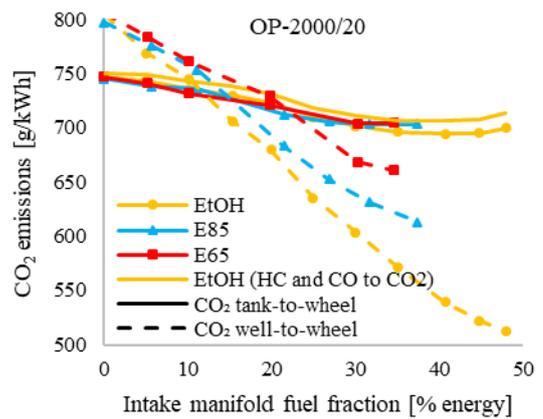


Fig. 13 CO₂ emissions at OP-2000/20, 0% EGR, MFB50% = 17.5°C AaTDC

As can be seen in the diagrams, the regenerative nature of the biofuel allowed remarkable well-to-wheel CO₂ savings. For these calculations, the 70% greenhouse gas saving potential of AGRANA’s bioethanol was adopted.

7. Combustion analysis

The analysis of the indicated data generally showed that at dual-fuel mode the combustion process was characterized by a shorter burning duration, Fig. 14 – Fig. 17. At low load in dual-fuel mode, more time was needed for combustion to start and the start of diesel injection had to be advanced from -11.3°C AaTDC to -13.4°C AaTDC in the case of 40%e. EtOH (Fig. 14). Compared to diesel reference operation, at medium and higher load, the ignition delay in dual-fuel mode was shortened and the same MFB50% was set by a retarded injection timing. The start of diesel injection had to be delayed from -14 in diesel only operation to -7.6°C AaTDC in dual fuel operation with 40%e. EtOH (OP-1500/15, Fig. 15) and from -10.7 to -2.8°C AaTDC for 30%e. EtOH at OP-2000/20 (Fig. 16). For achieving the same MFB50% of 7.5°C AaTDC at OP-2000/20 in dual-fuel operation with 25%e. EtOH, the diesel SOI had to be retarded from -24.1 to -14.4°C AaTDC (Fig. 17). After ignition occurred, the main combustion phase was completed faster than at “diesel only” operation. The post-combustion phase, which is distinctive for the oxidation of the soot formed during the diesel main combustion phase, was in most cases shorter and with a lower heat release. From an efficiency point of view, all these aspects lead to a thermodynamic beneficial cylinder pressure curve shaping.

Besides the above-mentioned, at some conditions a very important combustion phenomenon was detected – a cyclic reproducible auto-ignition of the premixed homogenous charge. Fig. 16 shows the heat release in diesel reference operation and dual-fuel operation mode with 30% energy share of ethanol or E85 at OP-2000/20, MFB50% was at 17.5°C AaTDC. As can be seen,

the combustion in the case of ethanol clearly begins before the start of the diesel injection. After a rapid and intensive heat release, the main combustion of the injected diesel continues at a moderate rate. With E85 (and E65), the combustion starts even earlier and the heat release of the premixed stage is remarkably higher. According to the octane number of the fuels, E85 is more reactive and ignites easier than ethanol. The described auto-ignition of the premixed charge was not accidental, but permanent and stable in every cycle. It appeared with late MFB50% at medium and high load and was more distinct at low speeds. As could be expected, it emitted more noise, but the cylinder peak pressure was only inconsequentially higher than in the diesel reference operation (approx. 150 bar). The experiments proved that the auto-ignition was existent even after the diesel supply had been completely turned off. Fig. 17 shows the same operating point, but with an advanced MFB50% of 7.5°CAaTDC. This combustion center position was much too advanced for diesel reference operation and resulted in an efficiency decrease due to a disadvantageous heat release timing. As can be seen in the graph, in the case of “diesel only”, a lot of energy is released long before the piston reaches the top dead center, thus high blow-by and wall heat losses occur.

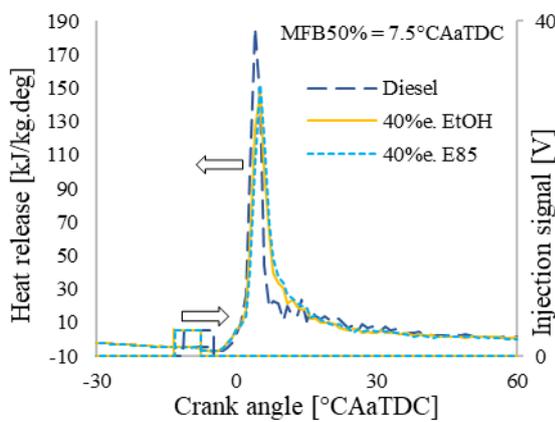


Fig. 14 Indicated heat release rates OP-1500/5, 30 % EGR, MFB50% = 7.5°CAaTDC

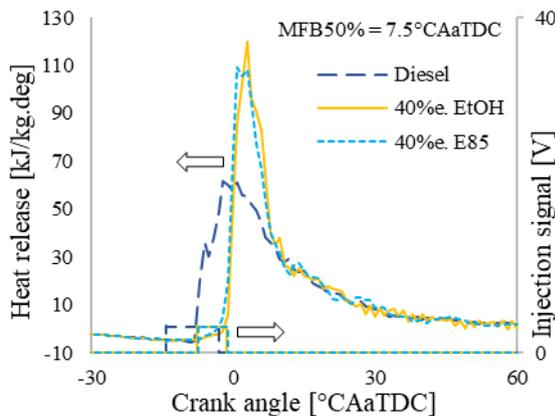


Fig. 15 Indicated heat release rates OP-1500/15, 15 % EGR, MFB50% = 7.5°CAaTDC

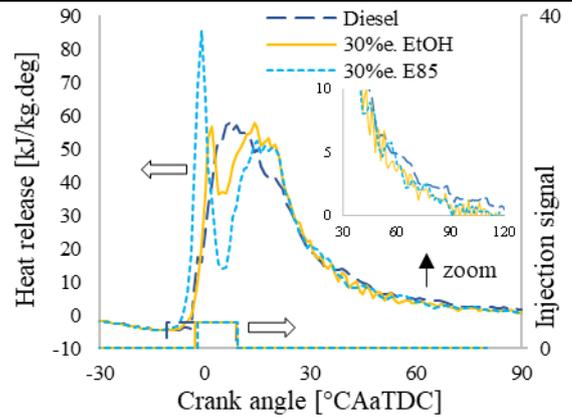


Fig. 16 Indicated heat release rates OP-2000/20, 0 % EGR, MFB50% = 17.5°CAaTDC

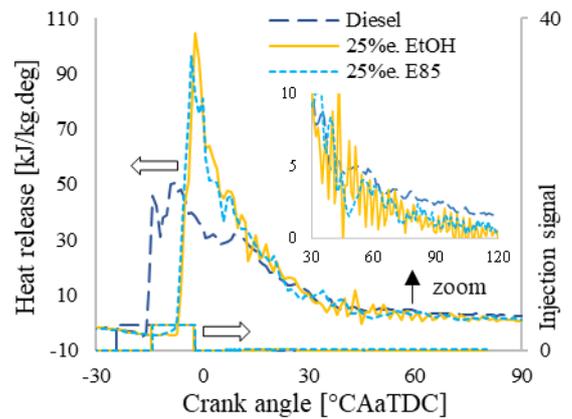


Fig. 17 Indicated heat release rates OP-2000/20, 0 % EGR, MFB50% = 7.5°CAaTDC

Since dual-fuel operation provided a much faster combustion, engine efficiency was even impressively increased at an MFB50% of 7.5°CAaTDC, see Fig. 11. Cylinder peak pressure was, however, in the range of 200 bar and the energy fraction of the alcoholic fuel had to be lowered to 25 %e. to prevent mechanical damage. The maximum pressure rise rate with 25 %e. ethanol was 8.5 bar/°CA at an MFB50% of 17.5°CAaTDC and 18 bar/°CA at an MFB50% of 7.5°CAaTDC. Compared to Fig. 16, advancing MFB50% in Fig. 17 demands a much earlier start of injection and the diesel combustion starts before the theoretical start of the ethanol auto-ignition, so no pre-ignition is detected.

It can be seen from the heat release curves in Fig. 14 – Fig. 17 that the combustion in dual-fuel mode is generally faster than in diesel-only operation. It is commonly known that ignition timing and burning duration have close relationship to emission output. In the case of accelerated combustion, the effect on NO_x emissions and noise becomes of particular interest. The ignition timing of the (not auto-igniting) combustion cycles was determined by the ignition of the direct injected diesel jet. Since both the ignition delay and the burning duration were changed with different substitution ratios in dual-fuel mode, for the following discussion, the combustion center position MFB50% is a more appropriate reference parameter than the ignition timing. Table 3 presents numeric values of the operation cases depicted in Fig. 14 – Fig. 17. The burning duration is specified as the difference between MFB90% and MFB05% and it was significantly shorter in dual-fuel mode at any load point. In OP-1500/5, the reduced burning duration in dual fuel mode was the result of a shorter post oxidation phase in comparison to the diesel reference and not of a higher heat release rate. The faster combustion at OP-1500/15 and OP-2000/20 was accompanied by a steep gradient and high peak value of the heat release. As Table 3 shows, this inevitably led to high noise emissions at dual-fuel mode. As could be expected, the first combustion stage of the auto-ignition

at dual-fuel mode with 30 %e. E85 in Fig. 16 produced a higher noise peak than the combustion with 30 %e. EtOH. Advancing the center of combustion MFB50% in OP-2000/20 from 17.5 to 7.5 °CAaTDC increased the noise level for both diesel and 25 %e. EtOH. It can be concluded that the higher noise emissions at middle and high load in dual-fuel mode are a disadvantage compared to the diesel reference operation mode.

Table 3: Combustion center position, burning duration, specific NO_x and noise emissions for selected points in both operating modes

	MFB50% °CAaTDC	Burning duration MFB90% - MFB05%	Noise dB	ind. NO _x g/kWh
OP-1500/5				
Diesel	7.5	33.7	95.7	0.68
40%e.EtOH	7.5	27.1	92.1	0.83
40%e.E85	7.5	28.8	92.7	0.78
OP-1500/15				
Diesel	7.5	43.8	87	2.94
40%e.EtOH	7.5	30.5	94.3	2.86
40%e.E85	7.5	31.9	94.4	2.92
OP-2000/20				
Diesel	17.5	49.6	86.6	4.3
30%e.EtOH	17.5	41.1	92.3	3.37
30%e.E85	17.5	45.7	97.8	3.36
Diesel	7.5	73.8	91	4.59
25%e.EtOH	7.5	41.8	97.2	6.84
25%e.E85	7.5	43.5	97.1	6.82

Higher noise emissions are normally associated with increased NO_x output. While this is an expected correlation within observations of different combustion shaping at equal operation points and the same engine and combustion process type, no direct relation between the heat release shape and the NO_x emissions can be sought when comparing two different combustion concepts like conventional diesel and dual-fuel operation. The heat release shape is just one of many parameters that influence the formation of nitrogen oxides in a combustion engine and these parameters widely differ for both observed combustion processes. The comparison between the dual-fuel results measured in OP-2000/20 with late and advanced MFB50% and similar substitution ratio (25 and 30 %e.) reveals an extreme increase of the pollutant generation with earlier combustion center position, Table 3. The noise emissions in operation with E85, however, were barely changed. The reason for this behaviour is the similarity of the heat release gradient at the start of combustion for both MFB50% cases: due to the auto-ignition, the operation with E85 produced high noise levels already at a MFB50% of 17.5°CAaTDC. Advancing the combustion center position in this case did not increase the noise level, but only the NO_x emissions due to higher in-cylinder peak temperature.

8. Efficiency loss analysis

An efficiency loss analysis is shown in Fig. 18 for low, medium and high load operation. In agreement with the engine test bed findings, compared to the diesel reference operation, the indicated efficiency in dual-fuel mode is reduced only at the 5 bar IMEP load point. The bar chart clearly visualizes the source of the major efficiency loss – a high percentage of incomplete combustion. At the other investigated load points, a higher share of incomplete combustion is also encountered in dual-fuel mode, it is however significantly lower than at OP-1500/5 and is being overcompensated by a series of efficiency gains.

Due to the changed fluid composition, the real charge and real fluid properties losses sink at all dual-fuel operating points. Since MFB50% and the pressure difference “exhaust – manifold” are held approximately equal for each pair “diesel only – dual-fuel”, the losses due to combustion phasing and gas exchange are also basically identical. The “real burn rate” shares in the diagram prove the accelerated combustion at dual-fuel mode to be one of the main reasons for higher efficiency.

Another loss that is generally reduced when operating with intake manifold alcoholic fuel injection is the wall heat transfer. Surprisingly, the calculated heat transfer loss at OP-2000/20 is equal for both combustion processes. The efficiency loss is calculated by relating the specific particular work loss to the work that would result from the brought-in fuel energy with no losses at all (100 % efficiency). The software calculates 2.6 bar mean effective pressure loss due to heat transfer for the “diesel only” case and 2.5 bar for the dual-fuel case with 30 %e. ethanol, so both cases differ by only 0.1 bar. However, dual-fuel mode operation also requires less fuel energy than diesel reference operation (2.73 vs 2.85 kJ/cycle), so the percentual heat loss share in this case is barely changed.

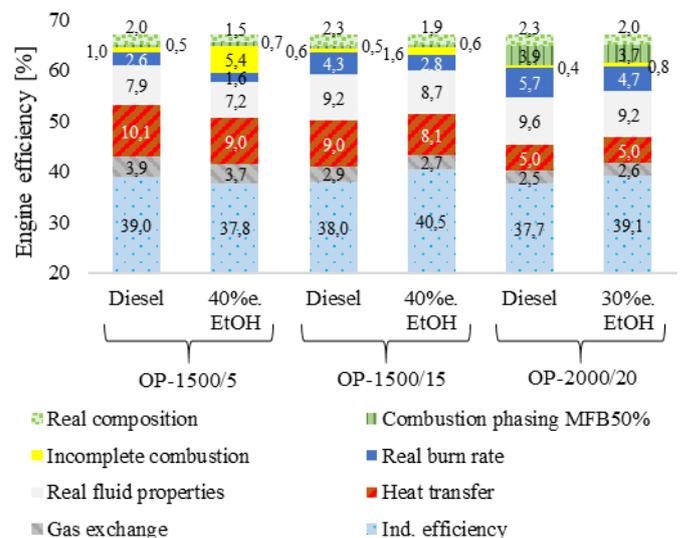


Fig. 18 Results of the efficiency loss analysis for three operation points. Comparison between “diesel only” operation mode and dual-fuel operation

It should be noted that the calculated values for wall heat transfer have to be interpreted with care when comparing stratified operation (as in pure diesel mode) and partially premixed operation (as in dual-fuel mode), as the wall heat transfer model used in the 0-dimensional approach (Woschni Classic) does not account for stratification effects. Therefore, wall heat loss for stratified operation tends to be overestimated in comparison to homogenous (or partly homogenous) operation.

9. Summary/Conclusions

The usage of ethanol and two different mixtures of ethanol and gasoline (E85 and E65) was investigated on a modified diesel engine designed to work in a dual-fuel combustion mode with intake manifold alcohol injection. The maximum ratio of alcohol to diesel fuel was limited by irregular combustion phenomena like degrading combustion quality and poor process controllability at low load and knock as well as auto-ignition at high load. At low load testing, up to 70 % of the diesel energy could be substituted by the alcoholic fuels, whereas at high loads the maximum substitution rate declined to approximately 30 %e. With rising alcohol amount, a significant reduction of soot mass and particle number was observed. At some testing points, substituting diesel with ethanol, E65 or E85 led to a reduction of NO_x emissions; however, the real benefit concerning the nitrogen oxides was introduced by the mitigation of the soot-NO_x trade-off. The indicated engine

efficiency was significantly improved with enhanced substitution ratios at high loads (by up to 6 %), whereas it dropped at low loads (by approx. 3 %). The combustion process tolerance against high EGR rates in dual-fuel mode allowed more exhaust gas to be recirculated without affecting engine efficiency the same negative way as in “diesel only” operation. The analysis of the indicated cylinder data revealed that at some high load conditions in dual-fuel mode, a cyclic reproducible auto-ignition of the premixed charge occurred. An engine-process simulation model showed that the high amounts of unburnt fuel at dual-fuel operation were the main reason for efficiency degradation, whereas the reduced losses due to real charge composition, real fluid properties, real burn rate and heat transfer enhanced the efficiency of the engine at dual-fuel mode. Substituting diesel with manifold injected alcoholic fuels reduced the engine CO₂ emissions at medium and high load operating points. The catalytic conversion of the high HC and CO emissions measured at low load testing would eventually lead to higher tailpipe CO₂ emissions at this operating point. A case study of the bioethanol production by AGRANA demonstrated fuel-related greenhouse gas saving potentials of up to 72 %. By using this bioethanol as a second fuel in the dual-fuel designed engine, the well-to-wheel CO₂ emissions could be cut down by almost 40 %. Conventional biofuels have the potential to contribute to the mitigation of the problems surrounding energy supply as well as food and feed production, provided that sustainability and socio-economic criteria are fulfilled and certified.

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11. Definitions/Abbreviations

°CA_aTDC	Crank angle degrees after top dead center
CEC	Certified test diesel fuel
DDGS	Distiller’s dried grain with solubles
E65	Mixture of 65 vol% ethanol and 35 vol% gasoline
E85	Mixture of 85 vol% ethanol and 15 vol% gasoline
EGR	Exhaust gas recirculation
EtOH	Ethanol
EU	European Union
FSN	Filter Smoke Number
GHG	Greenhouse gas
GMO	Genetically modified organism
IMEP	Indicated mean effective pressure
MFB	Mass fraction burned
OP	Operating point
SOI	Start of injection