

Optical and Thermodynamic Investigations of Reference Fuels for Future Combustion Systems

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ABSTRACT

The finite nature and instability of fossil fuel supply has led to an increasing and enduring investigation demand of alternative and regenerative fuels. An investigation program is carried out to explore the potential of tailor made fuels to reduce engine-out emissions while maintaining engine efficiency and an acceptable noise level. In this paper, fundamental results of the Diesel engine relevant combustion are presented. To enable optimum engine performance a range of different reference fuels have been investigated. The fundamental effects of different physical and chemical properties on emission formation and engine performance are investigated using a thermodynamic diesel single cylinder research engine and an optically-accessible combustion vessel.

Depending on the chain length and molecular structure, fuel compounds vary in cetane number, boiling temperature etc. Therefore, different hydrocarbons including n-heptane, n-dodecane, and l-decanol were investigated. To gain knowledge about the emission formation process with future fuels, engine experiments are performed at four part load conditions and full load. Investigations show that an increased boiling temperature leads to significantly higher soot formation.

In order to understand the effects of emission formation (in particular the higher soot formation of n-dodecane) an equivalent common-rail Diesel injection system is installed in a combustion vessel, which provides nearly quiescent high-pressure and high-temperature conditions. The combustion and soot formation are analyzed using two different optical measurement techniques. The hot reaction zone is visualized using OH* chemiluminescence imaging. Additionally, images of the integral natural soot luminosity of the flame are

acquired. The ignition delay, the flame penetration length and the unsteady lift-off length are determined. In particular, the results indicate that soot formation depends strongly on the fuel cetane number under the investigated part-load conditions.

In this integrated approach, investigations in a thermodynamic engine and an optically accessible combustion vessel offer the potential to analyze the effects of different fuel properties on the emission behavior in detail. The dependences of emissions formation on different fuel properties in a Diesel combustion system are identified and discussed. In particular, the impact of the cetane number, boiling temperature and oxygen content of the fuels is investigated. The relatively high soot formation of n-dodecane is reproduced in the optical vessel. The emission behavior can be explained by the overlap of the established flame and the liquid-phase fuel.

INTRODUCTION

Renewable biological fuels have the potential to permanently reduce greenhouse gas emissions. These fuels can be tailored to the demands of modern engines taking into account the catalytic production processes. With the establishment of these fuels an additional degree of freedom, a simultaneous optimization of the engine and fuel is possible.

The Cluster of Excellence “Tailor-Made Fuels from Biomass” (TMFB) adopts an interdisciplinary approach, through the application of optimized synthesis processes, towards research of new, biomass-based synthetic fuels, in order to verify their potential, with regard to modern combustion technologies, while simultaneously reducing the dependence on fossil fuels. The long-term goal is to determine the optimal combination of fuel components and

their production processes, which is based on renewable raw materials and new combustion processes.

Such a tailor-made fuel is a well-defined blend of distinct molecular components with optimized physicochemical properties for future combustion systems, which can be produced by sustainable and economical production processes. Starting from the combustion side, desired fuel properties for a low emissions combustion system have to be determined to establish a starting point for fuel production research. Therefore pure chemical compounds are investigated.

In the search for new fuels to reduce engine emissions several investigations have been recently published. Most of them see a positive impact on particulate emissions when decreasing the cetane number. A lower cetane number leads to a longer ignition delay which results in a better premixing. The lower constraint of the cetane number is an appropriate cold start behavior which needs to be guaranteed [1, 2, 3, 4].

Furthermore, different studies have shown the positive influence of oxygen content in the fuel on the emission behaviour. Soot formation can be reduced linearly with increasing oxygen content. Since the carbon atoms are partly bounded to the oxygen atoms, the formation of polycyclic aromatic hydrocarbons, which are precursors of soot particles, can be limited. Soot formation can be suppressed totally for fuels with oxygen contents higher than 38 wt-% [5,6]. The effectiveness of oxygenates at reducing soot is also discussed in detail in Reference [8].

It was demonstrated previously that soot formation in DI Diesel sprays depends strongly on the fuel/air stoichiometry at the lift-off location [7, 8, 9]. This was an important reason for further investigations on the stabilization of flame lift-off in References [10, 11, 12] and references therein. However, most of the previous work in this field focuses on quasi-stationary lifted Diesel flames, which are generated by sufficiently long duration of fuel injection. In contrast, relatively short injection duration and the resulting highly unsteady flames are investigated in the present work. It is demonstrated below that the dynamic features of fuel injection and flame lift-off are important for soot formation in this case. In particular, the influence of autoignition is studied. This is achieved by using three reference fuels (n-dodecane, n-heptane, and 1-decanol) with significantly different cetane numbers in consecutive experiments. Fuel effects on combustion and soot formation were previously investigated in References [1, 2, 3, 4, 5, 6, 8, 10, 11, 12, 13, 14] and the references therein. In general, it is expected that soot formation increases with increasing cetane number as noted previously, because the available time for fuel/air mixing between injection and autoignition, i.e. the ignition delay, decreases. This may lead to a richer fuel/air mixture at

the lift-off location. This mechanism is studied for the three selected reference fuels in the current work.

In addition, soot formation in unsteady and quasi-stationary Diesel combustion may depend on the cetane number of the fuel, because a strong influence of the cetane number on the location of flame lift-off, i.e. the lift-off length, was observed in previous work [10-11]. These results suggested that lift-off stabilization is basically determined by autoignition due to mixing with hot ambient air. Soot formation may depend on the lift-off length because the fuel/air mixture is generally expected to be richer towards the nozzle [7].

However, the influence of autoignition on lift-off stabilization is still under scientific debate. Recently the lift-off stabilization mechanism was studied by forcing ignition upstream of the natural lift-off length using a laser-induced plasma [15]. This experiment indicated that neither autoignition by mixing with ambient air nor flame propagation, as in the case of lifted atmospheric-condition flames [16], explains lift-off stabilization. Thus the authors suggested an alternative stabilization mechanism, which is based on mixing with hot combustion products at the jet edges [15]. This picture was supported by another experiment, which aimed at separating the effects of the chemical and physical ignition delays on flame lift-off, using different fuels with ignition enhancers [12]. In particular, no statistically significant correlation between ignition delay and lift-off length was observed. Thus, the relationship between ignition delay and lift-off length is investigated again in the present work by changing the fuel type and the temperature of the ambient air respectively. The variation of the ambient air temperature was also attempted in [12], but the expected influence on flame lift-off was not observed. The authors concluded that it was difficult to control the in-cylinder temperature in the used optical engine. In contrast, the present lift-off length and ignition delay measurements are conducted in an optically-accessible vessel, in which the ambient conditions of the common-rail jets can be controlled more accurately. The vessel provides high-pressure, high-temperature conditions up to 50 bar and 800 K [11].

Furthermore, it is often believed that the soot emissions of Diesel combustion depend on the penetration length of the flame, because this may affect the amount of oxygen available in the soot oxidation process [17,18]. Thus, the penetration length of the flames is also studied in the present work. Fuel effects on the penetration of the vapor phase of similar common-rail jets were previously investigated by using a backlight technique [19]. The results showed that the penetration of the vapor phase differed insignificantly for different fuels.

Lift-off length, ignition delay, and flame penetration length are measured by OH* chemiluminescence imaging at about 310 nm in the present work. This diagnostic was previously

used for determination of the lift-off length in [7, 8, 9, 10, 11, 12, 20]. The ignition delay was measured by OH* imaging in [20]. Excited-state OH* results from near-stoichiometric high-temperature combustion [21]. Thus, it is expected that the OH* images essentially show the same flame penetration length as the vapor phase measurements by backlight illumination mentioned above, as soon as the diffusion flame in the periphery of the spray is fully established [19]. Soot formation in the flame is detected by luminosity imaging in the visible wavelength range in the present work [22].

The results of the optical measurements are compared to corresponding soot emission and ignition delay data of a thermodynamic single-cylinder research Diesel engine, as noted previously. Essentially the same fuel injection system is used for the engine and the measurements in the vessel.

EXPERIMENTAL SET-UP

SINGLE CYLINDER DIESEL ENGINE

The engine used in this study was optimized to achieve best combustion behavior for lowest engine-out emissions and highest fuel efficiency. The hardware enhancements include high fuel injection pressures for improved mixture formation, high EGR levels and intensified charge air and EGR cooling to lower the temperature in the combustion chamber for a cooler combustion. Details of the test engine are listed in [Table 1](#). Further information about the research single cylinder test engine are available in different publications [23, 24].

The research engine is capable of achieving Euro 6 emissions performance by means of a lower compression ratio, a higher maximum cylinder peak pressure, a maximum rail pressure of 2000 bar and intensified EGR cooling. These engine enhancements are expected to be commonly used on advanced diesel engines in the near future. As such, the testing capabilities on the engine exceed those of today's production engines.

ENGINE OPERATION

Engine tests for the fuel investigations are carried out in four part-load points, as shown in [Figure 1](#). Three of the four part-load points are located within the reference area of the New European Driving Cycle (NEDC) while the highest load point at 2400 rpm and 14.8 bar IMEP is outside this range. Based on the new European Driving Cycle (NEDC) the maximum Indicated Specific NO_x (ISNO_x) and Indicated Specific PM (ISPM) emissions were estimated for each part-load point and different Euro emission levels for a 1590 kg vehicle. These targets take into account the expected impact of transient operation and cold start and include an appropriate engineering margin. The estimated NO_x and PM target levels are summarized in [Table 2](#) and [Table 3](#). Investigations on a chassis dynamometer confirmed these estimated targets.

All experiments are carried out without pilot injection and simulated closed loop combustion control. This is done by adjusting the fuel injection timing on each fuel so that the center of combustion (CA50) is always located at the same degree crank angle in the engine cycle, chosen to maximize the engine efficiency, regardless of the fuel properties. Fuels with lower cetane number (CN) and longer ignition delay require earlier fuel injection timing to maintain the centre of combustion at the required crank angle position. The difference between the energizing time of the injector and the begin of injection is measured to 160 µs. The conditions at the four part-load operating points are chosen with respect to today's achievable ranges and are listed in [Table 4](#). The intake manifold temperature is measured and controlled directly after the intercooler and before the EGR mixing.

FUEL PROPERTIES

The investigated, single component fuels are chosen to describe the influence of different fuel properties on engine out emissions and efficiency for a conventional compression ignition engine. One main target is to understand the effects of soot formation in a Diesel engine and to analyze the potential to reduced soot emissions using fuels with optimized properties. Three fuel parameters are identified to be most important, thus the chosen starting fuels differ in cetane number, boiling temperature and oxygen content. Namely the fuels are n-heptane, n-dodecane and 1-decanol. As a basis standard EN590 Diesel fuel is also investigated. The corresponding fuel properties are listed below in [Table 5](#).

Probably the most important parameter associated with diesel combustion is a fuel's autoignition characteristics. This autoignition behaviour is responsible for the compression ignition of the usually heterogeneous fuel/air mixture in diesel engines. For diesel applications the ignition delay of one specific fuel is described by the cetane number. A longer ignition delay (decreased cetane number) results in a better mixture formation and leads to an increasing fraction of premixed combustion in the whole energy conversion process. Generally the cetane number of hydrocarbons decreases with increasing molecular stability. Therefore it gets higher with increasing molecular size and decreases for double bonds and cyclic structures.

The boiling curve influences the mixture formation in the combustion chamber. Generally the boiling temperature increases with growing number of C-atoms. Lower boiling points lead to faster evaporation and as a result to a faster mixture formation, which leads to a lower soot formation. The volatility and the cetane number of the investigated fuels are illustrated in [Figure 2](#).

Generally variations in one fuel property (like cetane number) also influence other fuel parameters (like volatility). To separate the effect of a single fuel parameter on the

combustion behavior a comparable molecular composition is beneficial. Therefore the three investigated fuels are mainly characterized by linear-chain hydrocarbons with saturated single bonds only. The lack of double bonds and cyclical structures leads to a uniform reaction mechanism during the combustion. N-heptane and n-dodecane are normal paraffins (n-paraffins) while 1-decanol is a long chain alcoholic hydrocarbon.

Three different reference fuels are investigated as noted previously for the optical experiments. N-dodecane is an alkane often used as a model fuel for Diesel combustion. Furthermore n-heptane is added as an alkane with a shorter chain length and a reduced, more Diesel-like cetane number. Also an alcohol (1-decanol) which represents an oxygen-containing Diesel-like reference fuel is investigated. Furthermore, a fuel blend which contains 50 volume percent n-heptane and 50 volume percent 1-decanol is investigated.

SET-UP OF THE OPTICAL EXPERIMENTS

The optical measurements are conducted in a high-pressure vessel with a constant air flow (20 m³/h). The ambient conditions inside the chamber are adjusted to a temperature of 800 K and 750 K, and a pressure of 50 bar and 43.8 bar, respectively, so that the density is kept constant. The optical access of the vessel is achieved using quartz windows as shown in [Figure 3](#). The vessel is also described in detail in [\[20\]](#).

The used piezo injector is equipped with a three-hole nozzle with an orifice diameter of 99 µm. All measurements are conducted with a fuel rail pressure of 900 bar. The energizing time (ET) of the injector is set to 400 µs.

For the measurement of the OH* chemiluminescence an ICCD camera (NanoStar, LaVision) is used (see [Figure 3](#)). An UV lens (B. Halle, $f = 100\text{mm}$, $f\# = 2$) and a filter combination (BP313, MSO Jena and DUG11X, Schott) with a maximum transmission around 313 nm is used. The filter combination isolates the OH* chemiluminescence in the range of 290 nm - 325 nm from the emissions of the soot and other species [\[11,25\]](#). The exposure time of the camera, which detects the OH* chemiluminescence, is 10 µs.

For the measurements of the integral natural luminosity a digital single-lens reflex camera is used. A zoom lens (Nikon, $f = 18\text{--}55\text{ mm}$, $f\# = 11$) is mounted on the camera to record the natural luminosity in the entire visible range. The gate length is set to 1 s to ensure that the complete combustion process is measured.

DATA REDUCTION PROCEDURE OF THE OPTICAL EXPERIMENTS

The lift-off length is determined using a weighted profile of the raw data image and a relative intensity threshold. The procedure is described in more detail in [Reference \[11\]](#). The same method is used to calculate the penetration length of the flame. Thus, the lift-off length corresponds to the lower edge of the visualized hot reaction zone and the penetration length to the upper edge, respectively.

Due to the assumption that the ensemble-averaged 3-d OH* distributions are cylindrically symmetric, an Abel inversion method is implemented to get 2-d OH* distributions in the central plane. The used algorithm is a modified discretization method presented in [References \[26,27\]](#).

The ignition probability (not shown in this work) is determined using the statistical analysis of the number of single-shot measurements with a significant OH* chemiluminescence signal [\[20\]](#). The ignition delay is defined as the time after start of actual fuel injection when the ignition probability has reached 50%.

RESULTS

The investigations focus on the emission performance of different hydrocarbons under engine like conditions. To gain knowledge about the emission formation process of fuels having different properties, engine experiments are performed at four part load conditions in the first phase. In order to understand the fundamental effects of emission formation (in particular soot formation) investigations in an optical combustion vessel are performed in the second phase. Besides the analysis of the cetane number and the resulting ignition delay also the impact of oxygen content on emission formation is investigated within this paper.

THERMODYNAMIC SINGLE CYLINDER DIESEL ENGINE

A practical engine must provide acceptable performance over the entire operating range. Even though the properties of the test fuels varied, the engine was able to match the full-load power achieved on the Diesel for all test fuels, within the constraints imposed by smoke level and exhaust temperature, and with very little impact on engine efficiency or engine noise. However, attention naturally focuses on the part-load test conditions that influence performance in the NEDC emissions cycle. For this reason, this section is largely devoted to an assessment of these part-load conditions. All results presented in this paper are illustrated at the Euro 6 NOx emission limit.

As described above, all engine parameters including rail pressure and center of combustion were kept constant to

maintain equal conditions. However, n-heptane with its low viscosity could not achieve the same high fuel injection rail pressure at the highest load point as the other fuels due to leakages inside the high pressure pump. But this did not significantly impact the performance of the engine.

The ignition delay time for the single cylinder investigations is defined as the time between the start of the main fuel injection to the point where 5 % of the injected fuel quantity is converted. The ignition delay time in an engine can be usually separated into a chemical ignition delay (time for first pre-reactions) and into a physical ignition delay (time for fuel evaporation, allocation of an ignitable mixture preparation). In a first approximation the chemical and the physical ignition delay can be summed up to the entire delay time. The cetane number describes the ignition delay time only for one defined operating point. Therefore it is not adequate to draw conclusions on emission formation without the knowledge about the influence of the cetane number on the ignition delay time. In [Figure 4](#) the ignition delay time of the three single component fuels and standard Diesel fuel is illustrated for the different load points. The cetane number differs from 50 (1-decanol) to 80 (n-dodecane).

The influence of the cetane number on the ignition delay decreases with higher engine loads. The maximum of the ignition delay is located at low cetane numbers at the lowest load point. The shortest ignition delay is located at the highest load point and the highest cetane number. Pressures and temperatures are much higher in this area. Furthermore, the influence of the cetane number on the ignition delay is not linear but increasing with lower cetane numbers. The effect of the cetane number is reduced with increasing load and therefore more determining for lower loads.

In conventional compression-ignition engines, a characteristic 'NOx/PM trade-off curve' is typically observed, where PM-emissions increase as EGR is increased in order to reduce NOx emissions. Within these investigations, this type of behaviour is seen for all three hydrocarbons at the higher part-load conditions, although the advanced engine design leads to significant increases in PM-emissions only below 0.5-0.7 g/kWh ISNOx. The indicated specific particulate emissions of the fuels differ from each other. At the lowest load point (1500 rpm, 4.3 bar IMEP) PM-emissions remained very low regardless of the NOx level. At the 1500 rpm, 6.8 bar IMEP part-load point, the same behavior is seen for n-heptane and 1-decanol, but regular diesel fuel and n-dodecane show a more typical NOx/PM trade-off. At 2280 rpm, 9.4 bar IMEP all tested fuels converted to the typical diesel NOx/PM trade-off curve, as illustrated in [Figure 5](#).

Due to the aromatic content diesel fuel leads to higher soot emissions, in comparison to n-dodecane. The aromatic and polycyclic aromatic species in the fuel are already precursors for particulate matter emissions. Generally, particulate matter

emissions increase with increasing cetane number. This is most likely caused by the reduced ignition delay time, which results in a disadvantageous mixture preparation and thereby in larger areas of locally rich air/fuel ratios. These rich areas cause soot formation during the combustion process. This general trend can be confirmed when n-heptane and n-dodecane results are compared. As described above, fuel properties are always depending on the molecular structure and therefore they influence each other. Generally, an increase of the molecular chain length leads to a higher boiling temperature due to an increased molecular weight. In consequence these fuels show a longer physical ignition delay even with increased cetane numbers. Thus, increased soot formation of n-dodecane in comparison to n-heptane can also be explained by a later evaporation due to the higher boiling temperature and higher heat of vaporization.

Since 1-decanol shows the most promising NOx/PM-trade-off curve the alcohol-group (OH-group) has to be responsible for this benefit. Even the negative influence of the higher boiling temperature in comparison to n-heptane which inhibits the mixture preparation, can be compensated. To explain the underlying process the molecular combustion chemistry has to be considered. The carbon atoms which are already bonded to the alcohol-group cannot provide soot precursors due to the high energy of formation between the carbon and the oxygen atoms. As [Figure 5](#) indicates this process is responsible for a soot emission benefit of 90 % when using 1-decanol in comparison to diesel fuel at the Euro 6 ISNOx-emission level at $n = 2280$ rpm and IMEP = 9.4 bar.

Fuel properties have some effect on HC-emissions although the effects are generally much smaller than for the PM emissions. At this high load point no significant differences can be observed within HC and CO emissions. Thus, the trends are also illustrated in [Figure 6](#) for $n = 1500$ rpm and IMEP = 6.8 bar. HC-emissions (ISHC) decrease with an increasing cetane number. In comparison to all other hydrocarbons n-dodecane shows less HC-emissions. The shorter ignition delay leads to a higher fraction of complete combustion, because the propagation of lean areas, which are outside of the flammability limits, can be limited. The significantly higher HC-emissions at smaller loads are caused by incomplete combustion due to quench effects. Generally, the formation of CO-emissions (ISCO) is based on a similar mechanism like the formation of HC-emissions. Therefore, it is not surprising that the CO-emission trends are broadly similar to the HC-emission trends.

For diesel engine applications, the engine noise is a critical limit with tightened constraints. The engine noise is characterized through the combustion sound level (CSL). CSL is a method for the prediction of engine radiated noise based on the analysis of cylinder pressure data, using an average of 50 cycles, obtained from the thermodynamic test bench measurements. The CSL value combines cylinder

pressure analysis with appropriate engine structure weighting functions for direct and indirect combustion noise as well as mechanical and flow noise. The exact methodology is described in [28].

Generally for all different part-load points the noise emissions of the tested fuels are within in the Diesel range. Fuels with a reduced ignition delay (n-dodecane) show decreased noise emissions due to a reduced part of premixed combustion. Fuels with reduced cetane number usually show higher noise emissions (1-decanol). This behavior can be observed in [Figure 6](#) and [Figure 7](#). Investigations were carried out without pilot injection.

Fuel consumption is measured and the engine efficiency is calculated for each fuel taking into account the different heating value. At constant engine load, indicated efficiency is for all tested fuels nearly constant. Only some small differences among the different investigated fuels without consistent trend can be seen. When comparing individual fuels, efficiency measurements tend to show an increase at some conditions and a decrease at others. None of the investigated fuels offers a particular advantage with respect to efficiency at any part-load condition point in this engine, when keeping the NO_x emissions constant.

OPTICAL VESSEL

Typical temporal sequences of ensemble-averaged OH* chemiluminescence images are given for n-dodecane, n-heptane, and 1-decanol, respectively, in the first three rows of [Figure 8](#). In addition, the corresponding results of a blend, i.e. 50% n-heptane and 50% 1-decanol by volume, are shown in the fourth row of [Figure 8](#), because future tailor-made fuels will be blends in most cases. The nozzle hole is located at a height of 0 mm in the images of [Figure 8](#). The liquid fuel jet is visible in the first two images of the first row, i.e. for n-dodecane, because it is illuminated by the flame. The fuel injection timing is also investigated by shadowgraphy (not shown). Essentially, fuel injection actually starts at a t_{aei} of 0.2 ms (at the nozzle exit), and it ends at 1.5 ms for the used injection system. The first row of images in [Figure 8](#) shows that the high-temperature flame is fully established well before the end of injection for n-dodecane. Accordingly, the ignition probability is found to be 100% after a t_{aei} of 1.2 ms for n-dodecane. In contrast, the lower three rows of images in [Figure 8](#) demonstrate that the first high-temperature reactions are detected only a short time before the end of injection for the other fuels. In fact an ignition probability of 100% is approached around a t_{aei} of 1.5 ms for these fuels, but slightly earlier for n-heptane as compared to 1-decanol. Accordingly, the ignition delay, which is determined from the OH* images as described in the last section, depends strongly on the cetane number, CN, as shown in [Figure 9](#). The ignition delay is given with regard to the actual start of fuel injection in [Figure 9](#). This diagram contains the results of two

measurements for n-heptane and n-dodecane with a T_A of 750 K and 800 K, respectively, in the vessel. The air density is equal in both measurements. A larger temperature variation can hardly be investigated at equal density, since no high-temperature reactions are detected for 1-decanol at 750 K, and at 700 K, respectively, for the other three fuels. However, [Figure 9](#) shows that the ignition delay depends on T_A for n-heptane and n-dodecane.

[Figure 8](#) shows that the Diesel-like flames are highly unsteady, due to the relatively short duration of fuel injection. Thus, the temporal behavior of the lift-off length and its pulse-to-pulse variability are investigated in the following. [Figure 10 a\)](#) shows that the ensemble averaged lift-off lengths are indeed unsteady. The lift-off length of the first three fuels are significantly different in the early phase, i.e. before a t_{aei} of 2 ms, whereas they are rather similar for later t_{aei} . The behavior of 1-decanol and the blend are very similar. However, all curves have a minimum in the early phase, which is presented in [Figure 9](#) for the first three fuels. Obviously, the minimum lift-off length depends strongly on CN and the temperature of the surrounding air. This indicates again that there is a strong influence of autoignition on the combustion process, at least in the early phase. The coefficient of variation (cov) in the lift-off length is shown in [Figure 10 b\)](#). Apparently, it depends on CN in the very early phase, i.e. before a t_{aei} of 1.7ms. The curves of 1-decanol and the blend are again very similar. These two curves and the one of n-heptane depend significantly on t_{aei} , whereas the one of n-dodecane is approximately constant with a cov of about 10%. Essentially the same variability of about 10% was found previously for comparable quasi-steady sprays using n-decane as the fuel [20].

[Figure 10 a\)](#) shows that the temporal overlap of fuel injection and combustion is significantly larger for n-dodecane than for the other fuels, as noted previously. In addition, the first two images of the first row in [Figure 8](#) indicate that there is also a spatial overlap of injection and combustion for n-dodecane, because there is no gap between the liquid fuel jet and the flame. Accordingly, the quasi-steady penetration length of the liquid fuel was determined to be about 19 mm for n-dodecane, and it is reached about 0.3 ms after start of injection [19]. A very similar injection system and approximately the same ambient conditions were used in [Reference \[19\]](#). Thus, [Figure 10 a\)](#) illustrates that the early lift-off length is indeed shorter than the liquid length for n-dodecane. The liquid length of the n-heptane jet is not readily available, but it can be estimated that it is shorter than that of n-dodecane, because the boiling temperature and heat of vaporization are significantly lower for n-heptane [29,30]. These thermodynamic fuel properties are generally particularly important for the liquid length as demonstrated in [Reference \[31\]](#). Thus, it can be assumed that there is no spatial overlap of the liquid jet and the flame based on the