

Mixture Formation and Combustion of Biodiesel Blended Fuels in a DI Diesel Engine

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Abstract

Mixture formation and combustion of biodiesel blended fuels in an optical accessible DI Diesel engine are studied. Optical experiments and a thermodynamic analysis have been performed on pure biodiesel, diesel and three mixtures. It is shown how spray formation and propagation as well as ignition delay and combustion are influenced by various properties of the different fuel mixtures.

Introduction

Biodiesel and its blends seem to be promising completions on a way to a modern energy mix in the traffic sector. Several groups have evaluated effects of biodiesel on exhaust gas emissions [1–5]. However not much is known about in-cylinder behaviour of such fuels. Apart from the higher cetane number of biodiesel some other crucial properties for spray formation and propagation, like viscosity and surface tension, considerably differ from those of diesel fuel. This has not only an impact when biodiesel as a pure fuel is used but also for all kinds of mixtures containing significant amounts of biodiesel. Therefore it is of interest to have a closer look on in-cylinder mixture formation and combustion of biodiesel blends in comparison to pure biodiesel and pure mineral diesel.

Since the problem of chemical stability of the sealing of the fuel pump and other peripheral parts has been solved, it is possible to use biodiesel in a larger scale. In order to use biodiesel as fully developed complement to the fuel mix of the traffic sector it is indispensable to know and understand the effect of the usage of biodiesel not only on emissions but also on in-cylinder mixture formation and combustion. Only few information concerning this topic has been published yet [6, 7]. However the different physical properties of both fuels lead to the assumption that they have implications on spray propagation and therefore on the mixture formation. For illustration the most crucial factors on spray propagation and ignition for pure biodiesel and diesel according to the european law standards are depicted in the following Table 1. For surface tension is another important property that is not regulated in european law is was measured and included into the table.

Table 1. Properties of biodiesel and diesel [8, 9]

Property	Diesel	Biodiesel
Cetane number	51,0	54 – 58
Density	0.820 – 0.845 kg/l	0.875 – 0.890 kg/l
Flashpoint	> 55 °C	100 – 135 °C
Energy density	11 – 12 kWh/l	8.9 kWh/l
Viscosity	2.00 – 4.50 mm ² /s	7 – 8 mm ² /s (20 °C)
Surface tension ¹	24 - 25 mN/m	28 - 29 mN/m

Particular attention should be laid on the differences in viscosity and surface tension as well as the cetane number. The former two leading to a greater loss of momentum at the nozzle tip and larger droplets during atomization the last mentioned having great influence on the ignition delay and timing. Therefore investigations like the presented ones give the capability for a deeper understanding of the mechanisms and phenomena of diesel combustion itself because they allow to draw conclusions about the impact of some physical properties on mixture formation and the subsequent combustion.

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Materials and Methods

This study was performed on pure RME biodiesel, diesel and three mixtures containing 25, 50, 75 volume percent biodiesel mixed with mineral diesel. The mixtures are named after their content of biodiesel so that B75 means 75 volume percent of biodiesel and 25 volume percent of diesel.

The experiments were carried out in an optical accessible DI diesel engine. The engine had two side windows and one optical access via the transparent omega bowl. The main data of the engine is collected in Table 2.

Table 2: Specifications of the DI Diesel engine

Bore	85 mm
Stroke	85 mm
Injector	6 hole solenoid
ϵ	14
Bowl type	Omega bowl
Number of inlet valves	2
Number of outlet valves	1

A solenoid injector with a six hole nozzle was mounted to the engine. The spray angle of this nozzle was 135° . The temperature of the inlet air was set to 25°C with a boost pressure of 1.2 bar. The temperature of the cooling circuit as well as the oil temperature were kept constant at 80°C . For the experiments several operating points were investigated. Thereby the engine speed was set to 500, 750, and 1000 rpm. The start of injection was varied from -12°CA to 0°CA to top dead center (TDC) and the injection pressure was varied from 600 to 1300 bar. The injected volume was 12.9 mm^3 . Injecting a constant volume lead to slightly different injected masses and maximum available heats of combustion for each fuel. Figure 1 shows the injected mass as well as the theoretical heat of combustion as a function of biodiesel volume fraction.

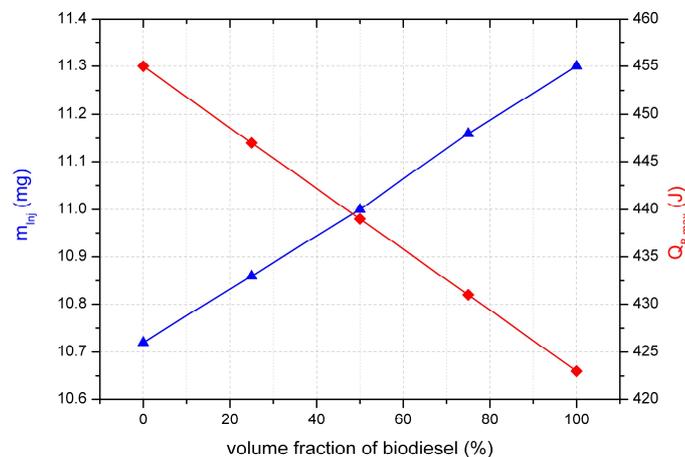


Figure 1. Injected mass and maximum available heat of combustion as a function of the volume fraction of biodiesel in the fuel mixture

Mie scattering of fuel droplets was combined with a high-speed camera to observe the complete spray propagation and flame luminescence of individual cycles. A continuous Nd:YVO₄ laser with an emission wavelength of 532 nm and 5 W output power served as a light source for the illumination of the injected spray plumes and the generation of the Mie signal. The laser was expanded by a telescope to illuminate the hole cylinder and coupled into the cylinder through a side window. A high speed camera with a recording rate of 15.000 fps detected the elastic scattered straylight of the spray and the flame and soot luminescence of the combustion. The resolution of this camera was 256x256 pixels.

In addition to observation of mixture formation and combustion, a thermodynamic analysis of the pressure traces was performed leading to histories of heat release and burnt mass fraction.

Results and Discussion

For each operating point the pressure trace and the high speed movies were used to derive the heat release per crank angle and the visible start of injection (VSOI). Due to relatively low standard deviation of below 1.0 bar in peak pressure the averaged pressure traces were calculated out of 10 consecutive cycles. Figure 2 shows the averaged pressure traces for all five mixtures at 500 rpm and an injection start at -9°CA .

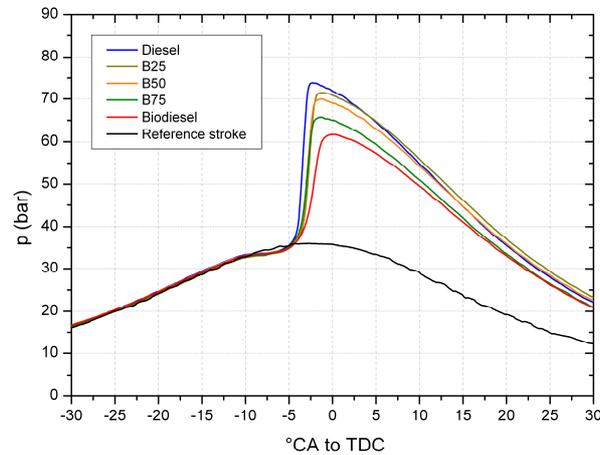


Figure 2. Pressure traces of the five fuel mixtures at 500 rpm and $t_{inj} = -9^\circ\text{CA}$

From the pressure traces it can be seen that the higher the amount of diesel in the mixture the earlier the pressure rises and the combustion starts. That is surprising according to the fact that the cetane number of biodiesel is slightly higher than that of diesel what should lead to an earlier ignition of the biodiesel containing fuels (see Table 1). The average heat release rates, that were derived from the pressure traces, confirm with that observation. These heat release rates were accumulated what lead to a total heat production that is shown in Figure 3 for the same operating conditions as in Figure 2.

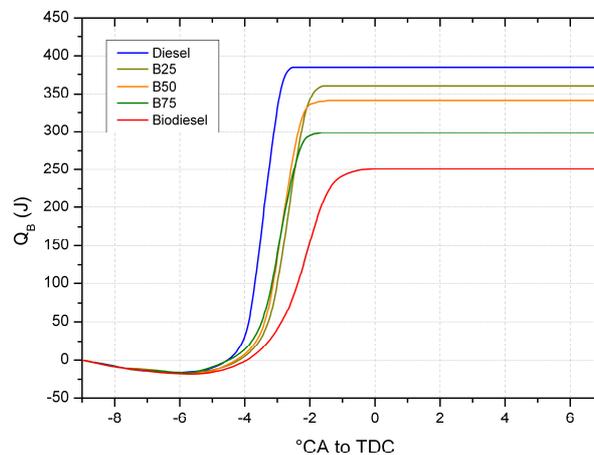


Figure 3. Integrated heat of combustion for all five fuel mixtures at 500 rpm and $t_{inj} = -9^\circ\text{CA}$

Figure 3 shows that the total heat production increases according to the diesel fraction of the fuel mixtures. Additionally it can be seen that the fuels with higher biodiesel fraction need more time to ignite which leads to a later heat production. However a direct comparison between the total heat production of each fuel is not valid because of their different maximum available heat of combustion (see Figure 1). In order to obtain a good criteria for comparing the heat release of the fuels the total heat production was normalized by this theoretical heat of combustion which lead to fuel conversion which is a quantity that is proportional to the burnt mass fraction of the fuels. Figure 4 shows the conversion rates of the fuels as a function of the engine speed at an injection timing of -9°CA .

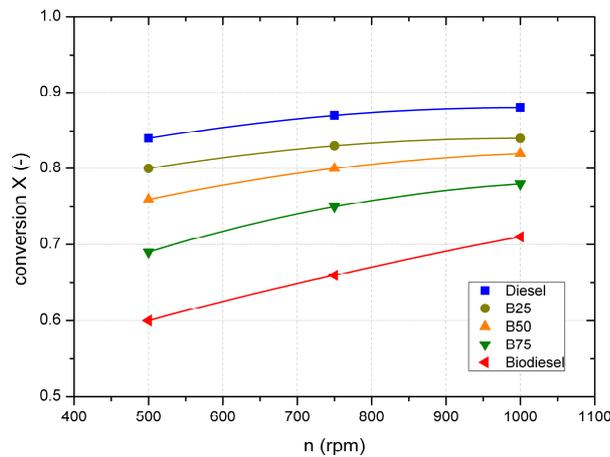


Figure 4. Fuel conversion as a function of engine speed at $t_{inj} = -9$ °CA

All fuels show an increase in conversion rate with increasing engine speed. The highest conversion rates were measured for pure diesel and the lowest ones for biodiesel. The conversion rates of the mixtures lay in between the ones of the pure fuels according to their biodiesel fraction. All the curves get steeper for higher biodiesel content. Thereby the conversion rate for the diesel is rising slightly from 0.84 to 0.88 whereas the conversion of biodiesel is rising more drastically from 0.6 to 0.71. That is a relative rise in conversion of 5% for diesel and 8.5% for biodiesel. Obviously the higher the biodiesel fraction the more does fuel conversion rise with increasing engine speed. A possible explanation is that the heat loss through the cylinder walls decreases with higher engine speed. This effect is more important in optical engines with inherent higher heat losses than standard engines. So an increase in engine speed in the optical engine is in fact an increase in cylinder load temperature which leads to better evaporation.

The fuel conversion as a function of injection timing at 500 rpm is illustrated in Figure 5. As with rising engine speed the fuel conversion also rises when the start of injection is shifted from -12 to 0 °CA. The diesel shows again the highest conversion rates followed by the mixtures lined up according to their biodiesel fraction. Biodiesel has the lowest fuel conversion at every timing. The rise in conversion of diesel is from 0.8 to 0.91 which is a relative increase of 13%. Biodiesel has again the highest relative rise of 29% ranging from 0.5 to 0.71. Apparently fuels with higher biodiesel fraction profit more from a shift in injection timing towards TDC as Diesel and fuels with low biodiesel fraction. The reason for that is, as was the reason when engine speed was varied, that a shift of the injection timing towards TDC means higher temperature and higher pressure e.g. air density at the injection. Both factors are leading to a better spray desintegration and faster evaporation.

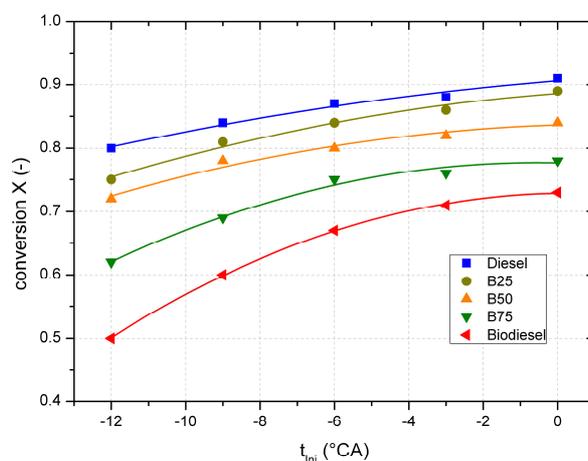


Figure 5. Fuel conversion as a function of injection timing at 500 rpm

However one important result of both variations is that the more biodiesel a fuel is containing the lower is its conversion rate and the higher is the ignition delay. This is completely against the expectations from a preconsideration of the cetane numbers. One possible explanation is that the physical ignition delay is much higher for fuels with high biodiesel fractions compared to diesel and fuels with low fractions. In order to investigate this spray penetration was evaluated in a spray chamber and compared to the Mie-images recorded with the high speed camera system at the engine. Figure 6 shows the results of the measurement of the penetration depth in the spray chamber. The parameters within the chamber were set to conditions inside the engine for an injection timing at -3°CA e.g. evaporating conditions. In addition to that the Mie-images recorded at the engine are shown in Figure 7.

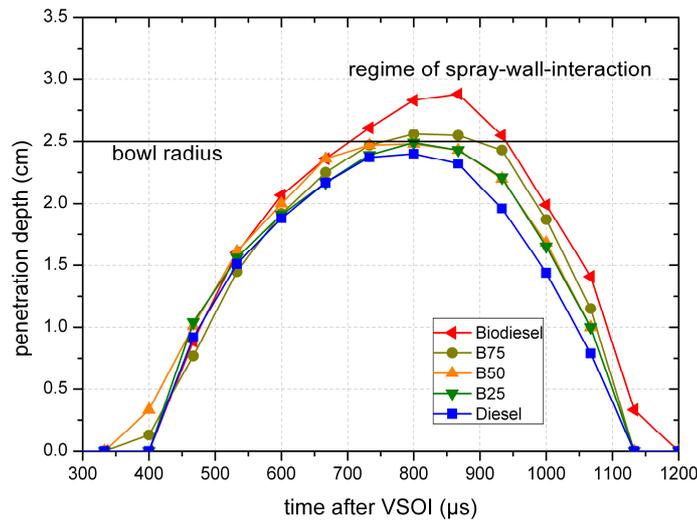


Figure 6. Penetration depth of all five fuel mixtures at $p_{inj} = 600$ bar with regime of spray-wall-interaction (black line at 2.5 cm marks the bowl radius)

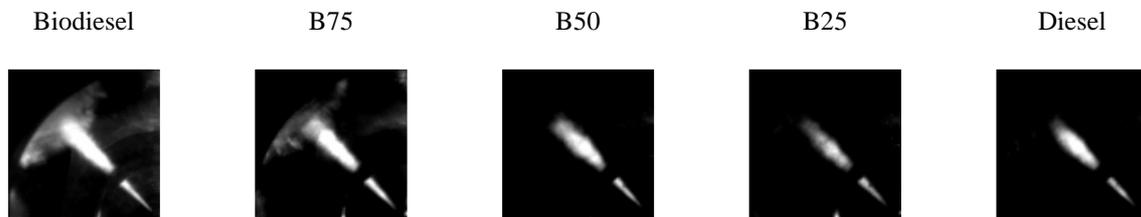


Figure 7. Mie scattering images of the spray-wall-contact for all five fuels

The penetration depths printed in Figure 6 show that biodiesel has by far the highest penetration depths followed by B75 and the other mixtures. Diesel has the smallest penetration depths. The difference between the maxima in penetration depths is about 7 mm for diesel and biodiesel. The black line at 2.5 cm marks the radius of the bowl. In the region above that line spray-wall-interaction occurs. The diagram indicates that the biodiesel and B75 sprays have spray-wall-contact when injected under the chosen conditions. A positive proof of the diagram can be drawn out of the Mie-images where each fuel spray is shown at its maximum penetration depth within the engine. It can be clearly seen that the biodiesel as well as the B75 spray have wall-contact. This wall-contact leads to a wetting of the cylinder bowl, retarded and uncomplete evaporation and therefore to higher physical ignition delays and lower fuel conversion rates.

Summary

The experiments showed that all fuels containing biodiesel have significantly higher penetration depths. It can be concluded that biodiesel causes poor spray disintegration with large droplets. This leads to a retarded

evaporation and therefore mixture formation with a higher physical ignition delay. The fuel blends with high concentrations of biodiesel (pure RME, 75%) show such high penetration depths that in the given engine spray-wall contact occurs (see Fig. 6 and Fig. 7). This leads to severe wetting of the piston bowl and a characteristic soot deposition.

These observations match the result of the thermodynamic analysis that showed reduced conversion rates for the fuels with high biodiesel content and longer ignition delays due to poor evaporation. Parameter variations revealed that biodiesel blended fuels benefit much more from hotter cylinder load.

Acknowledgement

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