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The impact of spray quality on the combustion of a viscous biofuel in a micro gas turbine

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HIGHLIGHTS

- Combustion of straight vegetable oil was tested in a micro gas turbine.
- The study is focused on the effect of fuel viscosity on combustion efficiency.
- A linear relation between viscosity and CO emissions was found.
- The pressure-swirl atomizer showed limited tolerance to increased viscosity.

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ABSTRACT

The relation between spray quality and combustion performance in a micro gas turbine has been studied by burning a viscous biofuel at different fuel injection conditions. Emissions from the combustion of a viscous mixture of straight vegetable oils have been compared to reference measurements with diesel No. 2.

The effect of fuel viscosity on pollutant emissions is determined by adjusting the injection temperature. The measurements confirm that a reduction in fuel viscosity improves the spray quality, resulting in faster droplet evaporation and more complete combustion. CO emission levels were observed to decrease linearly with viscosity in the tested range. For the pressure-swirl nozzle used in the tests, the upper viscosity limit is found to be 9 cP. Above this value, droplet evaporation seems to be incomplete as the exhaust gas contains a considerable amount of unburned fuel.

Additionally, the influence of increased injection pressure and combustor temperature is evaluated by varying the load. Adding more load resulted in improved combustion when burning diesel. In case of vegetable oil, however, this trend is less consistent as the decrease in CO emissions is not observed over the full load range.

The outcome of this study gives directions for the application of pyrolysis oil in gas turbines, a more advanced biofuel with high viscosity.

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1. Introduction

1.1. Background

Depletion of fossil resources and concerns about the environment have led to increased attention to energy production from biofuels. Although various renewable fuels and conversion technologies have been developed to address these issues, gas turbines can play a major role in the transition to a more sustainable energy supply. Gas turbines offer some important advantages over other technologies, including high power-to-weight ratio, high reliability, high flexibility and low pollutant emissions [1]. Furthermore, the scalability of gas turbines allows for the production of heat and power in a decentralized manner. With overall energy efficiencies above 80% in cogeneration plants, this technology is considered to be promising for distributed applications [2,3].

When it comes to application of liquid biofuels, the relatively robust burning characteristics in gas turbines is of particular interest. While reciprocating engines are very sensitive to problems such as clogging and delayed ignition, gas turbines have shown good potential to cope with alternative fuels. Experimental studies have shown that most common biofuels, such as bioalcohols and biodiesel, can already be burned in pure form in standard or slightly modified combustor designs without any significant problems [4–9].

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Nevertheless, with the eye on the environment and sustainability, it is more interesting to focus on second generation biofuels derived from biomass residues. In this class of biofuels, fast pyrolysis oil is considered as a promising example [1,10–12]. Pyrolysis oil, also referred to as pyrolysis liquid or bio-oil, is produced by thermal decomposition of biomass. By quenching of the condensible vapors formed in the cracking process, a combustible liquid is obtained containing a wide variety of chemical species.

Due to the rather low feedstock quality and the random nature of the decomposition reactions, the properties of pyrolysis oil are very different from those of conventional fossil fuels. For this reason, pyrolysis oil is not directly suitable for use in combustion devices, including gas turbines. The few attempts that have been made revealed major problems as incomplete combustion, flame instability and coke formation on combustor walls [4,5,13–16]. Therefore, it is generally concluded that fuel blending or thorough modification of the gas turbine is required for successful application.

The poor combustion behavior of pyrolysis oil is partly a result of its unfavorable chemical properties. For instance, the low heating value and the presence of non-volatile matter slow down the mass and heat transfer inside a combustion chamber considerably. Researchers have been trying to improve the quality of pyrolysis oil for several decades by optimizing the production process or upgrading the raw product [10,17,18]. Unfortunately, these studies did not yet result in a feasible method for obtaining pyrolysis oil with properties similar to fossil fuels. For this reason, more research has been initiated with a focus on adapting existing combustion devices to operate on pyrolysis oil in its present form.

In the development of such fuel flexible combustion systems, it is of crucial importance to look into the influence of fuel properties on the spray quality. The atomization process governs the evaporation and distribution of fuel inside the combustion chamber and is therefore closely related to the combustion performance [19]. In case of pyrolysis oil, most of the conventional atomization techniques tend to deliver a fuel spray of poor quality. This issue is mainly attributed to the high viscosity of the oil, which has a dampening effect on the fuel breakup mechanism [20,21]. Since the time required for complete evaporation is considerably higher when the fuel is poorly atomized, some fraction of the fuel may impinge on the liner. The increased evaporation time can be a major cause for the aforementioned problems that were encountered in pyrolysis oil burning tests.

Aside from the effect on the evaporation process, atomization also has a significant influence on the combustion kinetics of pyrolysis oil. Due to the chemical instability of this biofuel, polymerization reactions at elevated temperatures lead to the formation of non-volatile matter inside the droplets during their flight in the combustor [15,22–25]. Research on individual droplets has revealed that these undesired reactions are suppressed in case the droplet heating rate is very high [26,27]. As the heating rate is highly dependent on droplet size, small droplets are required to avoid deterioration of the pyrolysis oil quality during evaporation. Hence, given the current properties of raw pyrolysis oil, improved atomization can be seen as a key factor towards efficient combustion.

Due to the adverse effect of high viscosity on spray quality, viscous fuels are normally preheated to a temperature at which the flow properties are acceptable. However, in case of pyrolysis oil, the allowable preheating temperature is limited to around 80 °C because of the chemical instability as explained above. This limitation implies that the viscosity of pure pyrolysis oil cannot be lowered to a level that is typical for distillate fuel oils. As a consequence, the spray quality might be affected even if the oil is preheated, so that the technical feasibility of burning pure pyrolysis oil in existing gas turbines should be questioned irrespective of the combustion kinetics.

1.2. Aim of the study

In this work, the effect of viscosity on the combustion efficiency in a micro gas turbine is investigated by changing the injection temperature of a viscous fuel. Here, the fuel viscosity at the pressure-swirl nozzle is used as measure for the spray quality, while CO emissions indicate the combustion efficiency. The main goal is to determine the sensitivity of the combustion process to an increase in fuel viscosity and to identify a typical maximum viscosity for which the atomization is still acceptable.

Straight vegetable oil has been used as the fuel for the experimental test campaign. This biofuel is selected to capture the effect of viscosity, while excluding the complex chemistry effects associated with pyrolysis oil. The stability of vegetable oil allows for preheating up to high temperatures, such that a wide range of viscosities can be tested. The experiments with vegetable oil have been compared to reference measurements with diesel. Next to viscosity effects, also the influence of load has been examined. The results from this study can be useful in formulating practical guidelines for good atomization of pyrolysis oil as a first and very important requirement.

1.3. Previous work

Literature on the combustion of straight vegetable oil in gas turbines is scarce; only few works have been found that are similar or closely related to the present study. Habib et al. [28] tested several vegetable oil-based biodiesels (soy, canola and recycled rapeseed), an animal-derived biofuel and their 50% blends with Jet A fuel in an unmodified 30 kW gas turbine. Engine performance and emissions are reported for all biofuels except for the pure animal fat biofuel due to its high sooting potential. However, no information is given about the injection system and fuel injection temperature.

Cavarzere et al. [29] investigated the performance of a Solar T-62T-32 micro gas turbine fed by blends of diesel and straight vegetable oils in different concentrations. The researchers were able to gradually increase the vegetable oil content up to 100% without running into any particular problems. Unfortunately, the authors did not report the injection temperatures used during the test runs, while they mention that the fuel tanks for vegetable oil blends were heated to reduce the viscosity.

Significant work was published by Chiaramonti et al. [30], who performed experiments with diesel, biodiesel and vegetable oil in a Garrett GTP 30-67 micro gas turbine equipped with a pressure-swirl atomizer. It was found that the combustion of pure vegetable oil and its blends with diesel required preheating of the fuel to at least 120 °C. At idle conditions, combustion could not be sustained even at this temperature. However, the results generally showed that preheating the fuels reduced CO emissions. The effect of fuel temperature was measured to be more pronounced at partial load. Also, it is concluded that the combustion efficiency for vegetable oil at 120 °C is very similar to that of diesel at standard conditions.

These conclusions are confirmed by field tests with a slightly modified Capstone C30 conducted by Prussi et al. [31]. In this micro gas turbine, straight vegetable oil was injected by air-assist atomizers at three different temperatures. CO emissions decreased with higher preheating temperatures, especially at low load conditions, and were nearly the same as for diesel in case of maximum load. Considering the strong influence on the engine performance, viscosity was identified as the most important parameter to be controlled. Comparison of the data obtained by Prussi et al. with other studies will only be possible to a limited extent, however, because the fuel temperatures are not reported explicitly.

The experimental work discussed above indicates that the research on this topic is in an early stage. Straight vegetable oil has been applied in gas turbines only few times, with varying degrees of success, and it can be said that little results have been reported on the quantitative relation between combustion efficiency and viscosity for this fuel.

2. Experimental setup

The combustion experiments have been conducted with the multifuel micro gas turbine (MMGT) setup at the University of Twente. This setup is developed for testing alternative fuels, or biofuels in this case, at varying fuel injection temperatures and loads. It consists of five main modules: a micro gas turbine, a generator, a fuel supply system, a resistive load and a control unit. Various sensors are used to collect data during the experiments. The gas turbine, the fuel supply system and the measurement equipment are discussed in the following subsections.

2.1. Micro gas turbine with generator

The micro gas turbine selected for the test setup is a DG4M-1, which was formerly used as auxiliary power unit. Thanks to its simple, radial design, the DG4M-1 is relatively robust to alternative fuels and easily accessible in case modifications are required. As shown in Fig. 1, it features a single stage, centrifugal compressor and turbine on a single main shaft. The power shaft, which is driven by the main shaft via a gearbox, is connected to a generator to produce electricity. The electrical power is supplied to three 15 kW fan heaters that can be switched on individually to vary the load condition. The fuel is burnt in a single silo, reverse-flow combustion chamber equipped with a pressure-swirl atomizer. This atomizer produces a hollow cone spray with an angle of 90–100°. The specifications of the gas turbine are listed in Table 1 [32].

2.2. Fuel supply system

In the original micro gas turbine configuration, the fuel supply was controlled mechanically and preheating was achieved by leading the fuel line through the exhaust. However, both the fuel control and preheating design were unsuitable for biofuel combustion. The standard feeding system was hence disconnected and replaced by a newly designed fuel supply system (FSS). This new system can be used to operate the gas turbine over a wide range of operating conditions and to switch between standard fuel and biofuel during

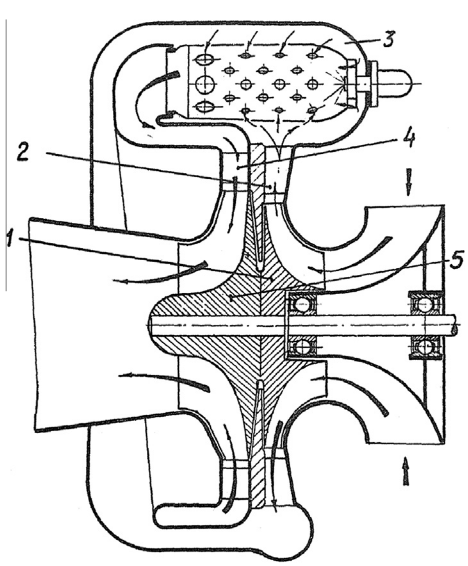

Fig. 1. Functional overview of the DG4M-1 gas turbine: 1 – compressor, 2 – diffuser, 3 – combustion chamber, 4 – turbine nozzle, 5 – turbine. [32].

Table 1
Specifications of the DG4M-1 micro gas turbine with generator [32]. Fuel consumption data are based on diesel No. 2.

Property	Unit	Value
Rotational speed of main shaft	rpm	27 600
Rotational speed of power shaft	rpm	6000
Maximum power	kW_e	47
Compression ratio	–	2.3–2.6
Air consumption	m^3/s	1.2
Fuel consumption at no load	L/h	48
Fuel consumption at full load	L/h	82

a run. In addition, the fuel preheating capacity has been increased considerably to handle viscous fuels.

A flowsheet of the FSS is given in Fig. 2. The diesel fuel, alternative fuel and cleaning fuel are stored in separate tanks. Cleaning fuel is only used for flushing the fuel lines and the atomizer. Diesel fuel is required for the startup and shutdown of the MMGT. After startup, the operator can decide to continue on diesel or switch to the alternative fuel, depending on the type of experiment. The fuel from all three tanks are led through can filters to prevent clogging in the nozzle. A small waste tank is used to drain contaminated fuel during the tests.

The fuel switching is achieved by using two pressurized circuits, one for diesel and one for the alternative fuel. These circuits are separated by a four-way valve that controls which fuel is recirculated and which fuel is supplied to the atomizer. In both circuits, the pressure is controlled by setting a needle valve in a bypass line. The pressure level is chosen such, that the thermal input after switching is similar and therefore the rotational speed of the MMGT is hardly influenced. Besides the fuel pressure, also the temperature is adjusted before switching to minimize the risk on a blowoff. The alternative fuel, which is a viscous biofuel in this research, is preheated in the tank and then further heated by two cartridge heaters in the fuel line.

2.3. Measurement equipment

The operating conditions of the setup are monitored by measuring pressures, temperatures and rotational speed. The pressure in the combustion chamber is measured using a GE PMP 1400 sensor with an accuracy of ±2% of its range 0–16 barg, including errors due to thermal effects. Two RS Type 461 sensors measure the pressure in the fuel line, one in the alternative fuel circuit of the FSS (see Fig. 2) and one just in front of the atomizer. These sensors work with an accuracy of ±0.25% in their range 0–100 barg and have been calibrated using a GE Druck DPI 104. All pressure sensors are connected to analog ports of the NI USB-6009 data acquisition device.

Temperatures are measured in the fuel tanks, in the fuel line close to the atomizer and in the exhaust using K-type thermocouples. These thermocouples have an accuracy of ±2.2 °C or ±0.75% of the measured value. The acquisition of temperature data is performed by an NI 9213 module with a maximum error of ±1.2 °C in the temperature range 0–500 °C. The combined accuracy of the sensor and measurement hardware is therefore ±3.4 °C for temperatures under 300 °C, but increases to ±5.0 °C at a temperature of 500 °C.

The gas composition in the exhaust is measured using several analyzers to cover a wide range of species and to reduce measurement uncertainty. An rbr-ecom KD electrochemical analyzer measures dry gas concentrations of O_2, CO and NO. Furthermore, a set of analyzers composed of a Siemens Oxymat 61 and a Maihak Multor 610 is used to measure O_2, CO_2 and CO in a conditioned gas sample stream. The Oxymat determines the O_2 concentration

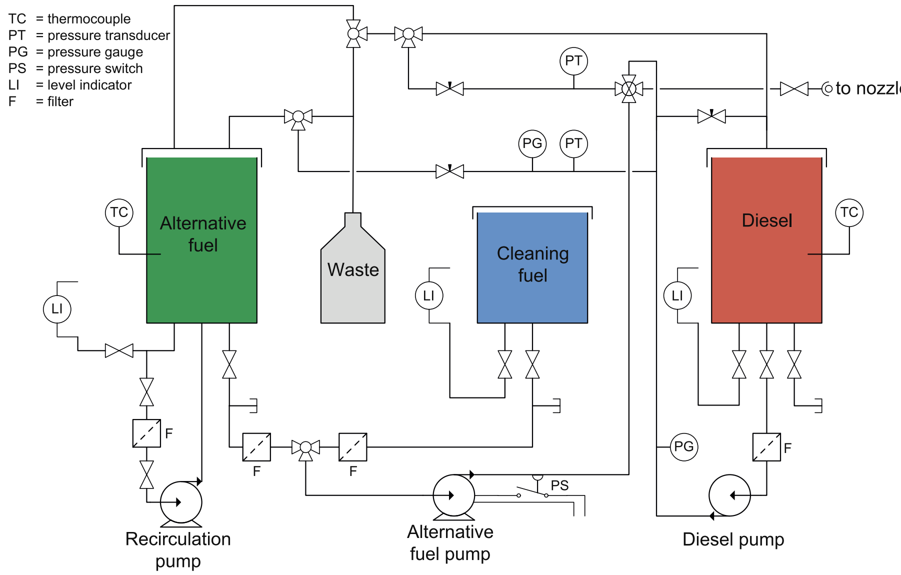

Fig. 2. Flowsheet of the fuel supply system.

based on the paramagnetic alternative pressure method. The Multor uses infrared absorption to detect CO_2 and CO. The measurement ranges and uncertainty levels for these analyzers are given in Table 2. In some of the experiments, the conditioned gas sample was also led through a Gasmet CX-4000 Fourier Transform Infrared (FTIR) analyzer to determine the concentration of unburned hydrocarbons (UHC). The hydrocarbons included in the FTIR library were methane, ethane, ethylene, propane, hexane and formaldehyde.

3. Fuel properties

The fuels tested in the present study are diesel No. 2 and vegetable oil. Diesel is a standard fuel for the gas turbine and was therefore selected as the reference fuel. Vegetable oil was used to investigate the influence of fuel viscosity on the combustion process. The fuel referred to as vegetable oil here is a mixture of plant-derived oils, commercially available as liquid frying fat. Typical fuel properties important for spray combustion are listed in Table 3.

Although the application of pyrolysis oil is the ultimate goal, there have been reasons to perform the experiments with vegetable oil instead. Firstly, the evaporation behavior as well as the combustion kinetics of vegetable oil and diesel fuel are similar. The differences in fuel composition will therefore have only minor effects, as was concluded by Panchasara et al. [37] after experiments in an atmospheric pressure burner. This leaves the atomization quality as the main factor influencing the combustion of this biofuel, which is dominated by the viscosity in the tested fuel temperature range [30,31]. It is hence most important that the viscosity of vegetable oil is representative for pyrolysis oil.

Table 2
Specifications of the gas analyzing equipment.

Analyzer	Species	Range	Uncertainty
ecom KD	O_2	0–21 vol.%	±0.3 vol.%
	CO	0–400 ppm	±20 ppm
		400–4000 ppm	±5% of value
	NO	0–2000 ppm	±5 ppm or 5% of value
Oxymat	O_2	0–25 vol.%	±1% of range
Multor	CO_2	0–20 vol.%	±2% of range
	CO	0–2000 ppm	±2% of range

Table 3
Properties of the two tested fuels and pyrolysis oil. Data obtained from the EN590 norm and from literature [33–35,1,36]. Viscosities of diesel and VO have been measured (see Fig. 3).

Property	Unit	Diesel No. 2	Vegetable oil	Pyrolysis oil
Lower heating value	MJ/kg	42–43	38–39	16–19
Density[a]	kg/L	0.82–0.84	0.92	1.1–1.3
Surface tension[b]	mN/m	28	31–34	30–36
Flash point	°C	>55	>230	>40
Viscosity	cSt			
at 20 °C		4.2	61	>50
at 40 °C		2.5	27	>14
at 80 °C		1.3	8.4	>4

[a] Density at 20 °C.
[b] Surface tension at 25 °C.

As indicated in Table 3, this property highly varies between the different pyrolysis oils because it very much depends on the feedstock and the production process [33]. Furthermore, once the oil is produced its viscosity can change over time due to aging reactions [41]. Although a comparison is not so straightforward for these reasons, it is clear that the vegetable oil mixture is sufficiently viscous to make it a good substitute for this research. The viscosity of most types of pyrolysis oil at a temperature of 80 °C, considered as the upper temperature limit for pyrolysis oil, can be matched by vegetable oil in the tested temperature range of 20–120 °C. This is also illustrated in Fig. 3, where the viscosity of diesel No. 2, vegetable oil and a number of pyrolysis oils are compared as function of temperature. The data shown for the first two fuels, indicated with solid markers, have been measured using a Brookfield DV-II + Pro viscometer. Viscosity data for the pyrolysis oils, indicated with open markers, are taken from literature [13,17,38–40].

4. Test conditions and procedure

To experimentally investigate the influence of spray quality on the combustion process, reference experiments with diesel No. 2 have been compared to experiments with vegetable oil. The experiments were performed under varying load conditions and

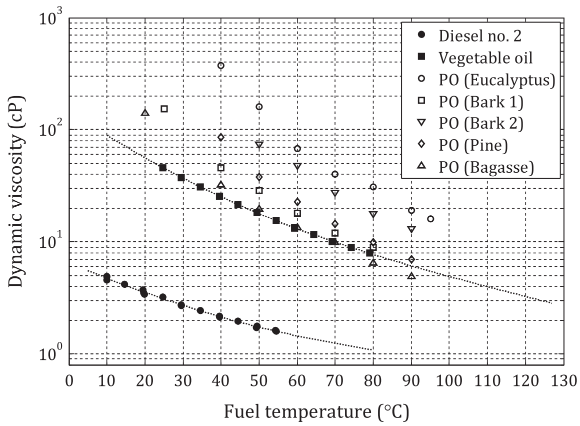

Fig. 3. Dynamic viscosity of diesel No. 2, vegetable oil and pyrolysis oil as function of temperature. Pyrolysis oil data is taken from literature [13,38,17,39,40].

fuel injection temperatures while recording the exhaust emissions. An overview of the investigated test conditions is shown in Table 4. Some of the experiments are overlapping to check the repeatability of the results.

The test procedure starts by preheating the alternative fuel in the tank and by setting the correct pressures of the two fuel circuits in the FSS. After startup of the MMGT on diesel, the operating conditions are adjusted to match the planned test case. Once the values reported by all sensors seem to be constant, the measurement continues for at least another minute to be sure that stationary conditions have been reached. The MMGT ran stably at all tested conditions and no flame extinction occurred during the series of experiments.

Continuous gas samples are taken from the exhaust and led to the gas analyzers ecom KD, Oxymat, Multor and sometimes also the FTIR. These analyzers together measure concentrations of O_2, CO_2, CO, NO and UHC. Especially the CO level is important as it indicates incomplete combustion due to poor atomization, bad mixing or cold spots inside the combustor. CO can therefore be regarded as a good measure for the efficiency of the combustion process. NO is an indicator for the (maximum) temperature in the reaction zone.

5. Results

This section presents the results of the experiments as described in Section 4. Firstly the results of the reference measurements with diesel No. 2 will be discussed. These measurements give information about the performance of the MMGT fired on its standard fuel. Subsequently the results of the vegetable oil combustion tests are reported and compared to the reference cases.

Table 4
Overview of the test conditions in the performed experiments.

Exp.	Fuel[a]	Load (kW_e)	Fuel temperature (°C)[b]
1	Diesel	0, 15, 30, 47	31[c]
2	Diesel	40[c]	31, 42, 59, 70, 80
3	Diesel	40	17,18,21,30,40
4	VO	0	73,80,83,117
5	VO	0	74,80,90,107,121,125,127
6	VO	18	73,78
7	VO	28	86,109
8	VO	46[c]	91,108
9	VO	0	92,112
10	VO	30	92,110
11	VO	46[c]	90,108

[a] Diesel = diesel No. 2, VO = vegetable oil.
[b] Fuel temperature at injection location.
[c] A fluctuation of ±1 was observed during the tests.

The measurement uncertainties of the gas analyzers reported in Section 2.3 are indicated in the graphs by error bars, except for very small errors that would be almost invisible. In case normalized gas concentrations are shown, the measured values have been normalized to 15 vol.% O_2 as follows:

$$[CO]_{norm} = [CO]_{meas} \cdot \frac{[O_2]_{amb} - 15\%}{[O_2]_{amb} - [O_2]_{meas}} \qquad (1)$$

where the subscripts 'norm', 'amb' and 'meas' indicate normalized, ambient and measured values. It must be noted that, strictly speaking, the normalized values come with larger relative error bars than the measured values because of the propagated uncertainty of the oxygen measurement (see Table 2). Nevertheless, as will be explained in Section 5.1, the oxygen measurements are found to be very accurate, so that it is justified to only reckon with the uncertainty of the measured concentration to be normalized (i.e. $[CO]_{meas}$ in Eq. (1)).

The O_2 and CO concentrations measured by the ecom KD on the one hand and the Oxymat–Multor combination on the other hand were found to correspond well, taking into account their ranges of uncertainty. For the sake of readibility of the figures, however, only O_2 and CO data from the Oxymat and Multor are reported, unless stated otherwise.

5.1. Diesel – tests at varying load

In experiment 1, diesel has been combusted at different load conditions without preheating. This test gives insight into some general characteristics of the MMGT running on standard fuel, but also shows the combustion efficiency at varying load. The fuel consumption is estimated using the following relation:

$$\dot{V} = FN\sqrt{\frac{\Delta p}{\rho_l}} \qquad (2)$$

where $\dot{V}$ is the volume flow rate, FN is the flow number of the atomizer, Δp is the differential pressure and ρ_l is the liquid density. The flow number of the atomizer was measured to be around $39{\cdot}10^{-8}$ m^2 prior to the tests.

The injection pressure as well as the estimated fuel consumption during the test are given as function of the load in Fig. 4. The graph shows that the fuel consumption increases linearly with the load from 48 to 82 L/h for idle and 47 kW_e, respectively. This linear trend was also observed in diesel combustion tests performed by Chiaramonti et al. [30] using a similar micro gas turbine.

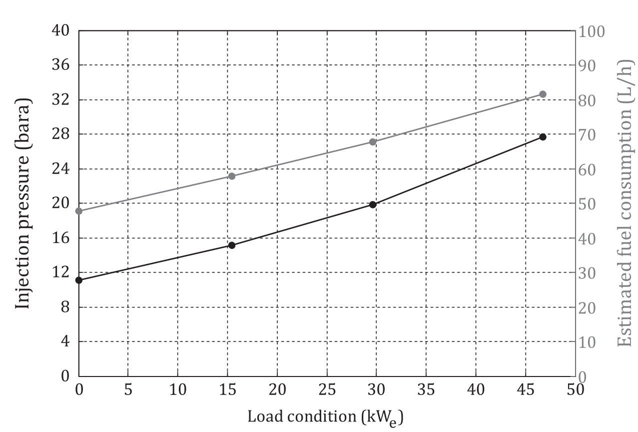

Fig. 4. Fuel injection pressure and estimated fuel consumption as function of load for diesel combustion (Exp. 1).

The fuel consumption data together with the intake air flow rate given in Table 1 define the fuel–air equivalence ratio (ϕ) at the different load conditions. These are shown in Table 5 for a stoichiometric air-to-fuel ratio of 14.7 for diesel. It can be seen that the overall equivalence ratio is low in all cases, which indicates that the combustor operates under very lean conditions and that the turbine inlet temperature will be relatively low.

Fig. 5 shows the concentrations of O_2 and CO_2 as function of the load. The exhaust gas temperatures (EGT) measured in these experiments are reported in Fig. 6. The linear decrease in $[O_2]$ and linear increase in both EGT and $[CO_2]$ confirm the consistency of the measurements. After normalization to 15 vol.% O_2 using Eq. (1), the concentration of CO_2 is found to be 4.3–4.4 vol.% for all load conditions. These values are in close agreement with the ideal combustion theory, given the fact that the H/C-ratio of diesel is typically around 1.9. It can therefore be stated that the Oxymat data are sufficiently accurate to neglect the propagating uncertainty of the oxygen measurements in the normalization.

Normalized emissions of CO are given in Fig. 7. The graph shows that the normalized CO concentration decreases significantly with increasing load. More complete combustion of the fuel is indeed expected, since combustor temperatures increase with load. Improved atomization due to higher injection pressures and hence faster evaporation can also play a major role here. It must be noted that, in absolute sense, the CO emissions from this gas turbine are quite high compared to typical values for modern designs. Such high emission levels have also been reported by Chiaramonti et al. [30] and are the result of the simple design and large air excess. The measurement data are however merely used to capture trends in this study, so that the absolute values are of minor importance.

The normalized NO emissions were found to be rather low, with concentrations increasing from 7 ppm at idle condition to 32 ppm at full load. This indicates that the temperature in the flame tube is relatively low, as can also be seen from the low overall equivalence ratios reported in Table 5. The reliability of the NO measurements is limited, however, considering the measurement uncertainty at such low concentrations (see Table 2).

5.2. Diesel – tests at varying fuel temperature

The influence of the injection temperature is investigated in experiments 2 and 3. In these experiments, the temperature was varied between 17 and 80 °C while all other parameters were held constant. Although the viscosity of diesel No. 2 is already low at room temperature, it still decreases by almost 3 cP over this range (see Fig. 3).

The normalized CO concentration as function of the injection temperature is shown in Fig. 8. The CO levels are only little affected at temperatures above 40 °C, where the viscosity of diesel is below 2 cP. When going towards lower temperatures from this point the CO concentration seems to increase substantially, but the trend is not fully consistent. The unexpectedly low concentrations at 17 and 18 °C can be explained by looking at the rotational speed of the MMGT, indicated by the combustion chamber pressure (CCP) in this figure. It can be seen that the CCP has a strong inverse effect on the CO concentration, possibly caused by improved swirl and mixing. This effect can also explain the slight deviation between the two data points around 30 °C, which should ideally fall on top of each other.

To make a fair comparison between the emissions at different temperatures, the measured concentrations should be corrected for changes in the CCP. However, since the exact dependence of CO emissions on the pressure is unknown, any suggestion for a

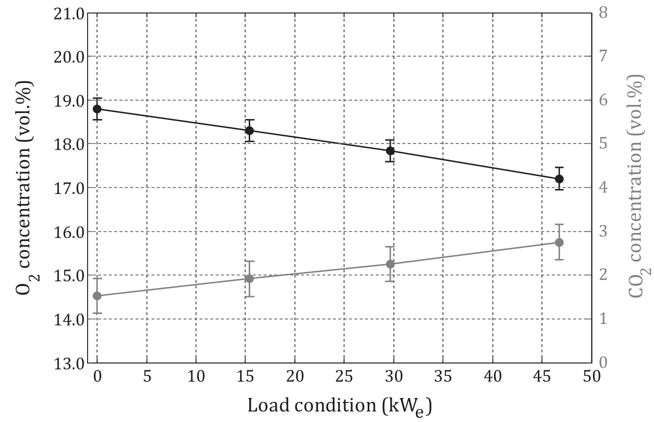

Fig. 5. Measured O_2 and CO_2 emissions as function of load for diesel combustion (Exp. 1).

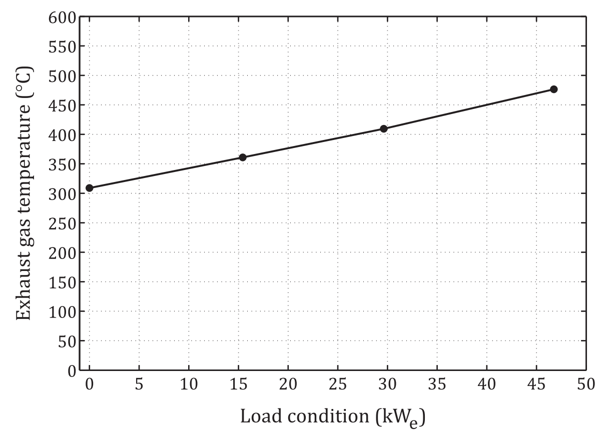

Fig. 6. Measured exhaust gas temperature as function of load for diesel combustion (Exp. 1).

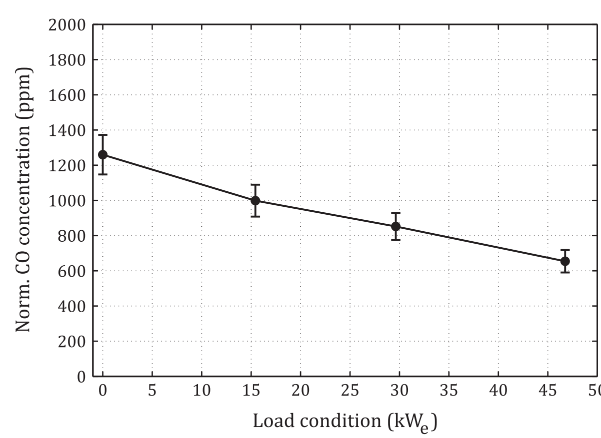

Fig. 7. Normalized CO emissions as function of load for diesel combustion (Exp. 1).

Table 5
Fuel–air equivalence ratios (ϕ) for diesel combustion at different load conditions (Exp. 1).

Load (kW$_e$)	$\phi_{overall}$
0	0.11
15.4	0.13
29.6	0.15
46.8	0.18

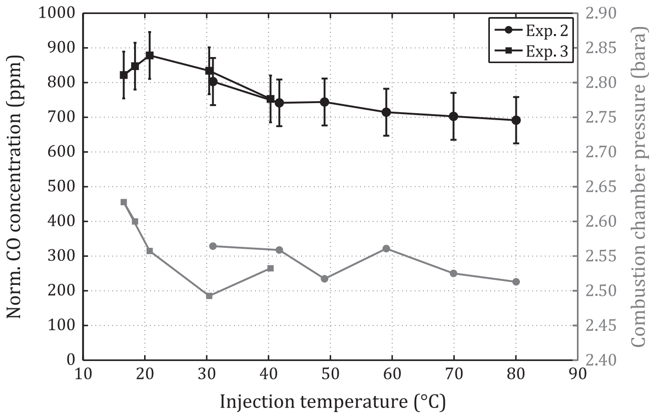

Fig. 8. Normalized CO emissions and combustion chamber pressure (CCP) as function of injection temperature for diesel combustion with 40 kW_e load (Exp. 2 and 3).

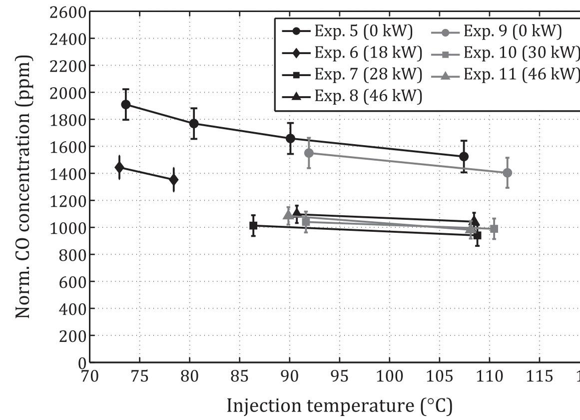

Fig. 9. Normalized CO emissions as function of injection temperature for VO combustion (Exp. 5–11). Exp. 9–11, shown in gray, have been performed to verify Exp. 5, 7 and 8, shown in black. The marker symbol denotes the load condition.

correction method would be speculative. Nevertheless, when only considering the data in the temperature range from 20 to 80 °C, the CO graphs in Fig. 8 show a consistently decreasing trend. The CO emissions are reduced by about 21% over this temperature range.

To avoid influence of combustion chamber pressure fluctuations, the rotational speed was controlled more precisely in the subsequent experiments. Therefore, a strong effect of the CCP on the presented results for vegetable oil can be ruled out.

Some comparisons with the results from experiment 1 can be made to validate the repeatability of the measurements. In experiments 2 and 3, two tests were performed with a fuel temperature of around 30 °C. This setpoint corresponds to the temperature condition in experiment 1. The data shown in Fig. 8 around 30 °C can therefore be compared to the CO emissions given in Fig. 7 at a load of 40 kW_e. It can be seen that the values are in good agreement. The normalized NO concentrations detected in experiments 2 and 3 fluctuated in the range 24–31 ppm for all temperature conditions. This result is very similar to the NO emissions measured in experiment 1 at higher loads. Emissions of UHC, measured with the FTIR during experiment 2, were found to be insignificant.

5.3. VO – tests at varying load

Although the effect of fuel temperature has been determined prior to the influence of load, the experimental results for vegetable oil at varying load will be discussed first to make the structure of the results section consistent. As reported in Table 4, always two temperature setpoints have been evaluated when running on vegetable oil with a certain load. The required injection pressures were about 10% higher compared to those for diesel, which is a combined effect of the lower heating value and higher density of vegetable oil (see Table 3).

In Fig. 9, the normalized CO emissions are reported for various load conditions. Different symbols are used to denote the load applied in the experiment. As is the case for diesel combustion, the influence of load addition is clearly more pronounced than the effect of injection temperature. The reduction can be attributed to the higher combustor temperatures and improved atomization. Also, the slopes of the graphs indicate that the sensitivity of CO emissions to fuel temperature diminishes with increasing load.

A further decrease in CO emissions can be expected when the load is increased from 28 to 46 kW, but the data from experiments 7 and 8 show no significant differences. Such unexpected behavior is not observed in case of diesel combustion at varying loads (see Fig. 7). To verify this outcome, a new series of test runs, consisting of experiments 9–11, have been performed at approximately the same test conditions. The results of these validation experiments are plotted in gray in Fig. 9 and generally confirm the values obtained in the previous tests. Possibly, the positive effect of higher combustor loading on the droplet evaporation rate was counteracted by an increase in spray penetration length. The injection pressure increases with load, so that the initial droplet velocity rises as well. Since the vegetable oil droplets are supposedly larger than the diesel droplets due to the higher viscosity, the vegetable oil spray is expected to penetrate further into the flame tube. For this reason, the combustion reactions might have been quenched sooner by the secondary air than in case of burning diesel.

5.4. VO – tests at varying fuel temperature

Experiments with vegetable oil at various temperatures have been performed to investigate the effect of fuel viscosity on the combustion process. High viscosity can lead to poor atomization, which in turn causes incomplete combustion or even blowoff. The fuel injection temperatures were varied between 73 and 127 °C, but the fuel switch from diesel to vegetable oil was always conducted at a temperature of 90 °C to prevent a blowoff.

The influence of fuel temperature on the injection pressure was observed to be insignificant. Indeed, variations in density in the tested temperature range are rather low, and the thermal input is only proportional to the square root of the density according to Eq. (2). Since fuel viscosity did not change the nozzle flow number, the inviscid flow assumption made in this equation is confirmed.

Fig. 10 shows raw measurement data obtained from experiment 4 using the Multor and the FTIR. The figure illustrates the evolution of the fuel injection temperature along with nondimensionalized concentrations of CO and C_2H_4 against time. The FTIR was used to measure several hydrocarbons (see Section 2.3), but only ethylene (C_2H_4) was found in significant concentrations. CO was measured by both the FTIR and the Multor. Note that the emission graphs are slightly shifted with respect to the temperature graph due to the time lag of the analyzers.

In the experiment the fuel temperature was adjusted in a high–low–high pattern, which has resulted in an inverse pattern for the concentration of both CO and C_2H_4 in the exhaust gas. Obviously the two species show the same trend, but only C_2H_4 approaches zero at high injection temperatures. The CO emission levels measured by the Multor and FTIR are in fair agreement.

Normalized CO concentrations acquired in experiments 4 and 5 are given as function of the fuel injection temperature in Fig. 11. It can be concluded that the fuel temperature has considerable

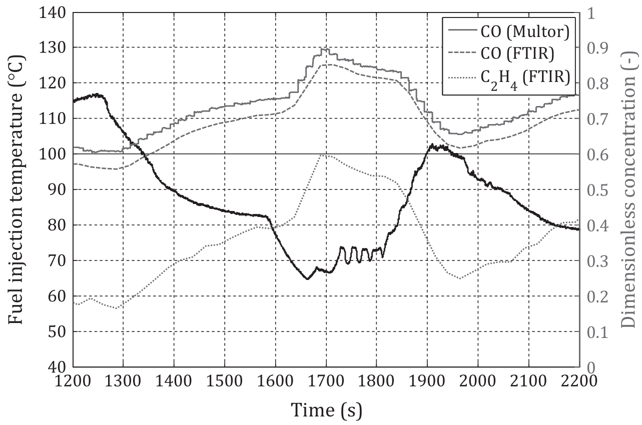

Fig. 10. Fuel injection temperature along with dimensionless emissions of CO and C_2H_4 as function of time. Raw measurement data for VO combustion without load (Exp. 4).

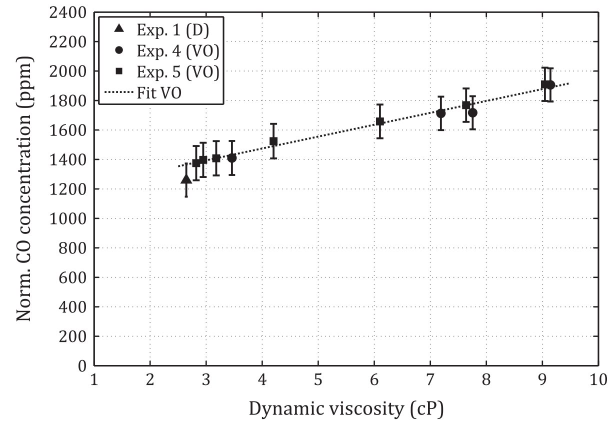

Fig. 12. Normalized CO emissions as function of fuel viscosity for VO combustion without load (Exp. 4 and 5).

influence on the combustion efficiency; an increase in temperature from 73 to 127 °C has resulted in a CO reduction of 28%. The normalized NO concentrations were 6–7 ppm for all test conditions. Emissions of NO were therefore identical to the diesel case without load and can be considered as insignificant.

It must be noted that the appearance of the exhaust gas changed from highly transparent to slightly opaque at fuel temperatures lower than approximately 70 °C. At such low temperatures blue smoke was observed, most likely because the time required for full evaporation of the poorly atomized oil was insufficient. The vegetable oil then starts to break down into glycerol and free fatty acids, which is known to give bluish smoke [42]. The observation of smoke in the exhaust prescribed the lower temperature limit in this parameter study.

The improved combustion efficiency at higher fuel temperatures is caused by enhanced atomization at lower fuel viscosities. To illustrate the effect of viscosity on the CO emissions more directly, these two parameters have been plotted against each other in Fig. 12. The graph shows a linear relation between CO and viscosity in the tested range.

An interesting result is that the normalized CO level for diesel and vegetable oil is similar when the viscosity is matched. The viscosity of vegetable oil at 120 °C is close to 3 cP, which almost corresponds to the viscosity of diesel at 31 °C in experiment 1 (see Fig. 3). In Fig. 12, it can be seen that the normalized CO concentration is approximately 1300 ppm in both cases.

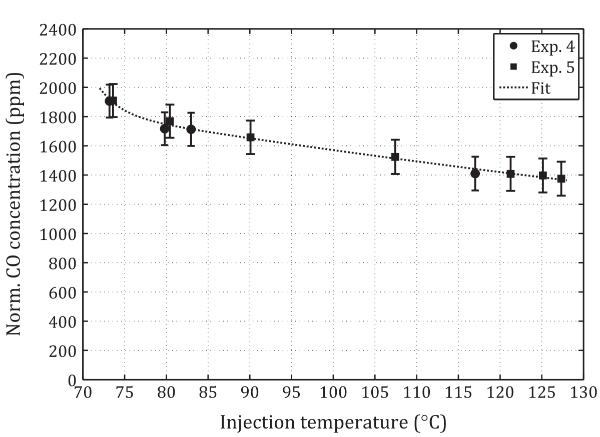

Fig. 11. Normalized CO emissions as function of injection temperature for VO combustion without load (Exp. 4 and 5).

The influence of fuel preheating on the evaporation curve of the droplets should also be considered in this discussion. Preheating adds a significant amount of energy to the fuel with respect to the total energy needed for evaporation. As a result, the volatility of the droplets is higher when they are injected into the combustion chamber. Compared to the evaporation time, however, the heat-up time of a droplet is relatively short and the wet bulb temperature is reached quickly [43]. The total evaporation time of the fuel droplets is therefore expected to be hardly affected by the preheating temperature.

The relation between CO emissions and injection temperature or viscosity presented in this section are confirmed by additional tests performed with biodiesel. The results from the biodiesel tests are reported in Appendix A and show similar trends as seen from the experiments with diesel No. 2 and vegetable oil.

6. Implications for pyrolysis oil combustion

The results from this study provide an important guideline for burning pyrolysis oil in gas turbines, a more advanced biofuel with relatively low impact on the environment. Atomization problems due to high viscosity form a major obstacle for clean burning of this biofuel according to earlier studies. From the tests with vegetable oil, it appears that the maximum dynamic viscosity at which the spray can still be evaporated is 9 cP. This value is in close agreement with the preliminary norm for pyrolysis oils as described by Oasmaa et al. [33], which sets the maximum kinematic viscosity at 7 cSt (8 cP) for application in gas turbines. However, in case pyrolysis oil is to be combusted in the MMGT, it is expected that the viscosity requirement must be more strict because of the lower volatility and slower kinetics compared to vegetable oil. The viscosity limit prescribed by the norm may therefore be too optimistic if pressure-swirl atomization is used for pyrolysis oil combustion in a conventional combustion chamber.

In many cases, this condition for viscosity cannot be met only by preheating pyrolysis oil up to the temperature limit of 80 °C (see Fig. 3). To further reduce the viscosity, the oil can be blended with another fuel that has good flow properties [44,45]. Here, it is worth mentioning the experimental work performed by Pander [46], who tested the combustion of diesel/butanol/pyrolysis oil mixtures in the same gas turbine as used in this study. Blends with the percent compositions 50/40/10 and 5/50/45 by mass, with respective viscosities of 3.4 and 7.0 cP measured at 23 °C, were successfully combusted at partial load without preheating. A third mixture, with percent composition 0/30/70 and a viscosity of 29 cP at 23 °C, failed to combust even after preheating up to 60 °C.

For applications of pure pyrolysis oil, it is strongly recommended to improve the spray quality by using other atomization methods. Twin-fluid atomizers are promising alternatives, because their performance is known to be less sensitive to changes in fuel viscosity compared to pressure-swirl nozzles. The potential of such atomizers for the combustion of a viscous, wood-derived biofuel has been demonstrated by Seljak et al. [47]. In this research, preheated liquified wood with a viscosity of around 100 cP could be burned in a jet engine combustor using air-assist atomization.

7. Conclusions

Combustion of a viscous biofuel has been studied in a multifuel micro gas turbine setup. The effect of viscosity on the combustion performance is investigated by changing the fuel injection temperature. Successful experiments have been conducted with a viscous mixture of straight vegetable oils. The results were compared to reference measurements with diesel No. 2.

From the present work, it can be concluded that the combustion efficiency, expressed by the CO emissions, can be significantly improved if the fuel injection temperature is increased. The measurements confirm that a reduction in fuel viscosity improves the spray quality, which results in faster droplet evaporation and more complete combustion.

The measurements with vegetable oil have revealed a linear relation between viscosity and CO emissions. CO concentrations were reduced by 28% when decreasing the viscosity from 9 to 3 cP. The same effect is observed in case of diesel combustion, with a CO reduction of 21% when the viscosity was lowered from 3 to 1 cP. At viscosities above 9 cP the exhaust gas contained considerable amounts of unburned fuel, presumably caused by incomplete evaporation.

Comparison of the results for the two fuels has shown that the CO emission is similar when the viscosity is the same. This implies that fuel preheating is an effective way to fully restore the combustion quality for viscous fuels. The influence of fuel temperature and viscosity on the CO emissions observed in the results for diesel and vegetable oil is confirmed by additional tests with biodiesel.

Combustion experiments at different loads have shown that the CO levels also strongly depend on the equivalence ratio. In case of diesel, increasing the load resulted in reduced emissions over the full range from 0 to 47 kW_e. This trend was not in full agreement with the results for vegetable oil, where no further decrease in CO level was observed above a load of 28 kW_e.

The experimental campaign with vegetable oil gives directions for the application of pyrolysis oil in gas turbines, a more advanced biofuel with high viscosity. Although the burning characteristics are different for these biofuels, the limited tolerance to higher fuel viscosity measured in this study questions the potential of pyrolysis oil combustion in the present setup configuration.

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Appendix A. Results for biodiesel combustion

In addition to the tests with diesel and vegetable oil (VO), combustion experiments with biodiesel (BD) have been performed. In these tests, the fuel injection temperature was varied between 17 and 61 °C while running without a load. The results are presented in Appendix A.2 after a short discussion of the biodiesel properties in Appendix A.1.

A.1. Biodiesel properties

The biodiesel used for the present experiments is produced by Sunoil in Emmen (The Netherlands). Sunoil produces fatty acid methyl esters (FAME) via transesterification of primarily used fats originating from plants and animals [48]. The Sunoil biodiesel is commercially sold as B100 and complies to the EN 14214 norm. Its properties are summarized in Table A.6 and can be compared with the properties of the other fuels, reported in Table 3.

In Fig. A.13, results from biodiesel viscosity measurements are compared to the viscosity data for the other fuels as presented in

Table A.6
Properties of the tested biodiesel. Data obtained from the EN 14214 norm and from literature [1,49]. Reported viscosities were measured (see Fig. A.13).

Property	Unit	Biodiesel
Lower heating value	MJ/kg	39–41
Density[a]	kg/L	0.86–0.90
Surface tension[b]	mN/m	26–32
Flash point	°C	>101
Viscosity	cSt	
at 20 °C		7.2
at 40 °C		4.1
at 80 °C		1.8

[a] Density at 20 °C.
[b] Surface tension at 25 °C.

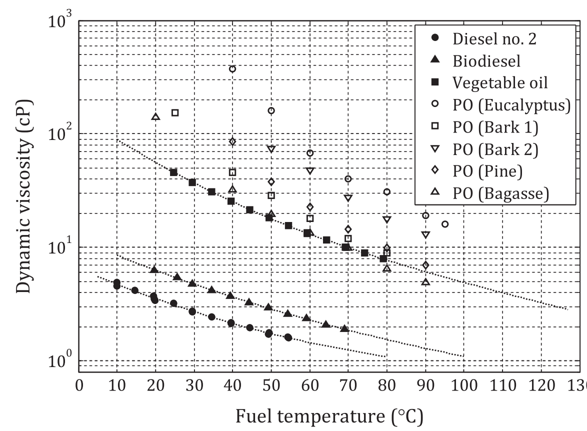

Fig. A.13. Dynamic viscosity of diesel No. 2, biodiesel, vegetable oil and pyrolysis oil as function of temperature. Pyrolysis oil data is taken from literature [13,38,17,39,40].

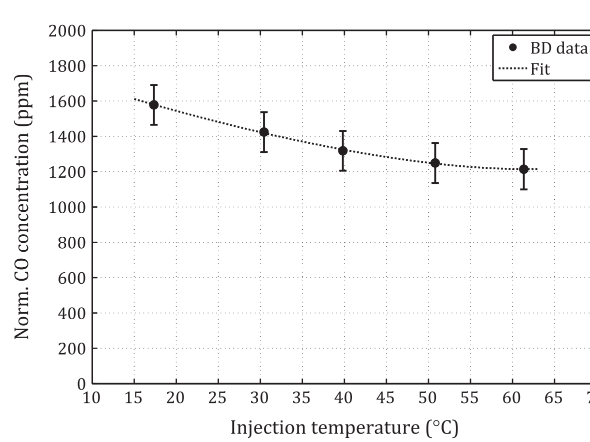

Fig. A.14. Normalized CO emissions as function of injection temperature for BD combustion without load.

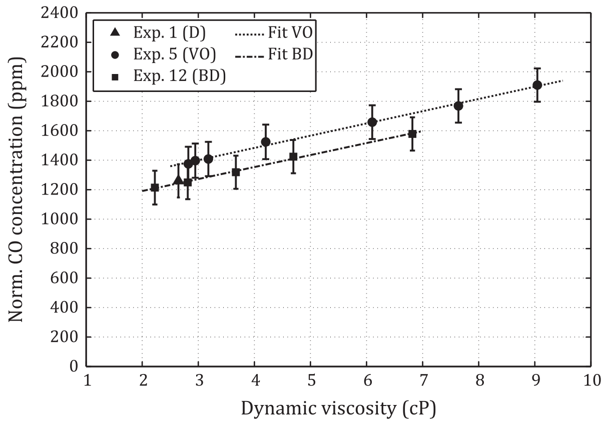

Fig. A.15. Normalized CO emissions as function of fuel viscosity for diesel, VO and BD combustion without load.

Fig. 3. The graph shows that biodiesel is slightly more viscous than diesel.

A.2. BD – tests at varying fuel temperature

The normalized CO emissions measured as function of the fuel injection temperature are shown in Fig. A.14. Emissions are reduced by increasing the temperature, as was seen in case of diesel and vegetable oil combustion (see Figs. 8 and 11). However, the effect diminishes until no further reduction in CO level can be observed around a temperature of 60 °C. At this temperature, the viscosity of biodiesel is approximately 2 cP. The result confirms the trend for diesel shown in Fig. 8 above 40 °C. In that temperature region, corresponding to viscosities below 2 cP, the effect of injection temperature is negligible as well.

Fig. A.15 presents the normalized CO emissions as function of the fuel viscosity. The results from the biodiesel tests have been added to the data shown in Fig. 12. The trends are linear in both cases and the slopes are in good agreement. In absolute sense, CO concentrations are lower in case of biodiesel combustion.

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