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Waste fish oil as a renewable fuel of IC engines

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Abstract. Bio-oils are potential fuels for internal combustion engines because of their advantages such as biodegradability, renewability, high oxygen content and low sulphur. However, their high viscosity, surface tension, and density of crude bio-oils pose challenges for engine use. Those properties affect fuel spray characteristics, mixture formation and combustion. In turn, these impact engine efficiency, power and emissions. This study investigated the use of crude fish oil (FO) at medium and low engine loads in an off-road engine. The injector had 6-hole high flow rate tips. The results were compared with those of fossil diesel fuel (DFO). Fish oil increased hydrocarbon (HC), carbon monoxide (CO) and particulate emissions of nitrogen oxides (NOx) and smoke, however, decreased. Crude fish oil also showed lowered total particulate number (TPN) at low load at low engine speed compared to DFO.

Keywords: diesel engine; bio-oil; combustion; gaseous emissions; particulate number.

INTRODUCTION

Bio-based fuels can provide economic, social, and environmental benefits, especially if the raw material used in their production is abundant. Consequently, the focus for biofuel production is switching to alternative raw materials such as non-edible vegetable oils, used cooking oils, fatty acids from algae, and animal fats (Sirviö 2018; Ching-Velasquez et al. 2020).

According to its 2009 Green Paper programme, Finland will be carbon-neutral by 2035 (Ministry of the Environment 2020). The already existing blend of biodiesel and fatty acid methyl ester (FAME) fuels are not a realistic way to increase the share of renewables to fulfil Finnish government and European Union targets (Sirviö, 2018). Many Finnish farms and factories are willing to increase the self-sufficiency of their energy production by utilising waste materials like crude fish oil as a fuel feedstock (Niemi et al., 2009; Niemi et al., 2011).

The crude oil extracted from discarded parts of marine fish may provide an abundant and stable source of raw oil to allow maritime countries to produce a biodiesel and help to reduce pollutant emissions (Lin & Li, 2000; Almeida et al., 2015). However, diesel engines can be even utilised by bio-oils, such as animal fats, vegetable oils, and waste oils. This increases the interest in crude bio-oils (Niemi et al.,

2009; Niemi et al. 2011; Hoang, 2011. The use of straight bio oils with minimal refining should be preferable since refining always consumes energy and adds carbon dioxide (CO₂) emissions (Niemi et al., 2011). Power plants with medium-sized engines fueled by neat bio oil are already in operation around the world (Ollus & Järvi, 2011; Niemi et al., 2011).

World fish production in 2011 was around 170 million tonnes, of which 1% was used for non-food purposes (Food and Agriculture Organization of the United Nations, FAO, 2012). More efficient and sustainable aquaculture production must be implemented since a large proportion - as much as 35% - of production is either lost or wasted. Improvements can be achieved through appropriate policies, regulatory frameworks, availability of breeding services and infrastructure, as well as physical access to markets. FAO, 2012.

Most of the fish oils used in the cosmetic, pharmaceutical, and human dietary supplements industries (Bruun, 2019). Fish oil is also used in stationary diesel engines (Bruun, 2019). The oil can be also processed further to produce biodiesel. The fish oil is already large potential source of raw materials for production.

Crude bio-oils have several advantages as fuels in ICE engines. Compared with DFO, many studies report a significant reduction of toxic emissions and noise; small or insignificant greenhouse gas (GHG) emissions and lower emissions of NOx, polycyclic aromatic hydrocarbons (PAHs), particulate matter (PM) and smoke (Howe et al., 2012; Muraleedharan, 2011; Eng, 2012).

Although liquid biofuels are a good alternative to fossil fuels, there are challenges associated with their use. For example, studies by Niemi et al. (2011), Deshmukh et al. (2011), Ean et al. (2011), and Sirviö (2011) concluded that the high viscosity of neat bio-oils affects fuel atomization and efficient combustion. They point out specific issues of large droplet size, oil spray penetration, formation of soot, injector coking, ring sticking, piston seize-up, lubrication, filter clogging etc. Other limiting factors for the use of neat bio oils are their lower heating value and calorific value, higher density and surface tension when compared to DFO, as well as their acidic and corrosive properties, plus their water and oxygen (Howe et al., 2012; Bruun, 2019; Hoang, 2011). Studies by Hoang (2011), Rajopulos et al. (2011) and Chahar et al. (2010) report that crude bio-oils significantly reduced power output and increased deposit formation in the combustion chamber and injector holes, resulting in increased carbon monoxide (CO) and unburnt hydrocarbon (HC) emissions.

Unlike FAME biodiesels, crude bio-oils do not have common quality specifications. Some producers have their own specifications for viscosity, density, water content, acid number, sulphur content etc. These limits are based on their experiences of using bio oils in diesel engines (Bruun, 2019). For example, according to Ollus & Järvi (2007), the acid number of crude bio oils should be below 5.0 mg KOH/g of water content less than 0.2% (V/V⁻¹); the sulphur content less than 500 ppm; and phosphorus content below 10 ppm.

The main research question of the current study was whether fish oil from left-over fish trimmings could be used as an alternative fuel in the diesel fishermen's vessels. The study was part of a project that investigated the potential to make more efficient use of fish trimmings and bycatches in Ostrobothnia, inland Skogetla, 2011. The present study investigated the use of crude fish oil in a high-speed, common rail diesel engine

equipped with 6-hole injector nozzles with high flow rates. The engine was driven at three loads and at two speeds. The results were compared to those when fuelled with DFO. All the engine parameters were unchanged. The measurements provided new information on the suitability of crude fish oil for a high-speed, off-road engine particularly with regard to total exhaust particle numbers (TPN). The results support and promote efficient use of renewable fuels in ICE.

MATERIALS AND METHODS

Experimental setup

The University of Vaasa (UV) conducted the experiments at the Internal Combustion Engine (ICE) laboratory of the Technobothia laboratory unit in Vaasa, Finland.

Engine setup

The experimental engine was an AGCO Power 44CWA, was a turbocharged and intercooled high-speed, four-cylinder diesel engine of non-road applications. It had a Bosch common rail fuel injection system but no exhaust gas after-treatment. The engine was loaded by means of a Honda WT30 eddy current dynamometer. The main specification of the engine is given in Table 1.

The current study was an extension to the research of how selected fuel injection nozzles affect the injection, combustion, and emission characteristics of a modern high-speed common-rail diesel engine (Hissa et al. 2020). The study compared sootoid driven injectors with 6-, 8- or 11-hole nozzles. The 6-hole injector nozzles were selected for use with the crude fish oil because the larger orifices of the 6-hole nozzles were more suited to the high-viscosity FO. Three different engine loads were used. Loads of 50% and 25% were applied at engine speed of 1,500 min^{-1} (intermediate speed): a 10%

load was applied at the engine's rated speed of 2,000 min^{-1} . The nozzles had a high mass flow rate (1.2 L min^{-1} at 1,000 rpm) and the spray angle was 149°. Modern diesel combustion systems use spray angles in the range of 145°–158°. The engine manufacturer optimized the injection map for the 8-hole nozzles, but the same map was used with the alternative 6-hole nozzles. Table 2 gives the specifications of the 6- and 8-hole nozzles.

Table 1. Main engine specification

Engine	AGCO POWER 44CWA
Cylinder number	4
Bore (mm)	108
Stroke (mm)	120
Swept volume (dm^3)	4.4
Rated speed (min^{-1})	2,000
Rated power (kW)	96
Intermediate speed (min^{-1})	1,500

Table 2. Specification of the 6- and 8-hole injector nozzles

Number of nozzle holes	6	8
Orifice diameter (mm)	0.2	0.162
Total orifice areas (mm^2)	0.18	0.165
Included spray angle	149°	149°
Nozzle flow rate (L min^{-1}) at 1,000 rpm	1.2	1.2

Physical properties of est liquids

The baseline fuel was a commercial low sulphur diesel fuel oil (DFO). The unprocessed crude fish oil (FO) was purchased from Storfjärden Fisk AB, Åland (Åland Islands) in Finland. The physical properties of the studied fuels.

FO contains saturated fatty acids (SAF) 18.1%; monounsaturated fatty acids (MUF) 43.9%; and polyunsaturated fatty acids (PUF) 37.4%. The extended measurement uncertainty for fatty acid was $\pm 1\%$. The fatty acid composition is related to the viscosity of the fuel. Fuel viscosity value decreases with the increase in the amount of unsaturated fatty acids (Chin & Velasquez, 2020; Deshmukh & Velasquez, 2012; Esteban et al., 2012).

The concentration of double carbon bonds (MUF or PUF) has also been found to affect carbon deposits formation in engines (Brun et al., 2019; Jayasinghe et al., 2012). It is stated that the key challenge for the feasibility of fish oil as a fuel is the recovery of the oil from the waste. A high PUF content decreases the thermal oxidation stability of the fish waste. This needs to be taken into consideration when managing the storage and transport of fish waste.

The acid number (AN) for FO was $2.09 \text{ mg KOH g}^{-1}$. This is substantially lower than the literature values for example, Brun et al. (2019) reported AN values of $1.725 \text{ mg KOH g}^{-1}$ for fish oils. The acids in bio-oils increase the corrosion risk and in the long term shorten the expected lifetime of certain engine components notably the fuel injection system. AN above $1.0 \text{ mg KOH g}^{-1}$ is considered detrimental to the engine. AN below 5 mg KOH g^{-1} is defined as not to increase the corrosion risk (Ollus & Juopila, 2009). Specifically fish oils are reported to have high AN due to the presence of water and PUF that are more susceptible to oxidation and free fatty acid formation (Chin & Velasquez et al., 2020). With an AN value well below 5 mg KOH g^{-1} the fish oil in this study does not pose a corrosion risk.

In literature, the kinematic viscosity value of crude fish oil has been measured at $28 \text{ mm}^2 \text{ s}^{-1}$ (Niemi et al., 2009), and that of DFO at $3 \text{ mm}^2 \text{ s}^{-1}$. FO's high viscosity hinders the production of fine fuel spray in a practical fuel nozzle. High viscosity modifies the droplet distribution due to the formation of larger droplets. Fuel viscosity increases sharply in cold conditions, which may cause restriction in fuel delivery that results in the reduction of the overall flow (Boesh, 2018). Adjusting the fish oil temperature can compensate for its high viscosity compared to traditional fuels (Bruun et al., 2012).

The density of this study was 900 kg m^{-3} . Fuel density affects the dispersion of the fuel injected into the cylinder. Higher density increases the compression ratio through the mass of fuel injected and the droplet diameter. The density directly impacts the injection timing and injection spray pattern. Increased density reduces fuel atomisation and

Table 3 Fuel properties

	Unit	DFO	Fish oil
Carbon content	wt-%	86.17	77.2
Hydrogen content	wt-%	13.71	15.1
Nitrogen content	wt-%	0.91	0.13
Sulphur content	mg kg ⁻¹	3.3	2.1
Ash content (775°C)	wt-%	< 0.00	< 0.010
Cetane number, IQT	-	54	*
LHV	MJ kg ⁻¹	43	37
Density at 15°C	kg m ⁻³	835	920
Acid number	mg KOH g ⁻¹	-	2.90
Kin. viscosity at 40 °C	mm ² s ⁻¹	3**	28**
Indie value	g 100 g ⁻¹	-	132
Water content	mg kg ⁻¹	< 0.09	0.9
Surface tension	mN m ⁻¹	28.5	33
Oxidation stability	h	-	0.86

*Fish oil was too viscous for measuring cetane number;
** literature value Niemi et al., 2009

mix it with air: this reduction is associated with high PM and NO_x emissions. According to a manufacturer of marine diesel engines, the density of fuel should be lower than 991 kg m⁻³ for diesel engines (Juoperi & Ollus, 2008; Jayasingh et al., 2012).

FO has a low viscosity measured by its cetane number (CN) by an ignition quality tester (IQT). In the study of Niemi et al. (2011) CN for crude fish oil was 49, which is not very low. In the current study, FO had a CN of 45 (NCAs manipulated to the ignition delay ID). A low CN increases ID, resulting in poor combustion and leading to a higher smoke emission (Hissa et al., 2012).

The LHV (lower heating value) of FO (37 J kg⁻¹) is substantially less than that of the DFO (43 J kg⁻¹) thus in raising the required fuel input to achieve the same energy output (Drenth et al., 2014). The presence of water in fish oil decreases its heating value. FO had high water content (over 9 mg g⁻¹). In engine use, water may cause corrosion of the equipment and containers (Adroit & Hawboldt, 2015; Bruu et al., 2019).

If the same volume of two fuels, with different fuel densities, are injected to an engine the fuel with higher density provides higher energy output. However, this occurs only if lower density values L(HV) do not differ greatly (Muntanen, 2014). In this study, the density of fish oil is significantly greater than that of diesel, the energy per injection is accordingly similar than based on LHV alone.

The surface tension of FO is 18% higher than that of diesel. Surface tension has a direct impact on the size of fuel droplets, so FO's higher surface tension might also contribute to an increase in its droplet diameters (Heywood, 2018).

Analysis Instruments

LabVIEW system-level software was used to collect the sensor data from the engine. The recorded variables were engine speed and torque, cylinder pressure and injection timing duration and quantity. A WinEM3 program provided by the engine manufacturer, GAO Power, controlled fuel injection according to load-speed requests. The basic settings of WinEEM3 were the same for all nozzles and fuels. Fig 1 is a schematic of the test bench setup.

A piezoelectric Kistler 6152C pressure sensor measured the in-cylinder pressure. The sensor was mounted on the head of the fuel cylinder. A charge amplifier filtered and amplified the signal which was then transmitted to a Kistler KIBOX combustion analyzer. The crankshaft position was recorded by a crank-angle encoder (Kistler 2614B1), which also outputs a crank angle signal with resolution of 0.1° CA by means of an optical sensor. The cylinder pressure data was averaged over 100 consecutive cycles to smooth irregular combustion. The averaged data were used to calculate the heat release rate (HRR).

The HRR and mass fraction burn rate (MFB) were calculated via AVL Co. nextos' data-processing platform, using the Thermodynamics2 macro. The macro used a calculation resolution of 0.2 °CA. The start of the calculation was set at -30 °CA. The data were filtered with the DigitalFilter macro and frequency of 20 Hz. For HRR results, the average values of in-cylinder pressure were calculated first. Thereafter, the macro was used to calculate HRR values. Finally, the HRR curve was filtered. In contrast, for MFB results, pressure data were first filtered, and then the macro was

used. The average values of 10 cycles were not used of the MFB results, establishing the standard deviation.

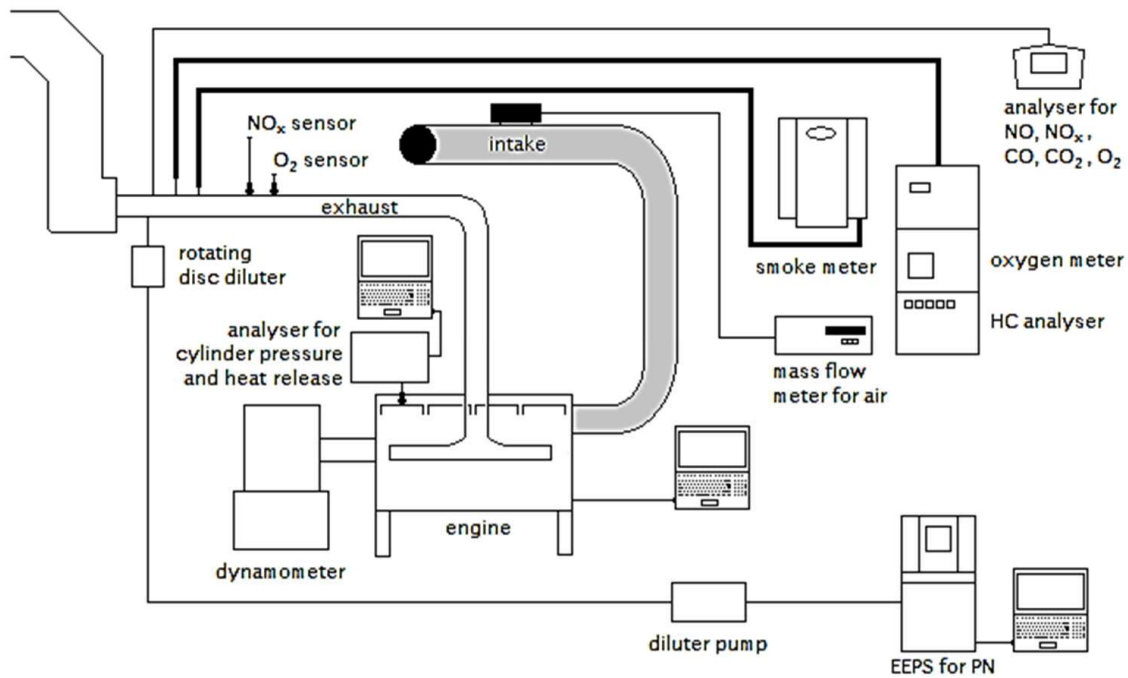


Figure Engine measurement setup.

The exhaust temperatures were recorded by K type thermocouples (NiCu NiAl). Air and exhaust pressures were determined by industrial transmitters. The engine air flow was measured by an ABB Sensyflo WFM700 P meter. Exhaust emissions were determined using the instruments listed in Table 4. The measured concentrations of gaseous emissions were used to calculate the brake specific emission according to the ISO 1878 standard (EN ISO 818-2:2008).

Table 4 Instruments of emission measurement

Parameter	Analyser	Technology	Accuracy*
CO	TSI CA-62 & CA-CALC	Electrochemical	0-100 ppm: $\pm 10\%$ 10000 ppm: $\pm 5\%$
O2	Siemens Oxymat 6	Paramagnetic	$\pm 0.25\%$
NO, NOx	TSI CA-62 & CA-CALC	Electrochemical	0-100 ppm: $\pm 10\%$ 10000 ppm: $\pm 5\%$
HC	J. UME 7	HFID	0-10,000 ppm $\pm 1\%$
Smoke	AVL 4S	Optical filter	$\pm 5\%$
Particle number	TSI EEP3090	Scatterometer	-

* Accuracy provided by the manufacturer.

An engine exhaust particle sizer (EEPS, model 3090 TSI Inc.) was used to determine the EPN with a particle size range of 560-560 nm. The exhaust sample was first diluted with ambient air by a rotational disc diluter (RDD - model MD19-E3, Matter Engineering AG), which had a constant dilution ratio of 50:1. Dilution air was kept at 5°C while the exhaust aerosol sample was conducted to the RDD. The diluted

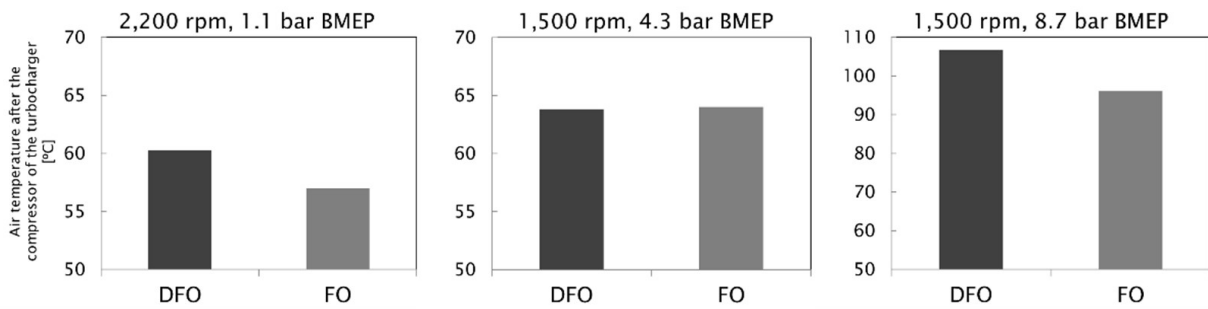


Figure 2 Air temperature after the compressor for the turbocharger.

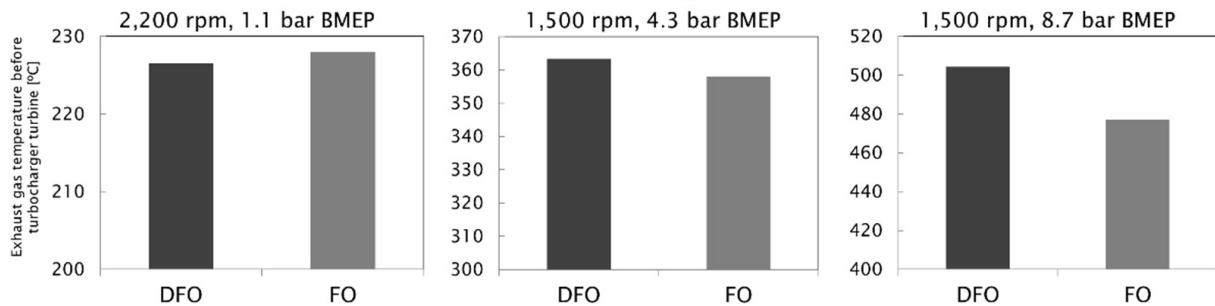


Figure 3 Exhaust gas temperature before the turbocharger turbine.

Injection parameters

For all test conditions, pilot and main injection were set before top dead centre (BTDC) and post injections occurred after top dead centre (ATDC). Exhaust timing and durations are shown in Table 6.

Table 6 Injection parameters for EO and FO

Fuel	Speed min ⁻¹	BMEP/ Load bar/%	Pilot injection (ATDC)		Main injection (ATDC)		Post injection (ATDC)	
			Start °CA	Duration °CA	Start °CA	Duration °CA	Start °CA	Duration °CA
DFO	2200	1.10	-1.8	4.6	-8	6.9	1.3	0
FO			-1.8	4.6	-8	7.6	1.3	0
DFO	1500	4.35	-1.2	3.9	-4.5	8.2	1.3	4.7
FO			-1.1	3.9	-4.4	9.6	1.5	4.7
DFO	1500	8.70	-8.7	3.5	-2.1	13	2.1	4.5
FO			-9.4	3.8	-2.5	16	2.2	3.4

Injection timing and durations were broadly similar with both fuels at 2200 min⁻¹, but the duration of the main injection was longer for FO because its lower heating value is less than DFO's.

At the load of 4.3 bar BMEP and engine speed 1500, pilot injection started 1 °CA earlier with DFO compared to FO. The main injection started at the same time with both fuels but injection duration with FO was longer than with DFO. FO's post injection started 2 °CA later than with DFO, most probably delayed due to FO's longer main injection duration. The duration of post injection was still similar for both fuels.

With FO at high load at 1,500 min^{-1} , pilot injection started earlier (9.4 CA BTDC) than with DFO (8.7 ° CA BTDC). An advanced injection may increase NOx emissions (Hosvet al., 2018; Buddin et al., 2018) in comparison with diesel. However, based on Heywood (2018) lean pilot systems shorten the duration of fuel by increasing in-cylinder temperatures for main injection. The main injection started 0.5 CA later with DFO and post injection started 1 CA earlier with DFO. The duration of the main injection was only slightly shorter with FO.

The volumetric amount of injected fuel was assumed to correlate to injection durations because fuel injection was controlled according to load/speed requests. The longer main injection duration is due to the higher viscosity and surface tension of FO, which increased the total injection duration due to decreased flow through the injector (Bae & Kim, 2016).

As expected, pilot injection duration increased when the engine load was reduced because the properties used especially at low loads to promote ignition, reduce ID and to smooth the increase of combustion pressure. Post injection is used to reduce particulate and soot emissions, primarily at lighter loads and low engine speeds (Heywood, 2018). This technique was observed at 1,500 min^{-1} where the post-injection duration did increase when the engine load was reduced. However, at an engine speed of 2,000 min^{-1} , post injection duration was a short pulse with both fuels.

Cylinder pressure

Injection timing primarily affects maximum cylinder pressure (MCP). However, the pressure also depends on the burned fuel fraction during the premixed combustion phase, and thus on the ignition delay (ID). ID is a period when injected fuel remains in the cylinder, atomizes and mixes with existing air but does not yet ignite. Chemical reactions start slowly and ignition occurs after the ID. ID has a direct effect on the heat release rate and an indirect impact on engine noise and exhaust gas emission formation (Aldhidhawi et al., 2017; Kuszewski, 2019). A long ID results in a rapid pressure increase in the combustion chamber when unburned fuel finally ignites. The rapid pressure increase leads to diesel knock, high soot emissions, malfunctions in engine operation and engine damages (CIMAC, 2014; Ogawa et al., 2018; Hissa et al., 2019; Kuszewski, 2019). A long ID and more fuel burn in the premix phase usually results in a temperature rise and high MCP (Hissa et al., 2019).

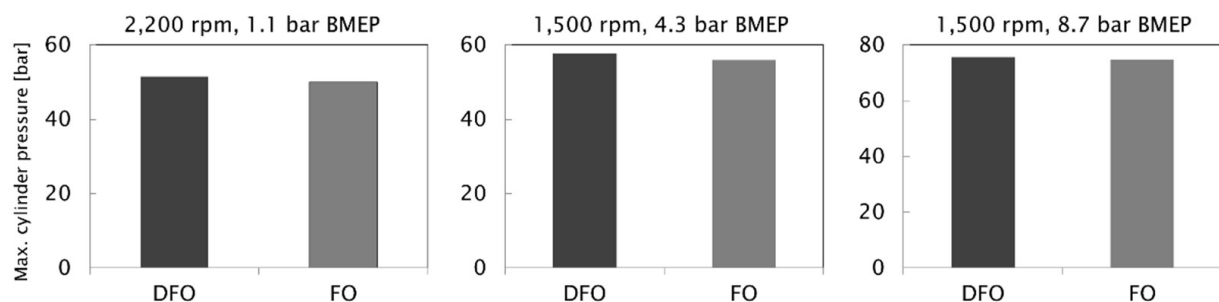


Figure 4: Maximum cylinder pressures at rated and intermediate speeds

Figure 4 shows that MCP values with DFO were slightly higher than with FO at all studied load points. The differences between DFO and FO increased with the load.

The average MCP values and their standard deviations of 100 consecutive cycles are given in Table 7.

Heat release rate (HRR)

Combustion starts with a rapid burn phase that lasts only a few CA degrees and produces the first spike in the HRR curve. It is followed by the main heat-release period with longer duration and a more rounded profile. The tail of the HRR curve is the remainder of the fuel's chemical energy released when burnt gases mix with excess air that was not involved in the main combustion (Heywood 2018) 5–7 show HRR curves for the studied fuels. A slight loss occurs at the beginning of each HRR curve due to the heat transfer in the liquid fuel of evaporating and mixing (Heywood 2018) 20.

Fig. 5 illustrates the HRR of two fuels at 2,200 min^{-1} and 1.1 bar BMEP. The FO curve indicates that its pilot did not ignite properly, so FO had a high crank HRR compared to FO. Of particular concern is the crank angle degrees later than DFO's.

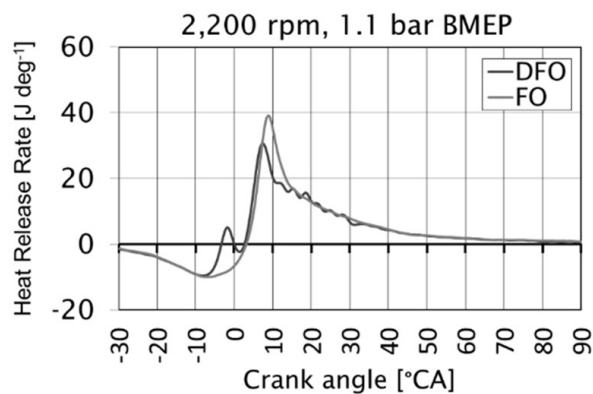


Figure 5 Heat release rate curves with FO and FO at 2,200 min^{-1} and 1.1 bar BMEP.

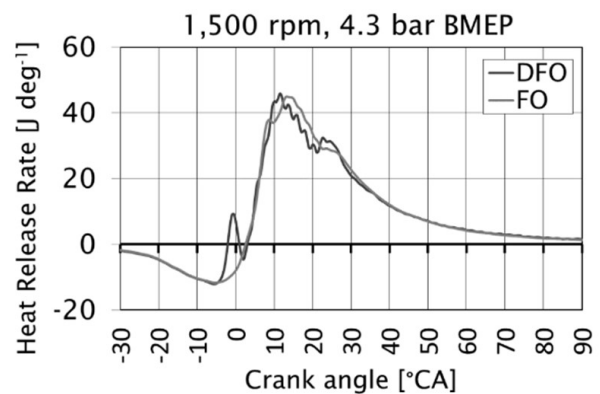


Figure 6 Heat release rate curves with FO and FO at 1,500 min^{-1} and 4.3 bar BMEP.

Fig. 6 depicts the two HRR curves at 1,500 min^{-1} and 4.3 bar BMEP. The HRR of DFO again shows a clear initial HRR peak and even an increase in the HRR at post-injection. In contrast, the FO curve shows unclear HRR peaks from either pilot or post-injection, and its general profile is more rounded than the DFO curve. Most likely, the high viscosity and surface tension of FO increased the droplet size of the fuel spray, impairing ignition. FO's lower CN would also increase ignition delay leading to retarded combustion (Bae & Kim, 2016; Hsueh et al., 2019). However, the lower compressibility and high expansion coefficient of FO may have accelerated the HRR of FO (Shah et al., 2013).

Table 7 Maximum cylinder pressures and standard deviations

Fuel	Speed min^{-1}	BMEP/Load bar %	Max. cylinder pressure (avg) bar	StDev (filtered)
DFO	2,200	1.1 bar	51	0.08
FO			50	0.05
DFO	1,500	4.3 bar	58	0.12
FO			56	0.07
DFO	1,000	8.7 bar	76	0.13
FO			75	0.06

Fig. 7 shows the two RR curves when the load was increased to 8.7 bar BMEP. Note that FO's RR peak from the pilot injection is clearly evident but is still seen later than that of DFO. FO shows no post-injection peak but is seen to be slightly faster than DFO in the later phase of combustion. Again, the high viscosity and surface tension of FO increased the size of fuel droplets, and the time required to evaporate the fuel droplets also increased.

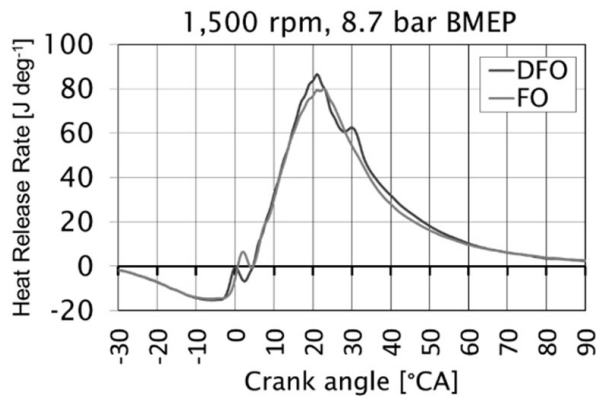


Figure 7 Heat release rate curves with FO and DFO at 1,500 rpm and 8.7 bar BMEP.

Mass fraction burned (MFB)

In the current study, the maximum compression pressure was at two degrees CA before the top dead centre at an engine speed of 1,000 rpm. The slight difference in measured results must be taken into consideration when the results are examined.

Table 8 presents mass fraction burned (MFB) values with the standard deviations. MFB values were very similar with both fuels. Only at MFB 90% at 2,200 rpm was there more than 1 °CA difference between them. Both fuels show MFB 5% at between 24 to 26 °CA at low loads and at 29 °CA at high load. Most probably, FO burns slightly more rapidly due to its oxygen content rather than a lower ignition.

Table 8 Mass fraction burned standard deviations

Fuel	BMEP/Speed bar/min ⁻¹	MFB 10 % °CA	StDev	MFB 50 % °CA	StDev	MFB 90 % °CA	StDev
DFO	11.2, 2000	12	0.41	24	0.75	65	4.3
FO		13	0.46	25	0.78	69	4.3
DFO	43.1, 5000	13	0.21	25	0.31	59	1.5
FO		14	0.17	26	0.24	58	1.4
DFO	87.1, 5000	18	0.17	29	0.26	58	1.2
FO		18	0.13	29	0.24	59	1.3

Combustion duration

Combustion duration (CD) can be defined either as the time interval between MFB 5% and MFB 90% (Fig. 8) or the time interval between MFB 10% and MFB 50% (Fig. 9).

The high viscosity and surface tension of FO generated larger droplets. Large droplets, again, require more time to evaporate and burn (Heywood, 1988). Figure 8 shows that at 11 bar BMEP at 2,000 rpm and at 8.7 bar BMEP at 1,000 rpm, CD 5–90% was longer with FO than with DFO. However, FO's high oxygen content may have improved combustion by decreasing combustion duration. In the CD 10–50% as shown in Figure 9, FO compared to DFO (Fig. 9).

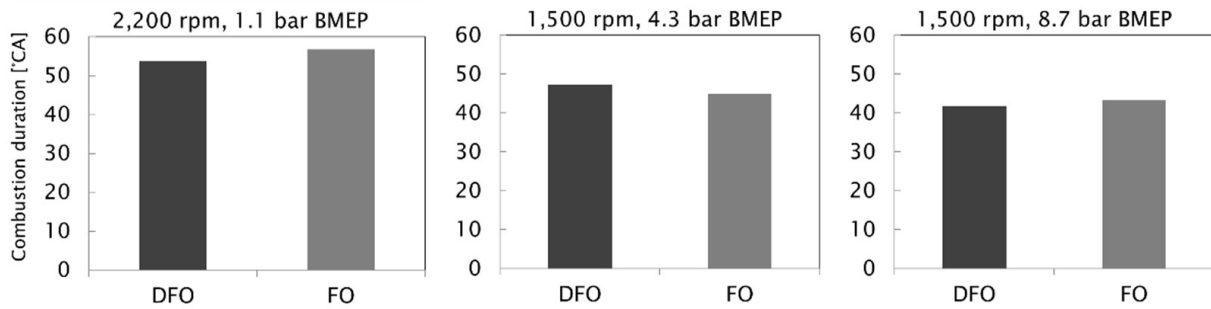


Figure 8 Combustion duration (°CA) at all engine loads, determined as crank angle between MFB 5 % and MFB 90 %.

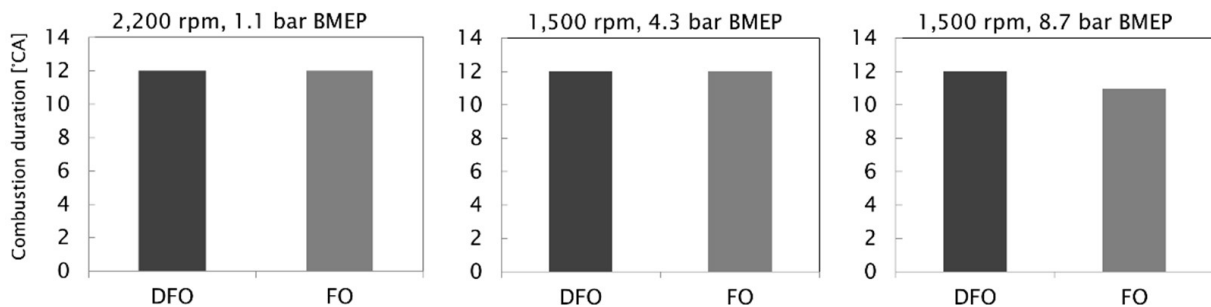


Figure 9 Combustion duration (°CA) at all engine loads, determined as crank angle between MFB 10 % and MFB 50 %.

Gaseous emissions, smoke and total particulate numbers (TPN)

Fig. 10 illustrates the brake-specific emission of NOx, CO, and HC. The smoke numbers are shown in Fig. 11 and total particulate number (TPN) are plotted in Fig. 2.1.

In broad terms, combustion of FO generated more NOx, CO and HC than when using DFO, but the difference between the fuels diminished when the engine load was increased.

At 1,500 rpm, FO emitted very similar NOx emissions of 2.6 g/kWh at both loads. The result at the higher speed was 4.0 g/kWh. Compared with DFO, FO increased NOx by 23% at lower load at 1500 rpm while at high load the difference was only 2%. However, at 2,200 rpm and 1.1 bar BMEP, DFO showed 12% higher NOx than FO. The higher NOx for FO may be due to the presence of molecular oxygen that promotes the oxidation of nitrogen (Shah and et al., 2011). Ignition retarded premixed combustion may also have affected NOx formation (Satyanarayana & Muraleedhara 2002).

Less excess air and higher combustion temperature promoted NOx formation. As seen in Fig. 10, increased engine load improved fuel-air mixture and oxidation. The improved mixture led to a reduction in CO, HC and smoke as engine load was increased. It is most likely that inadequate spray formation of FO caused higher NOx, CO and HC emissions (Niemi et al., 2009).

FO produced more CO than DFO at both loads. At 1,500 rpm, CO emissions were 2.2 g/kWh for FO and 0.5 g/kWh for DFO. At higher load, FO emitted 0.9 g/kWh and DFO 0.4 g/kWh. At 2,200 rpm, CO emissions were very high at 2.3 g/kWh while DFO generated approximately 3 g/kWh. FO's high CO emission indicates poor fuel-air mixing and incomplete combustion especially at very

low loads. Ollu s& Ju o pi (2007) reported that liquid bio fuel (LBF) increased CO emission in a medium speed diesel engine and the reason for increase may be that the fuel has less oxygen content than because of combustion process. Satyanarayana & Muralidharan (2012) reported that atomized and compressed combustion with sun heated palm oil of high viscosity was used as engine fuel. However, reduced CO formation was reported when the neat oil was preheated.

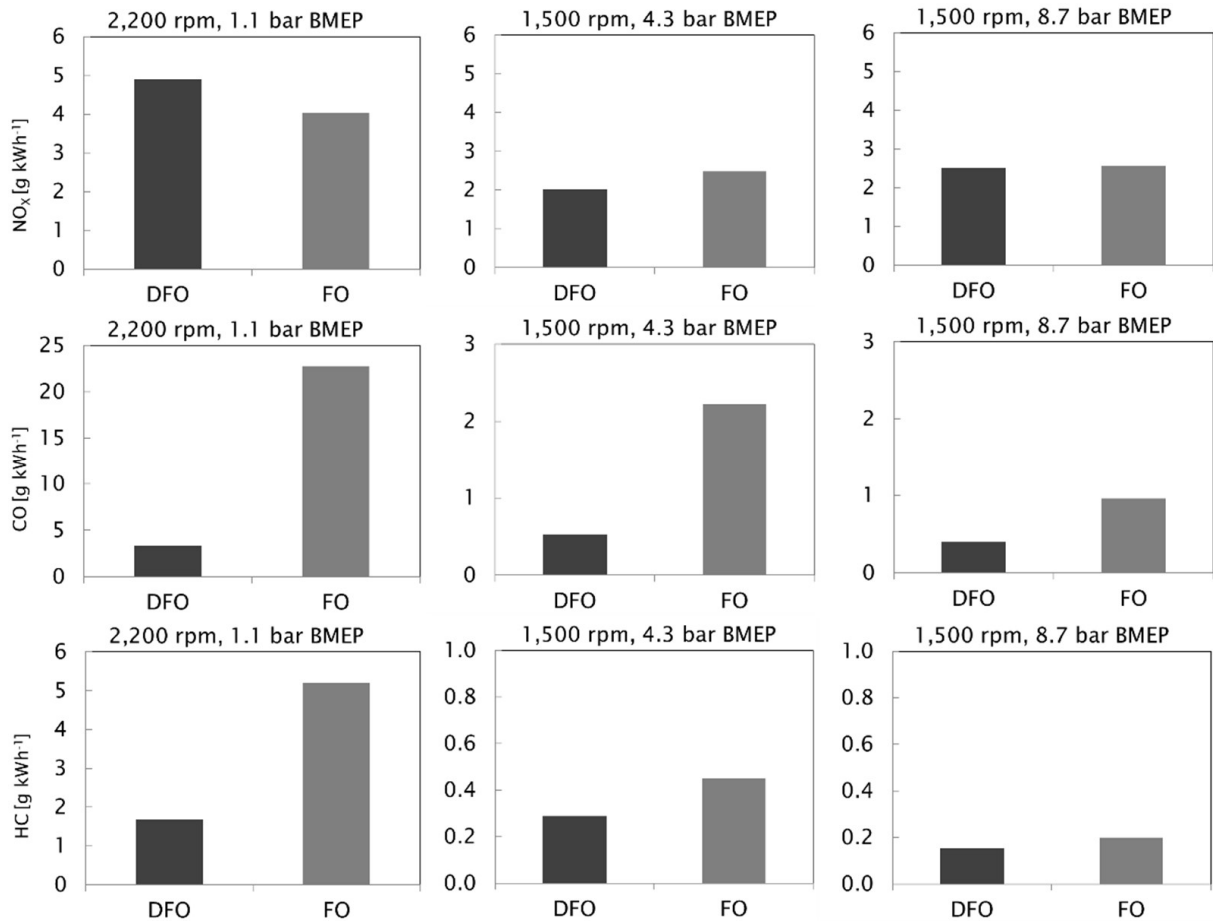


Figure 10 Brake specific emission of NO_x, CO and HC for FO and DFO.

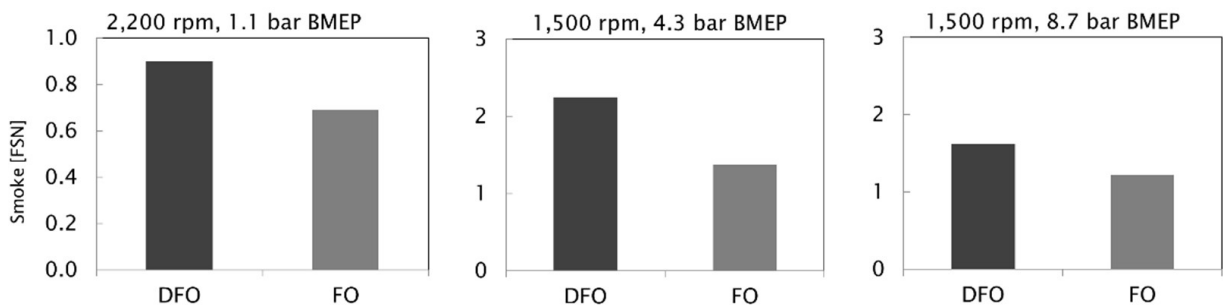


Figure 11 Smoke numbers for FO and DFO.

HC emissions also increased clearly when a FO was burned instead of DFO, compared to the air-fuel ratio and combustion with FO was inferior at the rather low loads. At 1,500 min⁻¹ at the low load HC was 0.4 kWh⁻¹ for FO and 0.2 kWh⁻¹ for DFO. At high load, FO emitted 0.2 kWh⁻¹ and DFO 0.15 kWh⁻¹. At 2,200 min⁻¹, HC was again higher for FO at 5.2 kWh⁻¹ while DFO generated 1.7 kWh⁻¹. Our HC and CO results correspond with those of Wang (2010) and 2011 study of preheated neat conventional diesel engine and tested heavy CO and HC emissions compared to DFO. The reason given as the incomplete combustion of the component by Satyanarayana & Mualeedharan (2012) also observed an increase in HC emissions with neat vegetable oils compared to DFO. Turunen & Niemi (2009) experimentally showed HC emission at low engine loads compared to high load and also to lean mixture areas, where fuel-air mixture ignites and burns poorly. Slow fuel injection speed may also increase HC emission. Another clear source for HC emissions in diesel engine is the sac inside an injection nozzle. The sac stores fuel after injection and fuel evaporates slowly through nozzle holes and is not participated to combustion (Turunen & Niemi, 2002).

Contrary to CO and HC, smoke decreased at all loads with FO. At speed of 2,200 min⁻¹ and 1.1 bar BMEP, FO generated 0.7 FSN, whereas the smoke reading for DFO was 0 FSN. At 4.3 bar BMEP at 1,500 min⁻¹, FO smoke number was 1.4, and at high load 1.6 FSN. The corresponding values for DFO were 2.2 FSN and 1.6 FSN. Niemi et al. (2009) also observed improved smoke for rapeseed oil compared to DFO in a high-speed diesel engine, concluding that the most probable reason was the high oxygen content of biofuels. High oxygenated fuels produce less smoke due to higher flame temperature and lower radiative heat losses in the cylinder (Chauhan et al., 2010; Shukla et al., 2013).

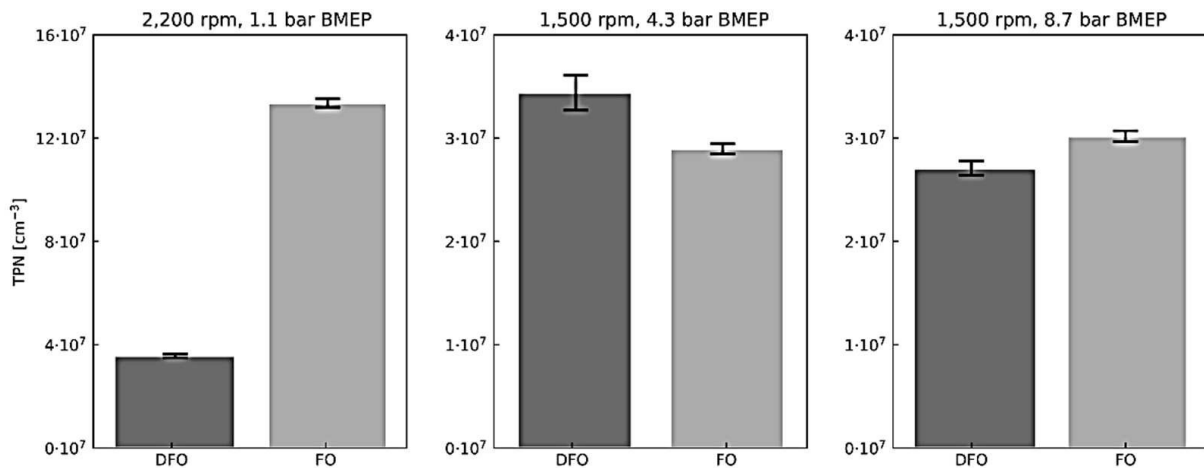


Figure 12 TPN emissions with FO and DFO at all loads and speeds

Figure 12 shows the TPN emissions with FO and baseline DFO at all loads and speeds. Each bar shows the TPN mean and error bars represent the standard deviation for HN during the measurement period of three minutes. At 4.3 bar BMEP load at 1,500 min⁻¹, FO reduced TPN, but at other loads the order of fuels was the opposite. Compared to the DFO baseline, FO emitted more particles at 8.7 bar BMEP at 1,500 min⁻¹. The greatest difference was at low load at 2,200 min⁻¹, where FO's TPN was 3.7 times higher.

than DFO's. The HC and CO emissions improve at all load conditions with FO compared to DFO although the smoke is high. A reduction in PM emissions can be expected if the sulphur content, density, viscosity and carbon-to-hydrogen ratio of the fuel are reduced (Nabati et al., 2012).

On the contrary, soot emissions may increase along with fuel viscosity because high viscosity can lead to less favourable fuel atomisation and hence combustion may not be completed (Kegl et al., 2013; Nabati et al., 2012). High fuel density may inhibit fuel spray formation during fuel injection potentially causing incomplete fuel burning and high emissions (Hissa et al., 2018). In this study FO's sulphur content of 21.5 gm/kg was less than DFO's sulphur content of 35.5 gm/kg. Density at 15°C was higher for FO (920 kg/m³) compared to that of DFO (835 kg/m³). Moreover, the FO had high kinematic viscosity (28 mm²/s²) than DFO (3 mm²/s²). The carbon-to-hydrogen ratio of FO (0.56) was also higher than that of DFO (0.5). With the exception of sulphur, the differences in the density, viscosity and carbon-to-hydrogen ratio that were all higher for FO may explain the high TPN of FO at low load at rated speed and at high load at intermediate speed.

Unlike our study several other studies have shown improvement in emission and performance with crude bio-oils. Preheating the bio-oil has reduced exhaust emissions and fuel consumption and increased the engine power output to within the viscosity of neat bio-oils to a level, comparable with DFO (Hosain et al., 2019). The high viscosity can also be lowered by blending bio-oil with lower viscosity fuel or processing through the transesterification method to produce biodiesel (Chauhan et al., 2010). However, a manufacturer of large engines does not recommend blending crude bio-oils (Ollu & Juoperi, 2010). Further progress for FO should include optimisation of injectors and injection setting and preheating of fuel to improve fuel-air mixing, decrease ID and improve combustion.

CONCLUSIONS

Crude fish oil (FO) at room temperature was investigated via high speed, four-stroke diesel engine. The engine was turbocharged, naturally aspirated and equipped with a common rail injection system and 6-hole high flow rate injectors. Measurements were made at two load conditions at intermediate speed and at a low load at rated speed. The results were compared to those of DFO.

The FO was classified as waste and could not safely be used in, for example, food production. It is produced from molal waste sources at moderate cost. Generally, these kinds of renewable fuels are seen as one alternative for fossil fuels with a targeting at reduction of greenhouse gas emissions.

Based on the results, the following conclusions could be drawn:

- The high viscosity and surface tension of FO inhibited fuel spray formation and air-fuel mixing.
- This and the low cetane number of FO, in raised ignition delay and retarded ignition resulting in incomplete combustion.
- Consequently, CO, NO_x, and HC emissions increased compared with DFO.
- Smoke, however, decreased with FO.

- Except for the total, the minimum, the maximum and the average compared to DFO. Density, viscosity, and carbon-to-hydrogen ratio were all high for FO, so the differences may explain the high HCN with FO.

- Optimisation of the injection system and preheating of the fuel are the main avenues to improve the engine performance and emissions with FO. Optimal ratio of very old waste must probably be also developed to avoid the high HCN.

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