Application of Waste Biomass Pyrolysis Oil in a Direct Injection Diesel Engine: For a Small Scale Non-Grid Electrification

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Abstract: The population which could not access to electricity was around 1.2 billion in 2010 and is distributed in many low developing countries. With the increase in the population and the economic growth in those countries, waste generation is growing rapid especially for the organic and the plastic, and the uncontrolled waste disposal is becoming more serious issues to manage it. The interest on waste to energy is growing by the above drivers. This research was carried out for aiming to the real world adaption at the minimum cost of the pyrolysis oil from waste biomass in a diesel engine, mainly for electricity generation. The proposal of the appropriate adaptable blend ratio was the major scope rather than the optimization of the engine parameters. For the sake of it, the pyrolysis oil of the waste biomass was produced from a gasification pilot plant in Japan and blended with biodiesel at minimum effort. A small single cylinder diesel engine (direct injection) was used for the experiment with regard to full load power-output, exhaust emissions and fuel consumption.

Key words: Waste biomass, pyrolysis, blend, direct injection diesel engine, exhaust emissions.

1. Introduction

Low developing countries such as African, South Central Asia and Southeast Asia/Pacific are facing with the development of economic growth along with continuous population increase [1]. In accordance to it, the uncontrolled waste disposal is increasing more and more [2] while in 2010, one of five people still remained as non-electricity [3]. This situation elevates the interest in waste to energy. The United Nations issued the outlook report on waste to energy in related to the global status, technical and economic considerations and market trends [4]. In the report, the methodologies producing energy from waste was

categorized as the thermo-chemical conversion, the bio-chemical conversion and the chemical conversion (esterification).

Considering the economic and the waste generation situation in the developing regions, the pyrolysis process of the organic and the plastic waste has the highest potential as most appropriate resources for non-grid electrification system in terms of the scale, the technology level to the readiness of the availability and the cost. Oil and synthetic gas are mainly produced through the pyrolysis process in small scale and on-site, and can be delivered to internal combustion engines to generate the electricity. The property of the oils from the pyrolysis is highly dependent on the raw material and the pyrolytic conditions. Diesel engines are most suitable in view of the conversion efficiency and the

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stable engine operation.

From the view point of effective usage of energy resources, much research concerning the application of pyrolysis oil obtained from biomass (bio-oil) have been carried out. Application and research results in many institutes to adapt bio-oil derived from woody biomass to compression ignition engines were reviewed [5]. Solantausta, et al. [6] ran a diesel engine fueled by bio-oil from hardwood feedstock and reported that, combustion duration of bio-oil was shorter than that of diesel fuel and found damage in engine systems such as cocked injection nozzle and combustion chamber deposits. Chiaramonti, et al. [7] operated a single cylinder direct injection diesel engine by using emulsified pyrolysis pine oil with diesel fuel and reported engine troubles in injector nozzle and fuel supply pump. Eucalyptus/diesel blends were tested in a single cylinder direct injection diesel engine and the smoke and THC (total hydrocarbon carbon) emissions were reduced according to increasing the ratio of blending up to 50%, while NO_x (nitrogen oxides) increased due to longer ignition delay which enlarged the portion of the premixed combustion [8]. Diesel spray was investigated for coconut oil/diesel blends and averaged liquid particle diameter in the spray describing the quality of air-fuel mixture formation, the smaller the better, increased by raising the blend ratio. Nevertheless, compared with diesel fuel, the smoke emission was reduced by 12% due to the presence of oxygen in coconut oil while NO_x decreased by 36% caused by shortened ignition delay [9]. Lee [10] reported on the performance of a dual-injection diesel engine fueled with diesel & biodiesel for pilot injection and sawdust pyrolysis oil-ethanol blend (30%-70% by mass) for main injection. The high volume of the ethanol was necessary to prevent from the polymerization. Despite the poor ignition quality of the blend oil, the ignition delay was not differed much from that of diesel fuel by the pilot injection. THC and CO emissions were increased, and NO_x and soot emissions were decreased due to the high content of

water and oxygen in the bio-oil/methanol blend. Prakash, et al. [11] experimentally studied on the engine test fueled by pyrolysis oil and diesel emulsions. The softwood obtained from the packing container boxes was pyrolyzed and the bio-oil was emulsified with diesel fuel by the surfactant Polysorbate-20. Three percentage of surfactant in volume was used to emulsify 10% of bio-oil with 90% of diesel fuel. A small single cylinder direct injection diesel engine was run with the emulsified oil. The exhaust emissions resulted in the lower NO emission and the higher CO/THC emissions for the emulsified oil than those of diesel fuel and he inferred that, the result might be due to the more water content in the emulsified oil. The water content might act to reduce the combustion temperature (less NO emission) and make the volatility worse (poorer atomization and higher CO emission) and lead to a longer ignition delay and incomplete combustion. They also investigated on the performance and the emission in the same diesel engine operated with wood pyrolysis oil/biodiesel emulsions. The wood bio-oil produced via a slow pyrolysis process and the methyl ester of *jatropha* (biodiesel from *jatropha*) produced by the transesterification process was emulsified by the surfactant of span-80. The blend ratio was 5%, 10% and 15% of the bio-oil and 95%, 90% and 85% of the JME (*jatropha* methyl ester), respectively by adding 2% surfactant by volume. NO and CO emissions were higher and HC (hydrocarbon carbon) and smoke emissions were lower than the case of the diesel fuel at the full load [12].

Those have been highly evaluated academically but limited in the point of the adaption into the real world. This research was carried out aiming to the real world adaption at the minimum cost of the pyrolysis oil in a diesel engine, mainly for electricity generation. Verifying the effectiveness of the pyrolysis oils in adaption to the diesel engine was the major scope. For this, the possible blending ratio was determined without any engine modification through the specified evaluation criteria. For the sake of it, pyrolysis oil

derived from Japanese cedar was produced in the gasification pilot plant in Japan. Engine operation points were picked up according to the US EPA (Environmental Protection Agency) standard and the specific gas emissions were evaluated in addition to the full load performance and the specific fuel consumption.

2. Materials and Methods

2.1 Waste Biomass Pyrolysis Oil

Bio-oil of Japanese cedar was produced using the pilot-scale gasification plant [13]. The bio-oil throughput which is available to run the engine was around 4.5% of the total output volume and its maximum output is the synthetic gas by around 68.8% [14]. The raw bio-oil obtained from Japanese cedar had a higher kinematic viscosity than diesel fuel as shown in Table 1, and many residues which were mainly composed of viewable solid particles and highly dense liquid conglomerate. Those are some of the reasons to make its direct adaptation to diesel engines difficult, due to unfavorable and unpredictable troubles in engine systems.

It was shown that, the bio-oil could not be permanently mixed with diesel fuel and the fuel delivery system was irreversibly failed from the combustion test fueled by the blend oil [13]. The upgrading treatment of bio-oil needs complicate processes and suffers from low product yield in addition to high cost. The raw bio-oil had been successfully mixed with biodiesel by adding methanol and there was no phase separation for the given observation [14].

2.2 Test Fuels

According to the methodology, the bio-oil was mixed with biodiesel and the blend fraction of the bio-oil to biodiesel was 50%, 30% and 10% in volume. The chemical and the physical property of the blend oils, biodiesel and diesel fuel, are respectively shown in Table 1. The density of the blend oils slightly

dropped but still out of the standards of diesel fuel (JIS (Japanese industrial standards) K2204 2007, 0.86 in max.) (China GB19147 2009, 0.81-0.85). The kinematic viscosity of the blend oils drastically decreased and positioned nearly close to the maximum number of (ASTM (American Society for Testing and Materials) 975, 4.1 in max. for No. 2-D) (EN590 2013, 4.5 in max.). There was a bit drop in the oxygen content but the water content was decreased in half of the raw pyrolysis oil. The heat value was recovered up to 77% of that of diesel even for the 50% fraction of the raw bio-oil whose heat value was half of the diesel.

2.3 Test Engine

For the engine test, a single cylinder direct fuel injection diesel engine was used and Table 2 shows the engine specification [15, 16]. The engine was very small and turned out a low engine output. The combustion chamber shape was a re-entrant type. The fuel injection timing was $17.5 \pm 0.5^\circ$ BTDC (before top dead center) CA (crank angle) and the injection fuel pressure was 20 MPa, built by the mechanical fuel delivery pump for all tests.

Fig. 1 shows the experimental equipment for the engine test. It consisted of a diesel engine, measurement devices and a dynamometer which controlled the engine operation at a certain constant engine load and speed. The in-cylinder pressure data were instantaneously measured by a crystal pressure transducer (KISTLER 6053B). THC and the smoke concentrations in the exhaust gas were measured by the HFID (HORIBA MEXA-1170HFID) and by the opacity type smoke meter (HORIBA MEXA-600S), respectively. CO, CO₂, NO_x and SO₂ emissions were measured by the NDIR (nondispersive infrared sensor) type exhaust gas analyzer (Iwatadengyo Co., Ltd FAST-3100). The intake air temperature and the exhaust gas temperature were measured by K-type thermocouples with the diameter of 1.0 mm. The fuel consumption was measured by using the electric

Table 1 Properties of the test oils.

Chemical						
% wt./wt.	Diesel (JIS2)	Bio-diesel	Bio-oil	10% blend	30% blend	50% blend
C	85.04	83.21	50.8	75.57	74.15	73.90
H	13.55	11.38	7.9	11.25	11.14	11.00
N	0	0.1	0.2	0.1	0	0.1
S	0	0	0	0	0	0
O	0	4.88	18.92	14.01	15.21	16.22
Physical						
Density (kg/cm ³) at 15 °C	0.830	0.861	1.055	0.873	0.903	0.939
Kinematic viscosity (mm ² /s) at 40 °C	2.744	3.800	7.032	3.911	4.157	4.222
Water content (% wt./wt.)	0	0	18.8	6.37	7.53	8.20
Low heat value (MJ/kg)	45.0	41.0	22.4	36.4	35.6	35.1

Table 2 Test engine specification.

Engine specification	
Cylinder number	1
Bore × stroke	70 mm × 57 mm
Displacement volume	219 cm ³
Compression ratio	20.6
Aspiration type	Natural
Rated power	3 kW/3,600 rpm
Combustion chamber	Re-entrant
Injection pressure	20 MPa
Injection timing	17.5 ± 0.5° BTDC
Number of injection hole	4
Diameter of injection hole	0.22 mm
Diesel fuel spray angle	95°

weighing instruments. All measurement data were continuously processed at the same time in the data collection unit (KEYENCE NR-HA08 and NR-TH08) and a personal computer.

At first, the test run fueled by the diesel was conducted in order to check whether the test conditions such as the engine performance and the measurement devices were properly prepared or not. Finishing the test fueled by the objective pyrolysis oils, the engine fueled by the diesel again operated to confirm that, there was no damage or change in the engine hardware by running with the objective pyrolysis oils.

2.4 Test Engine Operation Points

The engine speed and the torque were specified based on the EPA regulation [17] as depicted in Fig. 2.

The 100% power output at the intermediate and the rated engine speed was controlled by conducting the full delivery of the fuel in the injection system for all tested fuels. It means that, no additional throttle opening was done to compensate the full load engine output depending on the fuel types. All powers in the partial load operation points were controlled to produce the output powers which were designated to the mode points. When evaluating the experimental result, R25 (25% load at the rated speed, R25) data were interpolated between 50% and 10% because the linearity was shown to be enough. The engine speed at the rated point was slightly reduced by taking into account of the uncertainty from engine running fueled by the blend oils. As mentioned above, the throttle position was set by full opening position, and this led to the maximum power at the rated point to be slightly higher than that of the engine specification shown in Table 2.

3 Results and Discussion

3.1 Full Load Performance

Fig. 3 shows the maximum reachable torque for all the test fuels. The full load performance was evaluated upon 100% diesel which is expressed as the red circle in the blend ratio 0% point on the horizontal axis, biodiesel which was designated as 0% on the horizontal axis in the black circle and three blend oils, the 10%, 30% and 50% blending ratios. For all the full

Fig. 1 Experimental engine apparatus (FTIR: Fourier transform infrared spectrometry).

Fig. 2 Engine test operation points.

load tests, the throttle position was fully opened. The maximum reachable torque was almost the same excepting for the case of the 10% blend and the 50% blend at 2,450 rpm. The standard deviation of the engine torque went slightly worse for the 50% blend at 2,450 rpm than other test cases but the engine was stably operated even for the 50% blend.

For the analysis of the torque decrease, the injection amount per single combustion is plotted in Fig. 4. The injection amount increased with the elevation of the blend ratio due to the increase of the density. In Fig. 5, the tendency is re-illustrated showing the relationship between the density and the injection amount. In overall, the injection amount almost linearly increased with the rise of the density excepting for the 10% blend and the 50% blend in the case of 2,450 rpm. This means that, the torque drop was caused by the

injection amount drop. There might be two possible reasons, one is that, there was some trouble in the linkage system between the control dial in the engine control system and the lever in the fuel delivery pump such as loosening [18] and another is that, the injection amount deteriorated due to the oil property which was caused by heterogenous blending.

Fig. 3 (a) The maximum reachable torque and (b) the standard deviation of the torque as a function of the blend ratio (0% red: diesel, 0% black: biodiesel).

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Fig. 4 Injection amount per single combustion as a function of the blend ratio (0% red: diesel; 0% black: biodiesel).

Fig. 5 Injection amount per single combustion as a function of the density of the oils.

For more detail investigation, combustion analysis [19] was done. Figs. 6 and 7 show the cylinder pressure and the rate of the heat release. The ignition timing was slightly delayed with increasing the blend ration up to 30%, and for the 50% blend, the ignition delay became more significant. With the enlarged ignition delay, the rate of the premixed combustion peaked higher. This trend was in detail summarized in Fig. 8. In overall, the ignition timing became delayed and the fraction of the premixed combustion phase increased with the increase of the blend ratio. For the blend 10%, the ignition delay was between the diesel and the blend 30% for the two engine speeds, and the fraction of the premixed combustion showed the same behavior. This means that, the start of injection and the injection rate was correctly operated. Thus, it can be said that, the torque drop was caused by injection amount due to the former reason, mechanical linkage loosening. In the case of the torque drop of the blend 50% at 2,450 rpm, the ignition delay seems to be more

Fig. 6 Cylinder pressure and rate of heat release at I100, 2,450 rpm (WBO: waste bio oil).

Fig. 7 Cylinder pressure and rate of heat release at R100, 3,500 rpm.

Fig. 8 Ignition delay and fraction of premixed combustion phase as a function of the density of the oils (0%: diesel).

excessively retarded and the fraction of premixed combustion phase did not increase with the enlarged ignition. This means that, there might be the deterioration in the injection rate due the latter reason.

In summary, the full load power was almost the same even with the increase of the blend ratio. With the rise of the blend ratio, the heat value per unit mass decreases as shown in Table 1. But the total amount of heat value per single combustion was compensated by the increased injection amount due to the increased injection rate caused by the higher density. Fig. 9 shows the simple calculation by multiplying the injection amount shown in Fig. 5 with the heat value listed in Table 1.

3.2 Ignition Delay

In Fig. 10, the ignition delay time and the fraction of the premixed combustion phase [19] are summarized. The ignition timing was delayed with the increment of the blend ratio [20-23] and it means that, the higher the blend ratio is, the longer the time is required [24].

Fig. 9 Heat amount per single combustion as a function of the blend ratio (0% red: diesel; 0% black: biodiesel).

The longer ignition delay time led to enlarge the fraction of the premixed combustion phase with the increase of the blend ratio. The tendency was more significant for the 50% blend. The ignition delay became slightly shorten at 3,500 rpm because of the elevation of the gas temperature at the end of the compression stroke as depicted in Fig. 11, due to the higher compression intensity by the increased engine speed.

3.3 Exhaust Emissions

3.3.1 NO_x Emission

In Fig. 12, NO_x emission is plotted in the form of the concentration at 2,450 rpm and at 3,500 rpm. With the rise of the blend ratio, the NO_x emission decreased for the case of 2,450 rpm and this tendency was clear for R10 but diminished with the load increase at 3,500 rpm [21, 22]. NO_x emission tends to increase with higher premixed combustion rate which leads to the elevation of the flame temperature [25]. But the NO_x emission went to the opposite direction or showed constant as shown in the top picture of Fig. 13 (normalized ones for the diesel as 100%). The NO_x emission might be affected by the water content in the blend oils as shown in the bottom picture [26-31]. For the case of 3,500 rpm, this NO_x reduction by the increase of the water content was not observed.

3.3.2 CO Emission

Fig. 14 shows the CO emission as a function of the blend ratio. The increase of the blend ratio resulted in a higher CO emission for both engine speeds.

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Fig. 10 Ignition delay and the fraction of premixed combustion phase as a function of the blend ratio at 2,450 rpm and 3,500 rpm (0%: diesel).

Fig. 11 Spatially averaged temperature at motoring.

Fig. 12 NO_x emission in the form of the concentration (ppm): (a) at 2,450 rpm, (b) at 3,500 rpm (0%: diesel).

Especially, in R10, the CO emission turned out twice higher for the 30% blend than that for diesel and around four times higher for the 50% blend due to the misfire which was led by the enlarged ignition delay.

Fig. 15 shows the CO emission as a function of the fraction of the premixed combustion phase and the water content in the blend oils. Even though the sensitivity according to engine loads is different, the correlation was found. In case of the oxygenated oil, plenty of OH (hydroxyl) radical can promote the

Fig. 13 NO_x emission variation as a function of (a) the fraction of the premixed combustion and (b) the water content in the oils.

Fig. 14 CO emission in the form of the concentration (ppm) and the specific emission (g/kWh): upper two at 2,450 rpm, lower two at 3,500 rpm (0%: diesel).

Fig. 15 CO emission as a function of the fraction of (a) the premixed combustion and (b) the water content in the oils.

oxidation of CO. But it resulted in the opposite direction. It may be inferred as followed. For the oxidation of CO to CO₂, it needs to keep the temperature in the combustion field to be higher and to be longer enough [31]. The higher the premixed combustion fraction is, the shorter the high temperature duration enough for the CO oxidation is. In addition to that, the higher water content might contribute to reduce the combustion temperature [5, 32].

3.3.3 Smoke Emission

The smoke emission is illustrated in Fig. 16, where for the 10% blend, the number of smoke is too high and might be erroneous when the measurement was carried out. The smoke opacity generally became higher for both engine speeds by the increase of the blend ratio. The smoke opacity for R50 in the 50% blend was 100%, and it might be the result of the accumulation of the smoke which was produced in the previous misfired engine operation for R10 as described in the CO emission. The lambda was

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Fig. 16 Smoke emission in the form of the opacity (%): (a) at 2,450 rpm, (b) at 3,500 rpm (0%: diesel).

calculated based on the injection amount, the air mass amount and the stoichiometric air-fuel ratio where the oxygen amount in the bio-oil was taken into account in the calculation. For the blend oil, the injection amount per one stroke was elevated in order to compensate the lower heat value to keep a designated engine torque at the part loads and increased by the higher density at the full load. And this made the lambda of the blend oils to be gradually reduced as depicted in Fig. 17.

In Fig. 18, the smoke opacity was re-plotted as functions of the lambda and the kinematic viscosity and showed a certain correlation especially in high load. In addition to the effectiveness of the lambda, the spray condition might affect the smoke opacity. The viscosity and the surface extension of the bio-oil are higher than those of the diesel and thus lead to the poorer atomization [33, 34]. The spray had been investigated for the blend oil of pine biomass oil with diglyme (diethylene glycol dimethyl ether, cetane number ranges from 112 to 130) by using the injector (4 holes, 0.28 mm hole diameter, 210 bar opening pressure) and the break-up length and the cone angle

Fig. 17 Lambda: (a) at 2,450 rpm, (b) at 3,500 rpm (0: 100% diesel).

fuel [35]. The SMD observed became higher with the increase of the kinematic viscosity [36]. The poorer spray condition might play in some extent role to increase the smoke emission even under the existence of oxygen in the blend oil.

3.3.4 THC Emission

The THC emission is depicted in Fig. 19. The THC emission increased up to the 30% blend ratio at 2,450 rpm. At 3,500 rpm, the THC emission was higher for the blend oil in the low load range. At R10, the THC emission of the 50% blend drastically increased around double magnitude and the engine was not possible to keep its stable operation by the intermittent misfire. It was because that, the ignition timing was too much delayed by the heavier blending and thus it made the firing difficult. The poorer spray condition might be one reason.

3.3.5 Exhaust Emissions of EPA Mode

The EPA mode exhaust emissions were evaluated with regard to C1-8 mode for the non-road mobile vehicle and D2-5 mode for the gen-sets.

Fig. 19 THC emission in the form of the concentration (ppm): (a) at 2,450 rpm, (b) at 3,500 rpm (0%: diesel).

In the case of C1-8 mode, the NO_x + THC emission was placed around the regulation limit and slightly increased with the increase of the blend ratio and this was caused by the increase of the THC emission. The NO_x + THC emission of D-5 mode for the diesel was almost the same level with those of the 10% and the 30% blends and went slightly over the limit. The test engine was not new one and had been run with various non regulated fuels such as bio-oils and other pyrolysis oils. All emission measurement devices were not fully pre-conditioned for the engine test such as official emission certification test. Considering such conditions, the NO_x + THC emission result can be acceptable. But NO_x + THC emission of the 50% blend in D2-5 mode became worse than the diesel due to the explosive THC emission caused by the misfire in R10 as shown in Fig. 20.

Fig. 21 illustrates the result of the mode CO and PM (particulate matters) exhaust emission. The CO emission result positioned around the regulation limitation (8 g/kWh) only for the both mode of diesel and for the C1-8 mode of the 10% blend. The specific CO emissions of R10 and I10 were difficult to meet the regulation and this CO emission increase appeared more significant with the increase of the blend ratio. Up to the 30% blend ratio, the PM emission was below the limitation for both modes. But because of the lower lambda at full loads and the misfire at R10, the PM emission of the 50% blend went far over the limitation.

3.3.6 Toxic Hydrocarbon Ratio

The toxicity of the exhaust gas from the combustion of the bio-oil was briefly commented. The necessity of measures to be taken and characterization of the toxicity exhaust gas products was emphasized [5]. In this report, toluene (C_7H_8), acetaldehyde ($\text{C}_2\text{H}_4\text{O}$) and formaldehyde (CH_2O) were defined as toxic hydrocarbon matters. The toxic hydrocarbon ratio was expressed as the ratio of the three toxic hydrocarbons content in the total hydrocarbon emission and is compared in Fig. 22.

Fig. 20 NO_x + THC mode emission for the diesel fuel (0%) and the blend oils.

In R10 where the operation is pretty scarce in the real world, the portion of the toxic hydrocarbons increased over two times up to the 30% blend ratio and three to four times for the 50% blend ratio than other operation points. But in almost all operation points up to the 30% blend, the portion was around 10% and was not nearly differed with that of biodiesel (plots inside red dot rectangle), whereas for the 50% blend, the toxic portion rose double to three times depending on the load at the rated engine.

3.4 Specific Fuel Consumption

Fig. 23 depicts the SFC (specific fuel consumption)

and the brake thermal efficiency in the maximum torque output, I100 and R100. The specific fuel consumption increased with the increase of the blend ratio due to the higher density and the lower heat

Fig. 21 (a) CO and (b) PM mode emission for the diesel fuel (0%) and the blend oils.

Fig. 22 Comparison of the portion of the toxic hydrocarbon in the THC (0: 100% diesel).

Fig. 23 (a) Specific fuel consumption and (b) brake thermal efficiency as a function of the blend ratio (0%: diesel).

Fig. 24 Possible engine operation points for the electricity generation.

value. For the 50% blend ratio, the deterioration level went worse because the ignition timing significantly delayed and hence the maximum cylinder pressure occurrence timing was more retarded from the best position where the diesel combustion produced. Up to the 30% blend ratio, the brake thermal efficiency was almost the same level but declined for the 50% blend due to the enlarged ignition delay.

The possible full load operation point for the electricity generation use according to the engine is

illustrated in Fig. 24. Up to 30% blend ratio, compatible engine power to the diesel was obtained. But for the 50% blend, there was uncertainty such as the power down at 2,450 rpm that might be caused by the decreased injection rate due to the higher blend ratio.

4. Conclusions

The pyrolysis oil (bio-oil) was produced from Japanese cedar in a pilot-scale gasification plant. The raw bio-oil was blended with biodiesel fuel in 10%, 30% and 50% volumetric ratio because of its high kinematic viscosity and poor ignition quality. A small single cylinder air cooled direct injection diesel engine was used for the combustion experiment to compare these blended oils with diesel fuel.

According to increasing the blending ratio, the full load power was kept to that of diesel fuel due to its higher oil density, CO, smoke and THC including the toxic hydrocarbon increased while the NO_x emission decreased.

Up to around 30% blending ratio, the engine operation was compatible to the diesel. It resulted that, almost identical full load engine output was achieved, the NO + THC and PM emissions were positioned around the EPA Tier1 standard while the CO emission went beyond the standard.

It should be careful about the increase of the kinematic viscosity depending on the raw biomass and the pyrolysis process because it may affect the spray condition in negative direction resulting in higher exhaust emissions such as smoke, CO and THC emissions. Additionally, higher water content may attribute to longer ignition delay resulting in misfire at part loads.

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