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An application of the Pyrolysis Oils to a Diesel Engine and its Performances

(熱分解オイルのディーゼル機関への適用と性能に関する研究)

By

Sunbong Lee

Submitted to the Department of Environmental Science and Technology,
Interdisciplinary Graduate School of Science and Engineering
in partial fulfillment of the requirements for the degree of

Doctor of Engineering

at the

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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 such as African, South Central Asia and Southeast Asia/Pacific. 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 minimum cost of the pyrolysis oils from the waste plastic and the waste biomass in a diesel engine, mainly for electricity generation. The determination of possible blend ratio was the major scope rather than the optimization of engine parameter. For the sake of it, all the pyrolysis oils which were used in this research were produced from the commercial or pilot plant in Japan and blended with diesel fuel for the oil from the plastic waste and with biodiesel for the oil from the waste biomass at minimum effort.

A small single cylinder diesel engine (direct injection) ran with the oils based on the US EPA standard test mode. Through the evaluation with regard to full load power-output and exhaust emissions, the blend fraction was defined. With the determined blend ratio, the engine continuously operated for 8 hours as one indication of the reliability. Finally, the waste pyrolysis oil can be used in a diesel engine with around the 20% blend oil or the 40% blend oil (one third engine power of the rated one at the intermediate engine speed) with no limitations such as endurance, exhaust emission and necessity of engine modification. The waste biomass oil needs more treatment even with low blend fraction to prevent engine (fuel delivery system) from the irreversible failure.

Thesis supervisor

Kunio Yoshikawa, Professor, Department of Environmental Science and Technology

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I had been interested in Waste-To-Energy since around end of 20 century and it went behind my job in Bosch Japan which started from 2002. One day in 2006 in my diminished memory, I found a sir who was researcher on WTE via internet. In 20011, I mailed about my history and wish and visited him, professor Yoshikawa. I finally started my voyage independently from my job. During the day some situation which was totally different from that in my job place made me difficult to proceed with it. Professor Yoshikawa largely encouraged me by his clear and simple solution and strong leading. Much discussion was openly and informally taken and clear guidance led me to go on the right way. I deeply thank professor Yoshikawa on where I am now standing and what I achieved. I pretty appreciate professor Kosaka in TIT for giving me a great chance to use the historically admirable test facility, technical supports and fruitful advices. Professor Yoshida in Nihon University gratefully shared his engine and resources with us and showed me a different world as one great scholar model, many thanks professor Yoshda. I sincerely appreciate professor Takahashi, professor Tokimatsu and professor Kamo for the great reviewing and contribution to improve the thesis quality. It was pretty thankful for me to have helpful support and encourage from assistant professor Tsuji and office support and warm welcome from sectary Mrs. Ohno.

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Chapter 1

Introduction

1.1 Research background

1.1.1 World energy perspective

According to the United Nations [1-1], the world population is constantly growing from 6.88 billion in 2010 to 9.49 billion in 2050 (Medium fertility prospects) as shown in Figure 1.1. In 2050, new 2.6 billion people will be mostly born from Africa and South/Central Asia region. By human nature which we are all want to become richer, economic growth will increase. The WEC (World Energy Council) forecasted that the global GDP will increase from 63 trillion US dollars in 2010 to 203 trillion US dollars in 2015 by around 200%. The main increase will be driven by the developing countries in Africa, Southeast Asia, East Asia and South/Central Asia.

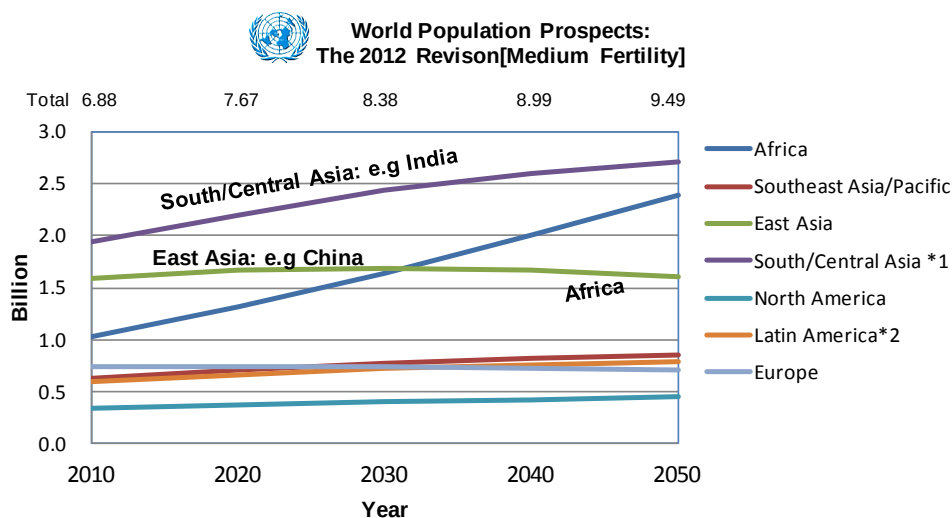


Figure 1.1 World population prospects: the 2012 revision

The world energy in 2010 was supplied by coal, oil and gas corresponding to 80% of the total primary energy as illustrated in Figure 1.2 [1-2]. The remainders are nuclear, biomass, hydro and renewable. In 2050, the coal, oil and gas will keep playing a main role even though the distribution will somewhat change. The gas and the renewable will increase by around 5% and coal and oil will decrease by about 2% and 7% respectively than in 2010.

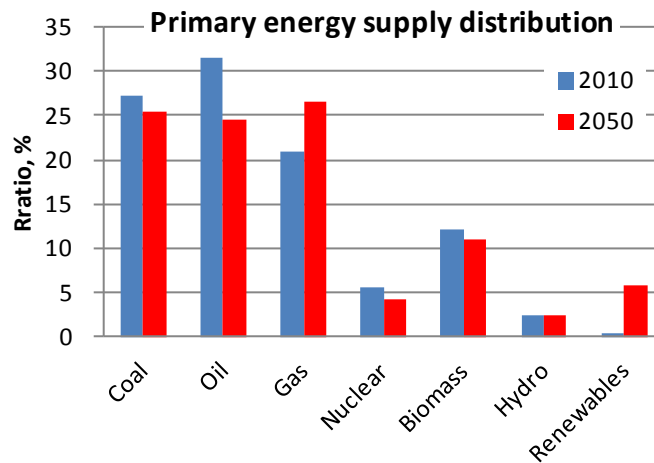


Figure 1.2 World primary energy supply distribution

The world GDP perspective and regional primary energy supply are illustrated in Figure 1.3 and Figure 1.4, respectively. The WEC (World Energy Council) estimates that the primary world energy supply which is equal to consumption will increase from 546EJ in 2010 to 879EJ in 2050 which corresponds to 61% increase [1-2]. The economy of North America and Europe will grow continuously by double to the level of 2010 but the primary energy supply will increase by about 10%. This means that the energy conservation will be developed more. For Africa and South/Central Asia, the energy consumption will drastically increase due to the increase in the population and the GDP growth. For other regions, the energy consumption will be driven by the GDP growth without the population increase.

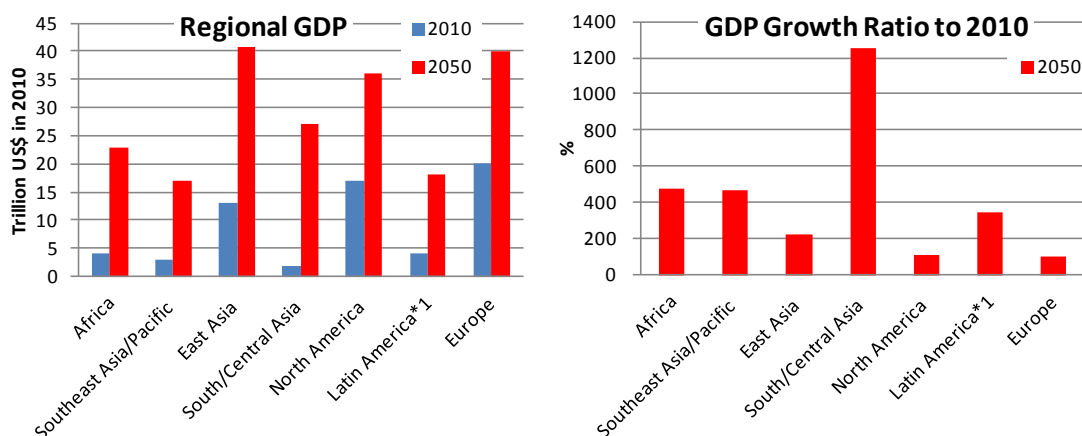


Figure 1.3 World GDP perspective and growth

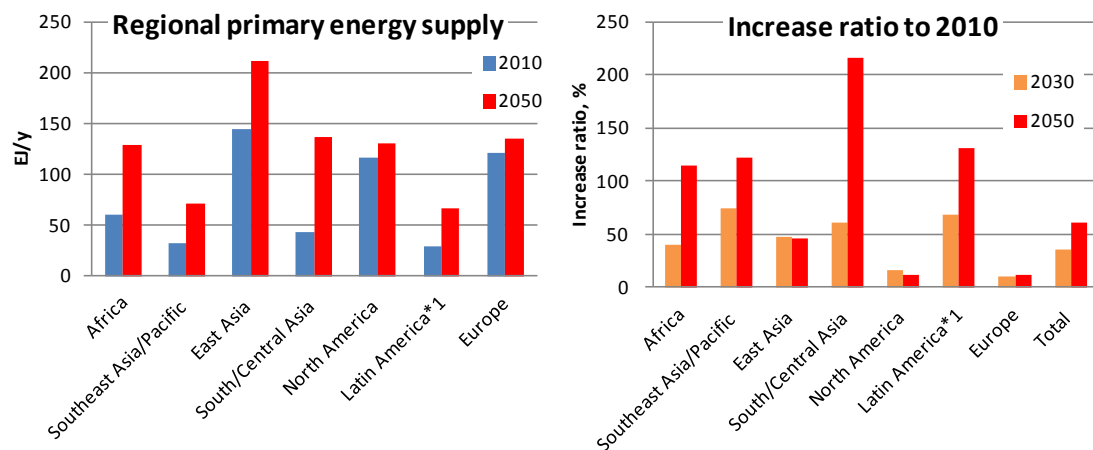


Figure 1.4 World regional primary energy supply and increase ratio to 2010

The population who could not access to the electricity as depicted in Figure 1.5 was around 1.26 billion globally in 2010 and it means that one of five men did not reach the electricity. It mainly distributed in Africa, South/Central Asia and Southeast Asia/Pacific. The current electrification is built on the grid system which needs huge investment corresponding to the high GDP. According to the WEC, the population without access to the electricity will decrease to 0.3 billion who will be still distributed in the three regions. But it can be realized only with the large investment to construct the grid system. Thus, the electricity generation by the non-grid and low cost system is necessary for the one and for the 0.3 billion.

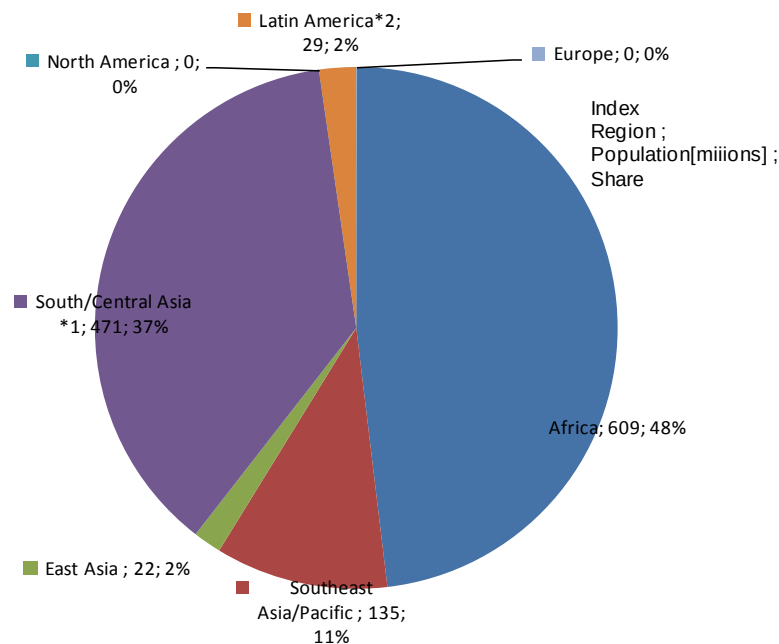


Figure 1.5 World population and share without access to electricity in 2010

1.1.2 World waste perspective

World Bank reported about the world waste situation and perspective in 2013 [1-3]. Figure 1.6 shows solid waste generation from the cities in the world. In 2010, 1.3 billion tons of solid waste were produced in the cities all over the world. The solid waste generation will increase to around 2.2 billion equal to 70% increase in 2025. According to the waste volume increase, the waste management cost will increase and it will be especially large burden to the low income and the lower middle income countries.

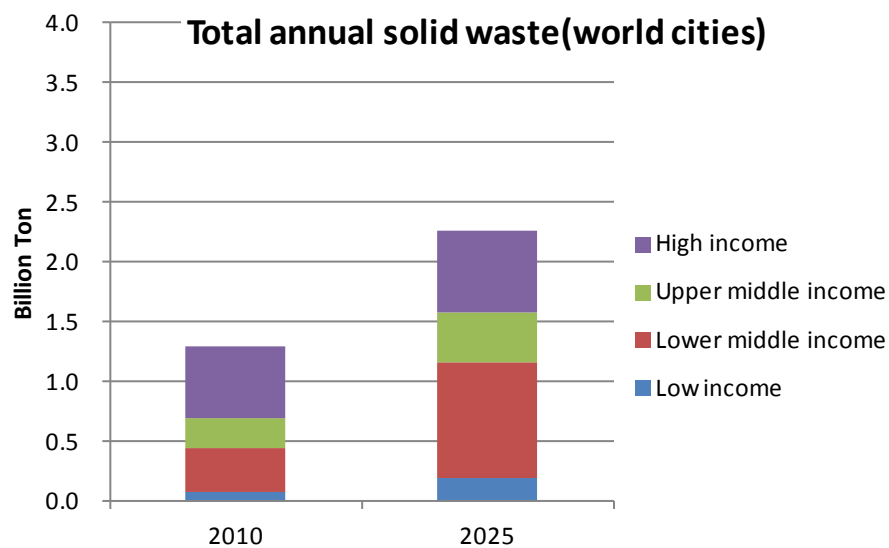


Figure1.6 Solid waste generation from the cities in the world

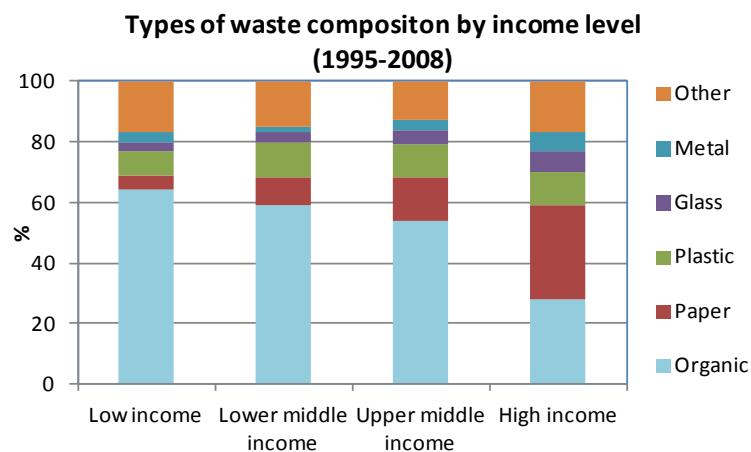


Figure 1.7 Waste type compositions by income level

Depending on the income level, the waste type compositions vary as shown in Figure 1.7. The organic waste composes around 60% for the low and lower middle income countries while 50% for the higher income and 30% for the high income countries. For the plastic wastes, they are in the range of 8 to 11 % between the countries. The organic and plastic wastes will increase by around double in 2025 as shown in Figure 1.8 and it will be driven mainly by the low and the lower middle income level countries as illustrated in Figure 1.9.

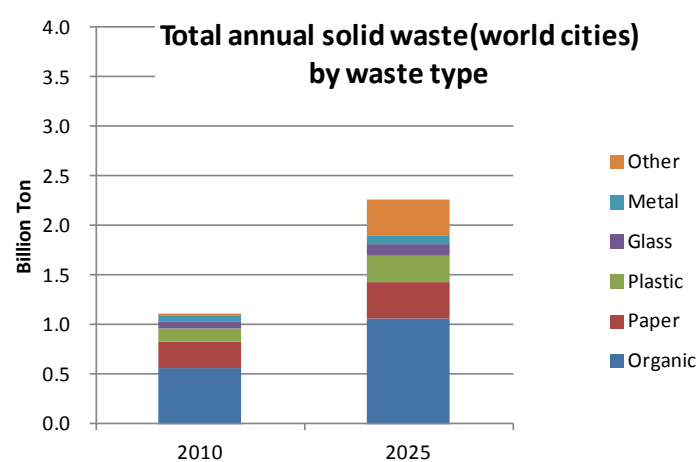


Figure 1.8 Total annual solid wastes from the world cities

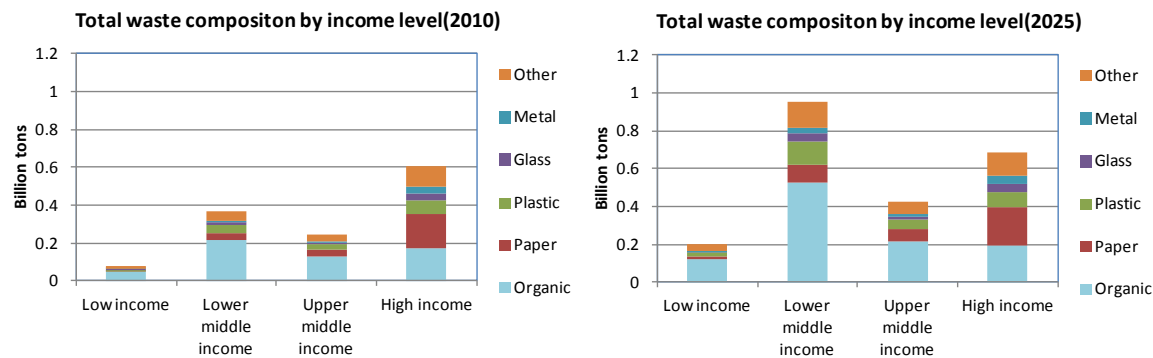


Figure 1.9 MSW disposals in the world

In high income countries, controlled municipal solid waste (MSW) management is carried out such as landfill, compost, thermal treatment and recycling as illustrated in Figure 1.10. In contrary to these countries, the solid wastes are dumped in the open field in the low income and the lower middle income countries. The uncontrolled solid wastes are impacting globally and locally. Open dumped solid wastes and uncontrolled landfills are a large source of methane which is very powerful green house gas and lead to air pollution and public health problem such as dengue fever.

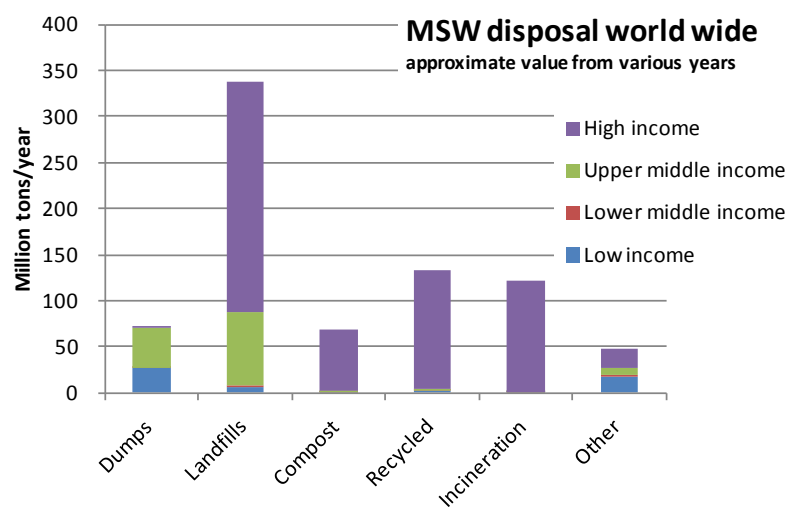


Figure 1.10 MSW disposal type by income level in the world (only approximate values, given that the data is from various years)

1.1.3 Summary and research approach

The population which could not access to electricity was around 1.2 billion in 2010 corresponding to one on five men and is distributed in many developing countries such as African, South Central Asia and Southeast Asia/Pacific, while the situation will be improved in the future by investing to build the grid electrification due to the economic growth in those regions. Depending on the future investment to construct the grid system, the scale of the population without electricity can vary. With the increase in the population and economic growth in those countries, waste generation is growing rapidly 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. The United Nations issued the outlook report of waste to energy in related to global status, technical and economic considerations and market trends [1-4]. In the report, the methodologies producing energy from waste was categorized as thermo-chemical conversion, bio-chemical conversion and 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 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 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 the view of the conversion efficiency and the stable engine operation.

1.2 Survey on the utilization of the waste conversion oils in diesel engines

1.2.1 Pyrolysis oil from biomass

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 [1-5]. Solantausta, et al. 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 [1-6]. Chiaramonti, et al. 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 [1-7]. Eucalyptus/diesel blends were tested in a single cylinder direct injection diesel engine and the smoke and THC emissions were reduced according to increasing the ratio of blending up to 50% while NO_x increased due to longer ignition delay which enlarged the portion of the premixed combustion [1-8]. 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 [1-9]. S. Lee 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 [1-10]. 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 emission 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 R. et al [1-11] 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 emission 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 were 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 respectively by adding 2% surfactant by volume. NO and CO emission was higher and HC and smoke emissions were lower than the case of the diesel fuel at full load [1-12].

1.2.2 Pyrolysis oil from waste plastic

The oil from the scrap-tire thermal-mechanical pyrolysis was run on a single cylinder diesel engine using automotive diesel fuel and two mixtures of the diesel fuel and the tire pyrolysis oil (TPO) with the volume ratio of 20% TPO and 40% TPO due to the lower cetane index of TPO. The 20% mixture of TPO and the diesel fuel showed no significant differences for power, fuel consumption and exhaust gas emissions [1-13]. The shredded PVC and PET bottle plastics were pyrolyzed in the laboratory scale pyrolysis device and the pyrolysis oil was mixed with diesel fuel using an ultrasonic vibrator due to the high kinematic viscosity around 3 times than that of diesel. 5% mixed oil of the diesel and the pyrolysis oil was tested in a single cylinder diesel engine at 50% load of the engine maximum power output and resulted in a slight improvement of the fuel consumption [1-14]. Plastics waste was converted into liquid oil by using the pyrolysis process including the catalytic cracking with the pre-treatment of the shredding. A blend of 50% and 70% with the pyrolysis oil by volume in the diesel fuel were evaluated for the thermal efficiency and the exhaust gas emissions by operating a single cylinder diesel engine at its maximum engine output speed with varying the engine load and showed the higher thermal efficiency and the higher gas emission for CO and THC [1-15]. Mukherjee et al [1-16] produced the waste plastic oil via the pyrolyzer without the catalytic processing and blended with the diesel fuel and the ethanol by the ratio of 20, 40 and 60% and tested in a direct injection twin cylinder diesel engine at the rated engine speed by changing the engine load. They concluded that the 20% blend oil has characteristics as close to diesel regarding to the thermal efficiency, NO_x and CO emissions. Pratoomyod et al [1-17] carried out the operation test in a direct injection 6-cylinder diesel engine by using the blend oil which was diesel blended with waste plastic oil with the ratio of 25%, 50% and 75% by volume. The engine operation point was ranged from the two thirds of the rated engine speed to the idle speed at a certain interval on the 100% load. The specific fuel consumption, the CO and the THC emissions were higher than diesel operation and the NO emission increased with the increase in the blend ratio. Several research results were reviewed by Patel et al [1-18] and Harshal et al [1-19] about the usage of the blended waste plastic pyrolysis oil with diesel fuel in the diesel engine on the engine operation range which was specified by the researchers.

1.3 Research objectives

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 minimum cost of the pyrolysis oil in a diesel engine, mainly for electricity generation. Engine modification and high effort pyrolysis oil upgrade are out of scope. 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 the evaluation criteria. After completing the definition of the blend ratio, the continuous engine operation at the rated power point was conducted during 8 hours in order to make sure of the sustainability of the blend ratio minimally.

For the sake of it, all pyrolysis oils derived from the waste biomass and the waste plastic which were used in this research were produced from the commercial or the pilot plant in Japan and these plants are introduced later.

Engine operation points were picked up based on the US EPA standard and the specific gas emissions were evaluated in addition to full load performance.

The production of the pyrolysis oil, the properties analysis of the raw pyrolysis oils, the selection of the engine operation point and the definition of combustion phases are followed in this chapter.

For better understanding of the interaction between the pyrolysis oil and the engine test result, the used oils for the combustion test and the experimental devices are described detail in each chapter.



Figure 1.12 Photo of the pilot-scale gasification plant

The mass balance of the bio-oil production is depicted in Figure 1.13. The bio-oil throughput which is available to run engines is around 4.5% of the total output volume and its maximum output is the synthetic gas by around 68.8%.

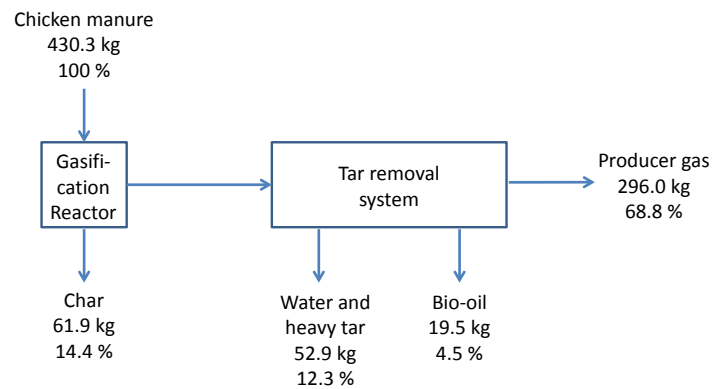


Figure 1.13 Mass balance of the bio-oil production from chicken manure

The raw bio-oil obtained from woody waste had a higher kinematic viscosity than diesel fuel as shown in Table 1.1 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 adaption to diesel engines difficult, due to unfavorable and unpredictable troubles in engine systems.

Table 1.1 Properties of the raw bio-oil (Woody waste)

Chemical		
%wt/wt	Diesel[JIS2]	Raw bio-oil
C	85.04	50.8
H	13.55	7.9
N	0	0.2
S	0	0
O	0	18.92
Physical		
Density [kg/cm ³] at 15 degC	0.83	1.055
Kinematic viscosity [mm ² /s] at 40 degC	2.744	7.032
Water content [%wt/wt]	0	18.8
Low Heat value [MJ/kg]	45	22.4

Literature review on physical and chemical properties

The property of biomass pyrolysis oil is summarized with focusing on the application to engines based on the reviews [1-20] [1-21] [1-23]. The bio-oil from biomass is typically a dark-brown liquid with a pungent odour, and the physical and chemical properties of the bio-oil are different from conventional fossil fuels.

Chemical composition

The chemical components of bio-oil are complicated and comprising of mainly water, acids, alcohols, aldehydes, esters, ketones, sugars, phenols, phenol derivatives, lignin-derived substances. The complexity of the bio-oil itself results in the difficulty to analyze and characterize [1-24](Wildschut, 2009). Gas chromatography-mass spectrometry (GC-MS) has been the technique most widely used in the analyses of the composition. The major components of one kind of crude bio-oil based on the GC-MS analyses are shown in Table 1.2.

Table 1.2 Composition of raw biomass oil [1-25]

Main components	RT/min	Area w/%
formaldehyde	1.42	3.14
aldehyde	1.51	6.52
hydroxyacetaldehyde	1.61	3.14
hydroxypropanone	1.72	2.70
butyric acid	1.82	0.96
acetic acid	2.07	29.76
glyceraldehyde	2.6	3.54
3,4-dihydroxy-dihydro-furan-2-one	2.77	3.27
2,2-dimethoxy-ethanol	2.86	6.83
furfural	3.13	6.56
2,5-dimethoxy-tetrahydro-furan	3.5	3.47
4-hydroxy-butyric acid	4.27	0.43
5H-furan-2-one	4.51	0.74
2,3-dimethyl-cyclohexanol	4.76	1.31
3-methyl-5H-furan-2-one	5.19	0.38
corylon	6.15	1.18
phenol	6.59	1.57
o-cresol	6.8	1.12
m-cresol	7	1.46
2-methoxy-6-methyl-phenol	7.79	1.78
3,4-dimethyl-phenol	8.99	1.14
4-ethyl-phenol	9.7	1.31
3-(2-hydroxy-phenyl)-acrylic acid	10.1	1.53
catechol	10.81	3.53
3-methyl-catechol	11.9	1.36
vanillin	12.7	0.24
4-ethyl-catechol	12.86	0.71
levoglucosan	14.73	9.95
2,3,4-trimethoxy-benzaldehyde	15.5	0.20
3-(4-hydroxy-2-methoxy-phenyl)-propenal	15.8	0.15

Heat value

The heat value of the bio-oil is calculated using the formula [1-26] [1-27].

$$\text{HHV} = 338.2 \times C + 1442.8 \times (H - O/8) \text{ (MJ/kg)} \quad (1)$$

$$\text{LHV} = \text{HHV} - 218.3 \times H\% \text{ (wt\%)} \text{ (KJ/kg)} \quad (2)$$

The heat value of bio-oil is only around half of the diesel and the gasoline fuel [1-28]. It is because of the higher water and oxygen contents. In order to improve the heat value, it is necessary to reduce the contents of water and oxygen by some methods of upgrading.

Water content

It is not avoidable that the water exists in the bio-oil because of the moisture in the raw material. The water content is generally in the span of 15-35 wt% [1-29][1-30]. Charles J. M. [1-5] reported that the bio-oil is typically microemulsion with an aqueous solution of

carboxylic acids, aldehydes, ketones, and sugar-like compounds as the continuous phase and mainly lignin-derived water insolubles as the distributed phase. The high water content and polar nature in the bio-oil makes it difficult to solve in crude oil based oils such as diesel and gasoline but easy to solve in ethanol [1-6]. It reduces flame temperature in combustion and may cause phase separation in storage. It also impacts on corrosiveness to carbon steels, aluminum and viton [1-5]. In contrast to it, it is advantageous to decrease viscosity and facilitate atomization [1-32].

Oxygen content

The oxygen content of the bio-oil ranges of 35-40% [1-33]. The high oxygen content leads to bad properties such as high corrosivity, high viscosity, low energy density and thermal instability [1-34]. Of course, a certain amount of oxygen in the fuel is helpful to improve exhaust emission, less particulate matter [1-35].

Density

The density of bio-oil is normally in the range of 1.1-1.3kg/m³, depending on the raw materials and pyrolysis conditions. This high density is caused by the existence of a large number of water and macromolecule such as cellulose, hemicelluloses, oligomeric phenolic compounds [1-33].

Viscosity

The viscosity ranges from that of heavy oil and conventional diesel oil and highly depends on the water content and the temperature of the oil. The higher the water content is, the lower the viscosity is [1-6].

Blending the bio-oil with methanol, acetone or ethanol can reduce the viscosity. The viscosity generally increases with time due to liquid phase polymerization and oxidation reactions [1-16].

The condition of the spray atomization of pyrolysis oil which is expressed by the SMD(Sauter Mean Diameter) is worse than that of diesel due to its higher viscosity [1-7].

Ignition quality

It has been reported that the ignition quality was poor and the estimated cetane number was approximately 0 [1-37], 5.6 [1-38] and 30-35 [1-39]. The cetane number of diesel fuel is 51 for EN590 (2013), 40 for number 2 diesel of ASTM D613.

Distillation

The woody pyrolysis oil starts its distillation below 100°C and ends in the range of 250-280 °C and 35-50% tar-like residue is left due to the polymerization reaction during the slow

distillation process[1-36][1-40]. It was reported that the polymerization before complete vaporization inside the combustion chamber might be one source of the higher deposit in the combustion chamber. [1-6][1-41][1-37]

Ash

Ash is the residue of bio-oil after its combustion, and the ash can be determined according to ASTM D 482. The ash of bio-oil usually varies in 0.004-0.03 wt% [1-33], which is also relevant to the raw materials and reaction conditions. In general, the ash content is higher for the straw oil than for other oils due to their originally higher amounts in straw than in wood [1-42]. The presence of ash in bio-oil can cause erosion, corrosion and making problems in the engines and the valves [1-43]. However, there is no effective way to reduce the content of ash by now [1-31].

Flash point

The flash point of a volatile liquid is the lowest temperature at which it can vaporize to form an ignitable mixture in air. Flash point is measured using a flash-point analyzer according to ASTM D 93. The test temperature usually employs the increase of 5.5°C/min in the range of 30-80°C [1-24].

Flash point is influenced by the raw materials and preparation method, because these will result in the differences in composition and content of the bio-oil from biomass. In general, the bio-oil from hardwood has a high flash point due to the low contents of methanol and evaporation residue of ether soluble [1-42].

pH

The bio-oil has amount of diluted water and volatile acids, such as acetic and formic acid, which results in the low pH values varied in 2-3. The presence of acids in the bio-oil is the main reason to account for the property of corrosion to materials in the storage and application processes [1-20].

Lubricity

The bio-oil cannot be used well in internal combustion engine because of the serious lubrication [1-43]. However, using emulsion technology to mix bio-oil with diesel fuel is one of the most convenient approaches to use bio-oil [1-38][1-44] and the average friction coefficient of the emulsified bio-oil was 0.130, which was lower than commercial diesel number zero (0.164). This result indicated that the emulsified bio-oil had better lubricity properties than commercial diesel number zero. The existence of oxygen might accelerate the corrosion wear on the rubbing surface.

Upgrading of the bio-oil [1-55]

The raw bio-oil cannot be miscible with crude oil diesel and gasoline due to the water content and it needs upgrading treatment. Ying had researched and reported about the upgrade of the raw bio-oil which was produced in the pilot plant as shown in Figure 1.14 and Figure 1.15. By the thermal dewaterization, the water content reduced around zero level whereas the kinematic viscosity increased up to $14\text{mm}^2/\text{s}$. Then through the 2nd pyrolysis and catalyzing process, the kinematic viscosity was decreased again (still over $6\text{mm}^2/\text{s}$) but the production gain to the raw input biomass declined to 2~1% (ratio to the input biomass). The blend treatment reduced the water content down to 15% in total weight in the blend oil, increased heat value and led to slightly decrease the density and the kinematic viscosity.

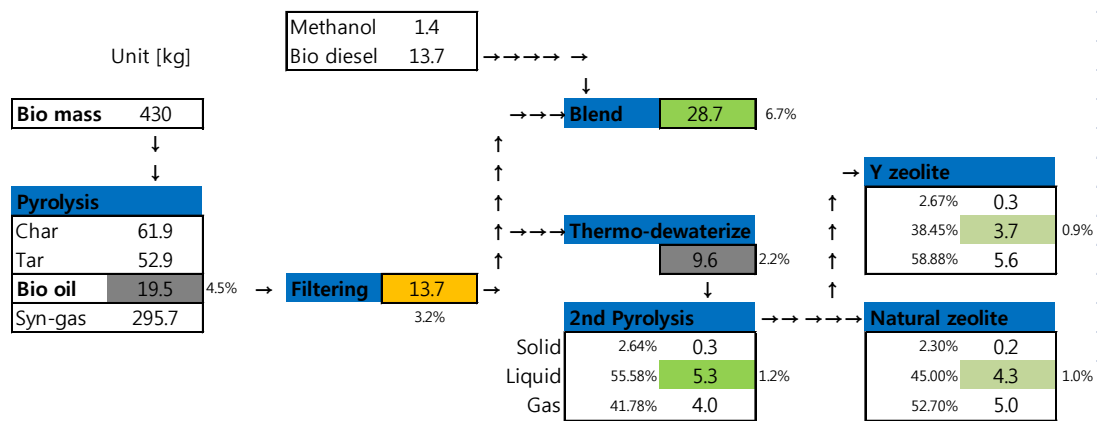


Figure 1.14 Summary flow of the upgrade treatment for the raw bio-oil

The optimal blend condition was parameterized for the ratio of the biodiesel (estrication of waste food oil)[1-57], the bio-oil, the surfactant of the methanol and mixing temperature and duration. The 5% methanol ratio (volume fraction to the sum of the biodiesel and the bioi-oil) was chosen for the 1 to 1 blend ratio (the biodiesel to the bio-oil) at 45°C for 60 min. mixing time. The observation for the phase separation was carried out for a certain time and there was no separation for the duration but it was not fully verified for the eternal or partial separation.

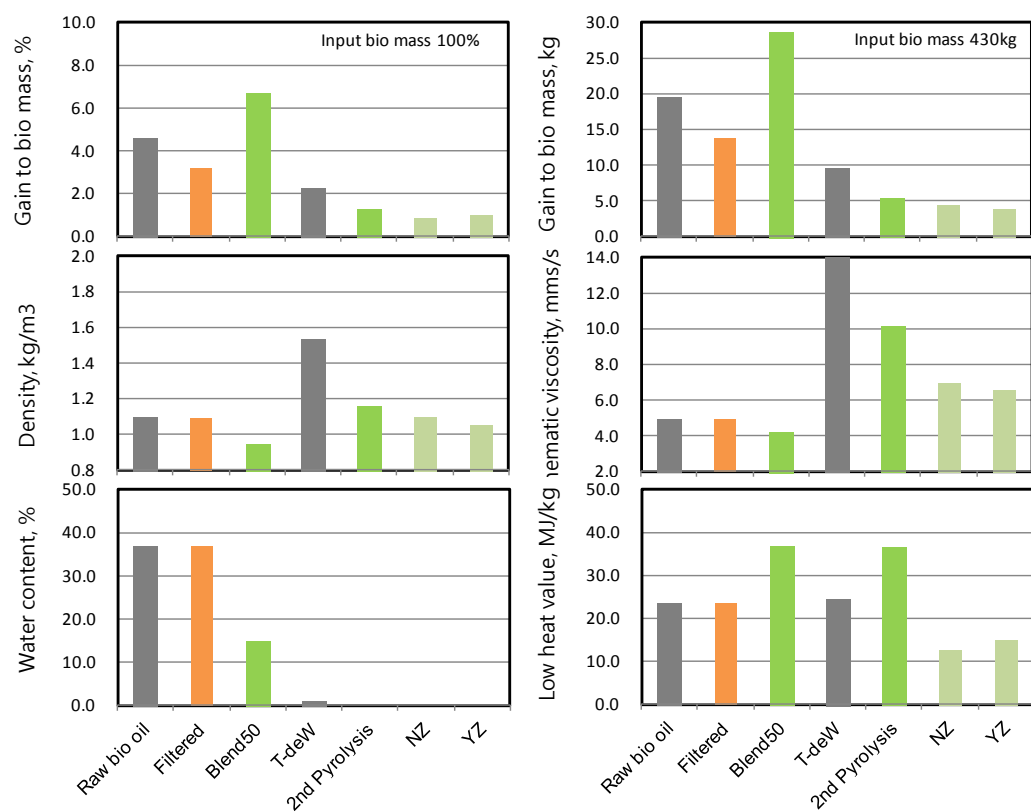


Figure 1.15 Property change by the upgrade treatment

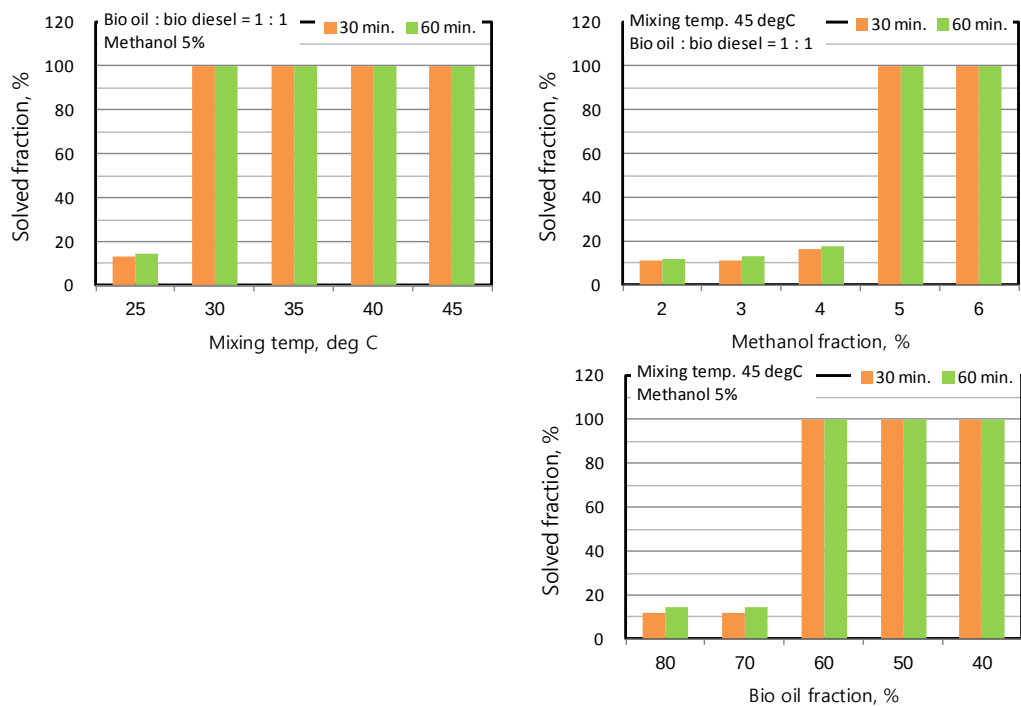


Figure 1.16 Parameterization of the blend treatment

1.4.2 Waste plastic pyrolysis oil [1-56]

Municipal solid wastes which are shown in Figure 1.17 are collected and transported to a waste disposal company. Many kinds of plastic wastes which are containing food, drinking beverage and water residues are packed in a plastic bag in the stage of disposal from households. There is no segregation process in the company as pre-process ahead of the pyrolysis due to the cost.

The conversion of plastics to oil is based on the pyrolysis under the absence of oxygen and the catalytic reforming and the condensation of the resulting gases which is illustrated in Figure 1.18. The semi-batch type reactor which is movable is used. The product gas from the pyrolysis reactor flows into the alumina-based catalytic reformer by the force of vaporizing pressure. The oil burner at the bottom of the jacket surrounding the reactor heats the reactor for the thermal decomposition reaction with the temperature controlled at 400°C and the exhaust gas flows into the outer side jacket of the catalytic reformer, heats the reformer and then exhausted out via the vent. The product gas is further cracked in the catalytic reformer. After passing the reformer, direct scrubbing by spraying the oil is carried out. The condensed oil is then collected in the oil receiver. The off-gas is supplied to the incinerator next to this pyrolysis system to completely burn. After finishing the pyrolysis reaction in a reactor, overhead crane lifts up the reactor and installs a new reactor which is already packed with the feedstock and the whole process starts again.



Figure 1.17 Collection of raw plastic wastes

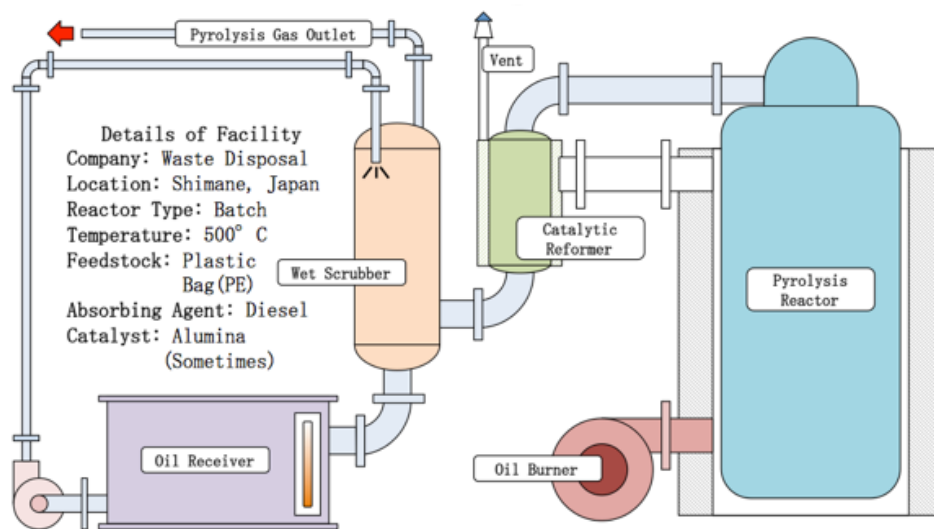


Figure 1.18 Pyrolysis process of the commercial plant

Table 1.3 shows the hydrocarbon composition analysis and the physical property of the raw pyrolysis oil and diesel fuel, respectively. The raw pyrolysis oil shows almost the same chemical content in carbon and hydrogen excepting for oxygen. It is inferred that the oxygen in the pyrolysis oil came from the contamination of the raw plastic wastes by food and drinking residues in the stage of consumer disposal. The amount of oxygen is low enough and can be neglected as one of the effective factors influencing the engine performances. The density and the kinematic viscosity are lower than those of diesel fuel. The density of the pyrolysis oil may affect the full load performance. The kinematic viscosity undergoes slightly less than the standard tolerance range of the diesel fuel in the United States [1-45]. When the raw pyrolysis oil is directly used in diesel engines without any treatments, it would be concerned in terms of the lubricity in the fuel delivery and the injection system of diesel engines [1-46]. The heating value is almost the same as diesel due to the similar content of hydrocarbon as stated above.

Table 1.3 Composition analysis

Composition analysis		
%wt/wt	Diesel [JIS2]	Raw pyrolysis-oil [Waste plastic]
C	85.04	85.87
H	13.55	13.71
N	0	0
S	0	0
O	0	0.421
Physical property		
Density [kg/cm ³]	0.83	0.75
Kinematic viscosity [mm ² /s]	2.744	1.19
Water content [%wt/wt]	0	0
Low Heat value [MJ/kg]	45	45.4

Figure 1.19 illustrates the carbon atom number distribution of the pyrolysis oil obtained from the commercial plant. The raw pyrolysis oil was analysed by GC-MS analyzer to compare the distribution of hydrocarbon atom to that of diesel. The carbon number distribution of the oil product from the commercial plant was obviously shifted from the hydrocarbon range of diesel. The large fraction of hydrocarbon was found in the range of 6-14, which is much lighter than diesel and is close to the range of gasoline. It reached the maximum fraction at the carbon number of 11. In contrast, the hydrocarbon range of diesel was in the carbon number of 8-25 and the highest fraction is seen at 17. The fractions of carbon atom number from both oils are totally different.

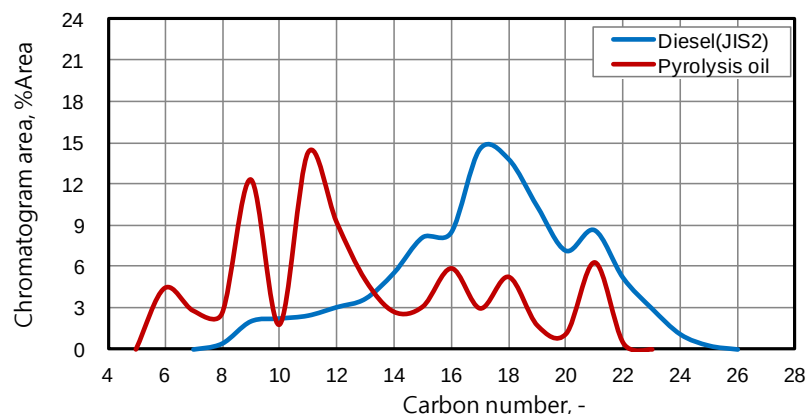


Figure 1.19 Carbon number distributions

The carbon class of the diesel fuel and the raw pyrolysis oil is illustrated in Figure 1.20. It mainly consists of iso-alkanes, n-alkanes and olefins in % area of 27%, 25% and 9% respectively. Over 30% content was not able to be defined due to its complicated chemical bond structure. The shape of the carbon class distribution of the raw pyrolysis oil is not like kerosene/jet oil [1-48] and gasoline [1-49]. Aromatic is one of important factors affecting the level of the particulate matter emission from diesel engines [1-47] and is very low. Comparing to the diesel fuel, the content of n-Alkanes which is good for self-ignition in compression ignition engine is lower by around 25% than that of diesel. Aromatics, Cyclo-alkanes (Naphthenes), isoalkanes and olefins which are not good for the self-ignition quality [1-50] compose around 40%. From this analysis, it can be estimated that the raw pyrolysis oil has very poor self-ignition quality.

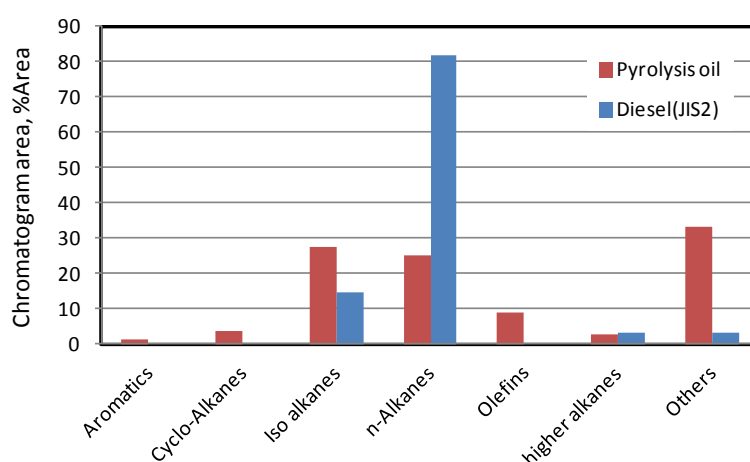


Figure 1.20 Hydrocarbon classes

1.5 Engine operation points

As previously reported, the present paper is aimed to investigate the compatibility of the pyrolysis oil to diesel engine, thus test points in the engine operation were selected based on the US EPA emission legislation test mode [1-51]. C1-8 mode which regulates the exhaust emissions for non-road vehicles operating in construction, air port, mobile cranes, agricultural and forest place is depicted in Figure 1.21. D2-5 mode is defined for the certification test of the exhaust emissions for generation sets illustrated on Figure 1.22.

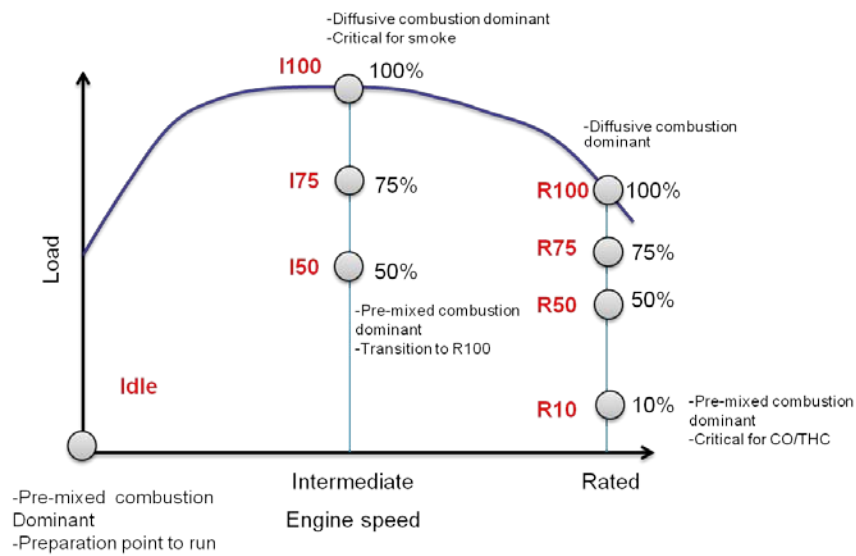


Figure 1.21 C1-8 mode of the US EPA

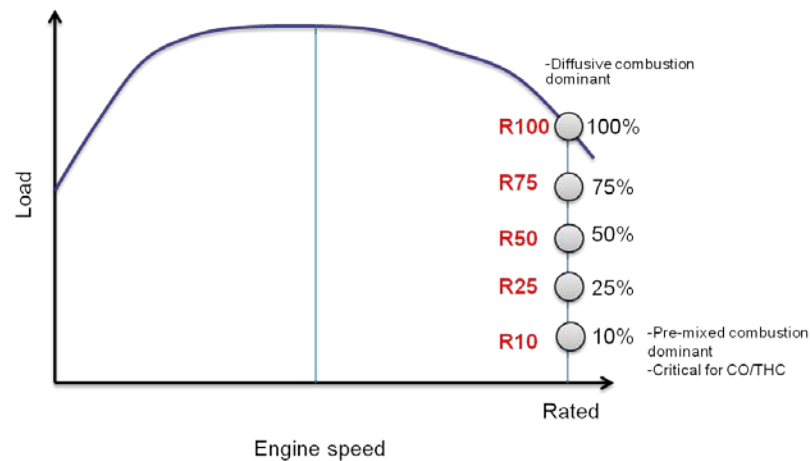


Figure 1.22 D2-5 mode of the US EPA

Finally, 8 engine test points (eight black-lined circle) summarized in Figure 1.23 were selected as the idle, I50/I75/I100 for the intermediate engine speed and R10/R50/R75/R100 for the rated engine speed. When evaluating the experimental results, R25 (25% load at the rated speed) data were interpolated between 50% and 10% because the linearity was shown to be enough. The weighting factor is shown in Table 1.4 which is needed for the calculation of the mode emission. Specifying the engine speed and the load for the engine which is used in the paper will be placed later in each chapter.

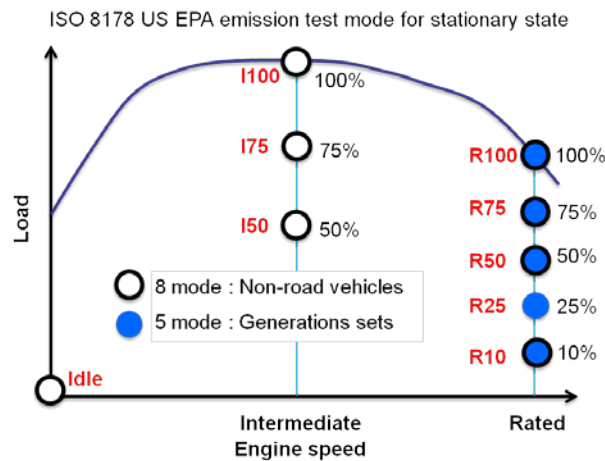


Figure 1.23 Engine operation points: eight black-lined circles

Table 1.4 Weighting factor

Operation point	Weighting factor	
	C1-8 mode	D2-5 mode
Idle	0.15	-
I50	0.1	-
I75	0.1	-
I100	0.1	-
R10	0.1	0.1
R25	-	0.3
R50	0.15	0.3
R75	0.15	0.25
R100	0.15	0.05

All exhaust emissions from diesel engine test were sampled as the concentration values in the experiment and processed to weight based values using the carbon balance and the fuel consumption so that the compatibility of the pyrolysis oil to diesel can be evaluated according to the US EPA regulation. The US EPA defines the Tier1 regulation as 10g/kWh of NO_x+NMHC, 8g/kWh of CO and 1g/kWh of PM for the test engine category [18] using the test mode described in the section 1.6. In actual, THC emission was evaluated and is actually slightly higher than NMHC (Non Methane HydroCarbon) which is defined as the regulation but the amount of the deviation was low enough to be neglected. For the PM, the smoke opacity was also converted to the mass value using the formula. [1-53][1-54]

The regulation emission numbers were used as just reference value in the report.

1.6 Combustion analysis

Heat release analysis which was developed by Goering C. E. [1-58] was used for combustion analysis.

$$\frac{dQ}{d\theta} = \frac{V \frac{dP}{d\theta} + \gamma P \frac{dV}{d\theta}}{\gamma} - \frac{dQ_w}{d\theta}$$

$dQ/d\theta$ = rate of heat release (J/CA)

CA = Crank angle

V=gas volume (m³)

P=cylinder pressure (Pa)

γ = ratio of specific heats

dQ_w = rate of heat transfer from the wall (J/CA)

The value of γ varies with the temperature and the gas temperature is needed to calculate the heat transfer to the wall. The spatially averaged temperature was calculated based on the ideal gas law.

Via the rate of the heat release and the injection timing, combustion phases were defined as below [1-59].

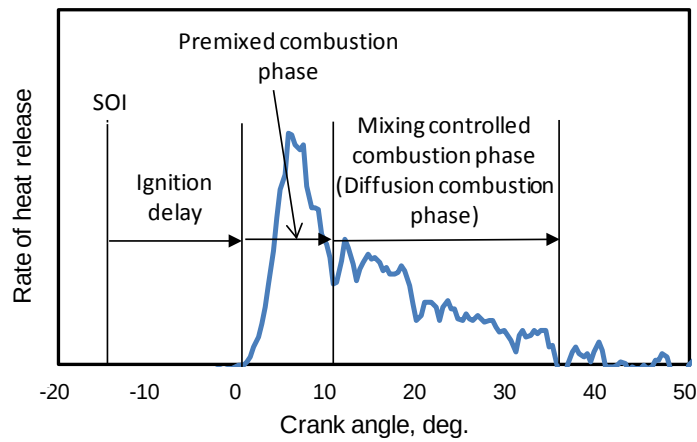


Figure 1.24 Heat release rate and combustion phases (SOI : Start of injection)

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Chapter 2

Combustion characteristic investigation of the waste biomass pyrolysis oil in a RCM

Before applying the biomass pyrolysis oil to a diesel engine, combustion characteristic was experimentally investigated and the result was presented in this chapter. The pyrolysis oil was produced from chicken manure in a pilot-scale pyrolysis plant in Japan. The raw pyrolysis oil had a very high viscosity and sediments which made direct application to diesel engines difficult. The bio-oil was blended with diesel fuel by 25% to 75% volumetric ratio at the normal temperature. A rapid compression machine (RCM) was used for the combustion test under the experimental condition corresponding to the medium operation point of a light duty diesel engine using diesel fuel and the 25% blend oil.

The RCM has several advantageous features in conducting combustion tests rather than real engine as below [2-1][2-2][2-3].

- 1) The combustion test is carried out for single cycle combustion.
- 2) The temperature of the combustion chamber is possible to be locally controlled.
- 3) The oil-free piston rings remove the impact of the lubricant oil on the exhaust emission.
- 4) The gas condition inside the combustion chamber at TDC, Top Dead Center, can be variously modified by changing the temperature, the pressure and the gas composition in the gas vessel which is lined to the intake port.
- 5) The combustion test can be possible with small volume of fuel.
- 6) Several kinds of optical diagnostic device can be accessed for single cycle combustion without difficulty

There had been much effort to produce the bio pyrolysis oil in the pilot plant like optimizing the pyrolysis condition and removing water and heavy tar. The first delivery of the bio-oil was small volume therefore the RCM was the most appropriate to investigate its combustion characteristic.

2.1 Test Fuels

The raw bio-oil had a higher kinematic viscosity than diesel fuel as shown in Table 2.1 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 adaption to diesel engines difficult, due to unfavorable and unpredictable troubles in engine systems. Many researchers have attempted to upgrade and refine the quality of bio-oil [2-4][2-5][2-10]. Those treatments can distinctively improve the compatibility with diesel fuel but need a large amount of effort and costs. In the present investigation, a least treatment process was made to adapt the raw pyrolysis bio-oil to diesel engines.

Table 2.1 Properties of diesel, blend 25 and raw bio-oil

Composition analysis			
%wt/wt	Diesel [JIS2]	Blend 25 [75%-25%]	Raw bio-oil [Chicken manure]
C	85.04	83.11	56.01
H	13.55	12.96	8.46
N	0	0.42	3.32
S	0	0	0
O	0	-	26.40
Physical property			
Density [kg/cm ³]	0.83	-	1.12
Kinematic viscosity [mm ² /s]	2.744	-	9.05
Water content [%wt/wt]	Trace	14.34	36.79
Low Heat value [MJ/kg]	45	41.7	24.95

The raw bio-oil was mixed with diesel fuel at the normal temperature with the volumetric ratios of 25% and 75%, respectively, and then filtered to remove the undesirable matter as mentioned above. A Whatman No. 41 filter was used for the filtration because the pore size 20-25 micro meter was compatible to that of the upgrade filter for commercial use in diesel engines. The processed bio-oil was named as blend 25 and its properties are shown in Table 2.1.

2.2 Experimental Procedure

The overall experimental apparatus is illustrated in Figure 2.1. It consists of mainly three parts: a rapid compression machine for combustion test (RCM), a common rail injection system and air delivery and exhaust gas emission measurement. RCM is not a real diesel engine, but can simulate the diesel combustion process for a single running cycle. The hydro-mechanical device drove a piston to compress conditioned artificial air very rapidly and stopped at its top position and then fuel was injected and combustion followed after a certain ignition delay. During combustion, the piston was kept stopped at the top position for the equal time with the compression process. Artificial air which consists of 21% oxygen and 79% nitrogen by volume was heated up to 493 K in the heating chamber and supplied to the cylinder controlling pressure by 0.23 MPa before the start of the compression process. After the combustion process, the whole burned gas was released outside of the cylinder and the smoke and NO_x concentrations were sequentially measured by the smoke meter and by the chemiluminescence meter, respectively [2-3], for comparing emission gasses. The fuel pressure at the high pressure line connected with the injector and the common rail and the in-cylinder pressure during compression and combustion were instantaneously picked up by using piezo-electric type pressure transducers during the single test cycle for analysis of combustion.

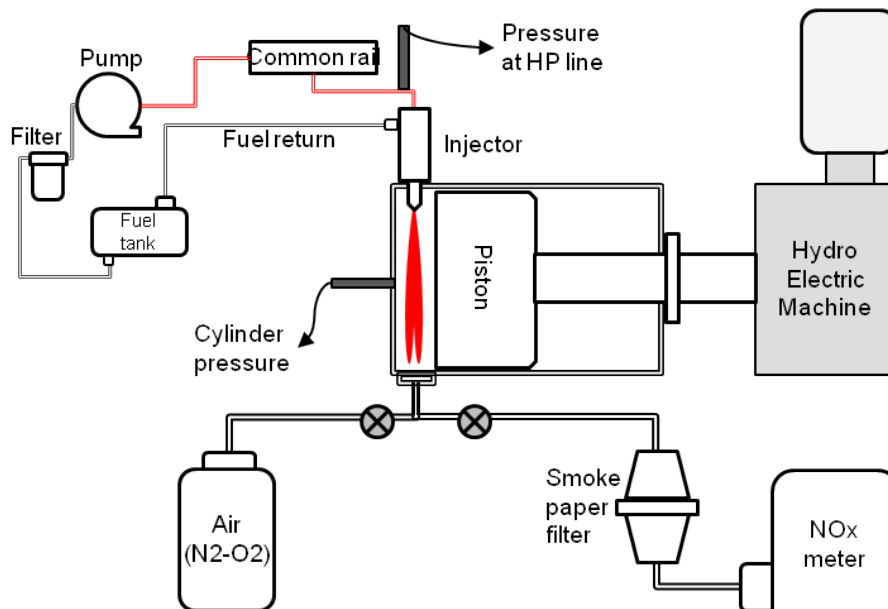


Figure 2.1 Experimental apparatus

The surrounding condition when the piston finished its compression process was 5 MPa in the pressure and 720 K in the temperature, which is shown in Table 2.2. This condition was

selected to simulate the middle engine operating point represented by the engine speed and the load of an off-road light duty diesel engine. Air excess ratio was 2 in the case of diesel injection.

Table 2.2 Surrounding conditions at the top position of the piston

Operating gas	O ₂ 21%, N ₂ 79%
Gas temperature [K]	720
Pressure [MPa]	5
Density [kg/m ³]	24
Air excess ratio, λ [-]	2

Combustion tests were done for diesel 100% and blend 25, and the injection conditions were equal for two combustion tests as described in Table 2.3. The amount for a single diesel injection shot was 32.4 mg under the injection parameterization which was 128 MPa in the common rail fuel pressure and 2.6 ms in the electrical charge injection duration. There was no direct metering for injection quantity of blend 25 and based on the density information of diesel fuel and bio-oil, it could be estimated as 35.2 mg making the air excess ratio a little bit smaller than 2. It can be negligible because the air excess ratio 2 is high enough.

Table 2.3 Conditions of the fuel injection at single event

Nozzle hole no. [-]	1
Nozzle diameter [mm]	0.24
Common rail pressure [MPa]	128
Injection duration [ms]	2.6
Injection quantity [mg]	
Diesel JIS2	32.4
Blend 25	35.2

2.3 Results

2.3.1 Combustion Analysis

Figure 2.2 illustrates the experimental results for a single combustion cycle. When a single injection event occurred, actual start and end of injection could be defined by analyzing the pressure wave in a high pressure line. The injection started around 45.5 ms where high pressure dropped suddenly from its original fuel control pressure of 128 MPa and finished around 49.5 ms where high fuel pressure recovered its controlling pressure after several pressure fluctuations. There is no significant difference in the two high pressure waves even considering the difference of the amount of injection. It is thought to be because 2.8 mg deviation in the quantity was too small to affect the high pressure wave. The cylinder pressure which was picked during the compression and combustion event was processed to acquire the heat release rate, the mass burned fraction and the mean temperature. The ignition delay was defined by the time from the start of the fuel injection to the start of combustion where the first rapid increase of the heat release rate took place [2-6]. The ignition delay of blend 25 was slightly shorter than that of diesel as shown in the small figure inserted in Figure 2.3. The pre-mixed combustion phase defined by the first mountain shape of the heat release rate followed the ignition delay, and the heat release rate of blend 25 was lower than those of diesel fuel due to the shortened ignition delay. However, during the mixing controlled combustion phase following the pre-mixed combustion phase, the heat release rate of blend 25 showed a higher peak than those of diesel fuel. It could lead combustion of blend 25 to finish in almost equal time to the case of diesel fuel as shown in the mass burned fraction graph in Figure 2.2 even though the injection quantity of blend 25 was 2.8 mg more than diesel fuel. It is thought that the combustion of blend 25 during the mixing controlled phase is more active than that of diesel fuel due to the presence of oxygen in blend 25 coming from bio-oil.

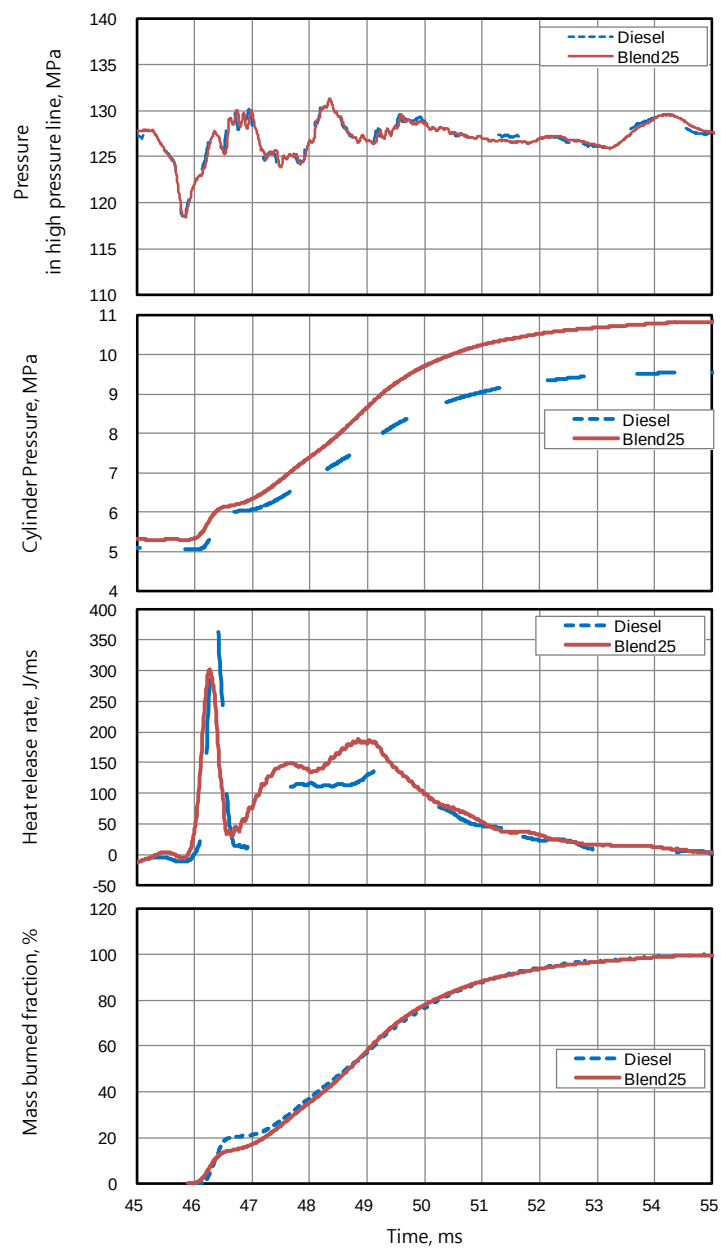


Figure 2.2 Experimental results for combustion analysis, rail pressure, cylinder pressure, heat release rate and mass burned fraction

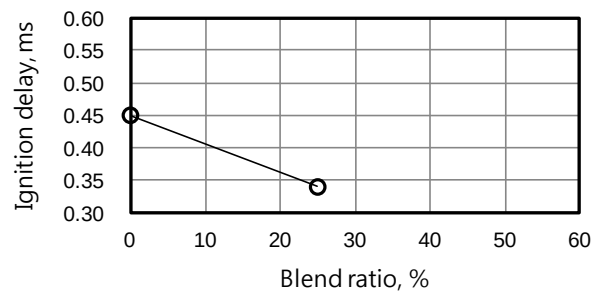


Figure 2.3 Ignition delay (0: diesel)

2.3.2 Exhaust gas NOx and smoke emissions

Figure 2.4 depicts the result of the exhaust gas NOx and smoke emissions. For blend 25, NOx increased up to about 40% and the smoke decreased by around 80%. Considering the discussion in the combustion analysis section above and the other research results on combustion and emissions of oxygen-containing bio fuels [2-7][2-8][2-11], this tendency makes sense. In the mixing controlled combustion phase, the flame temperature which is referred to as the mean temperature here shown in Figure 2.5 was higher in the case of blend 25. This may result in the increase of the NOx emission, while the oxidation of soot was enhanced by the aid of oxygen contained in blend 25. There seems to be some potential to improve the trade-off characteristics of NOx and smoke emissions by optimizing parameters of the injection system [2-9].

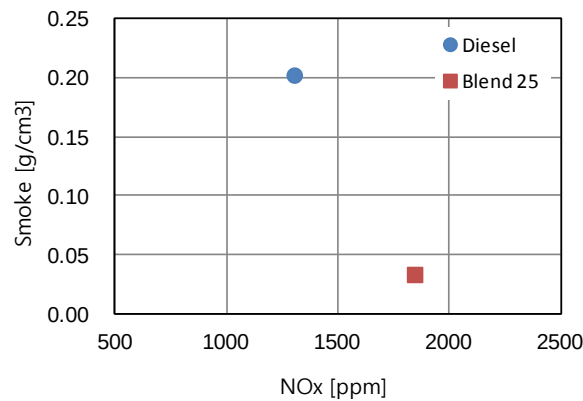


Figure 2.4 NOx and smoke emission

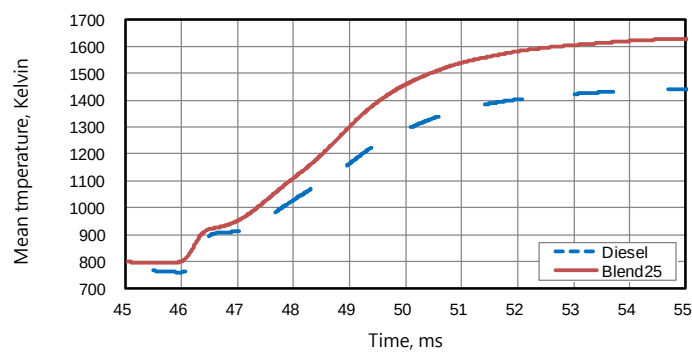


Figure 2.5 Mean temperature

2.3.3 Apparent combustion efficiency

For comparing the efficiency, the apparent combustion efficiency was introduced by dividing the actual heat release due to combustion by the theoretical potential heat in fuel. The theoretical heat of fuel was calculated by multiplying the low heat value and the fuel injection quantity listed in Table 2.1 and 2.3, respectively. For the actual heat release due to combustion, the heat release rate shown in Figure 2.2 was integrated in the range of combustion as illustrated in Figure 2.6. Actually, the heat accumulation at the time of 90% mass burned fraction was used, because in many cases it is difficult and prone to error to define the end of combustion in the late combustion phase from the heat release rate.

Blend 25 showed about 25% higher apparent combustion efficiency than that of diesel fuel as illustrated in Figure 2.7. Considering possible errors in the process of calculation of the apparent combustion efficiency and its dependability, this efficiency improvement seems to be slightly over estimated but the tendency appears to be acceptable based on the result of the combustion analysis.

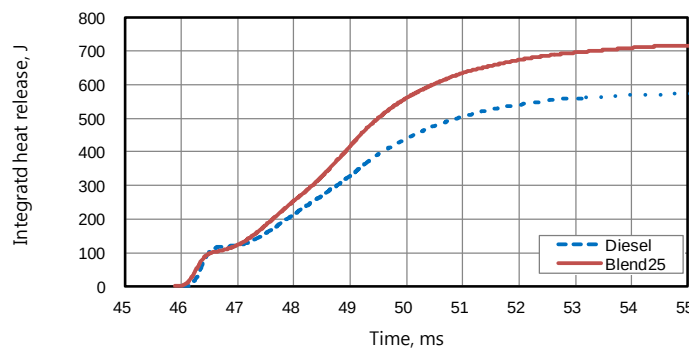


Figure 2.6 Integrated heat release

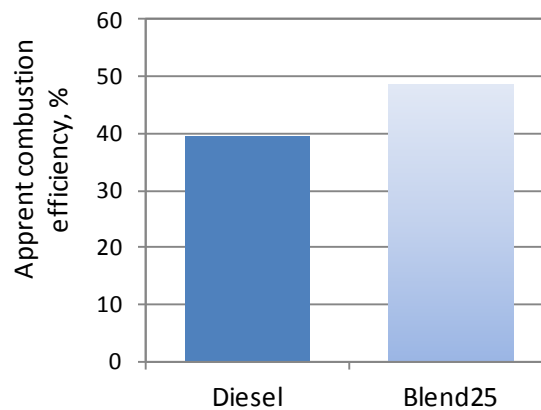


Figure 2.7 Comparison of the apparent combustion efficiency

The mixing controlled combustion dominated around 72% for diesel fuel and even higher of 77% for blend 25 in the whole combustion phase shown in the left of Figure 2.8. The heat release speed presented by the averaged heat release rate, which was calculated by dividing the accumulated heat by the combustion duration for each phase, shown in the right of Figure 5, was 20% faster for blend 25 than for diesel fuel, and this could contribute to terminate combustion in almost equal times for both fuels as mentioned above.

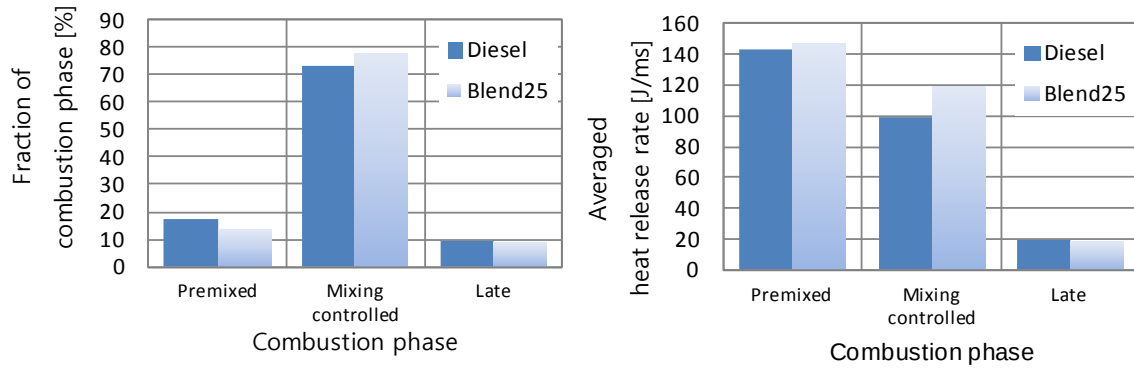


Figure 2.8 Left : Fraction of the combustion phase, Right : Phase averaged heat release

2.4 Conclusion

Raw bio-oil derived from chicken manure was blended with diesel fuel and filtered because of its low flow characteristic and unfavorable matter. A rapid compression machine which simulates single diesel combustion cycle was used for a combustion experiment with regard to diesel only and blend 25 made with the mixing ratio of 25% bio-oil and 75% diesel fuel in volume.

For the given experimental conditions, the ignition delay of blend 25 was slightly shorter than that of diesel fuel. The combustion period was almost dominated by the mixing controlled combustion phase for the two fuels. During the mixing controlled combustion phase, blend 25 showed a higher mean temperature and a faster heat release rate contributing to complete combustion in almost the equal time even with more injection amount than that of diesel. This resulted in less smoke emission by 80% and higher apparent combustion efficiency while NO_x emission increased by 40% comparing to diesel fuel.

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Chapter 3

Engine performance test of the waste biomass pyrolysis oil in a direct injection diesel engine

In chapter 2, 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. Upgrade treatment of bio-oil needs complicated process and low product yield in addition to high cost.

In this chapter, woody wastes were converted into the oil through the pyrolysis in the pilot facility which was described in the section 1.1. The raw bio-oil had been successfully mixed with biodiesel by adding methanol and there was no phase separation for the given observation.

The full load performance, the exhaust emission and the specific fuel consumption were investigated from the view point of the compatibility to diesel based on the US EPA regulation mode.

Through the evaluation, the effectiveness of the pyrolysis oil adaption to diesel engine will be verified and in moreover the appropriate blending ratio will be defined.

3.1 Materials and methods

3.1.1 Test fuels

According to the methodology of the blending which was described in the section 1.4.1, the bio-oil which was derived from the wood waste biomass was mixed with biodiesel. The blend fraction of the bio-oil to biodiesel was 50%, 30% and 10% in volume.

The hydrocarbon composition and the physical property of the blend oils, biodiesel and diesel fuel, respectively are shown in Table 3.1 and plotted in Figure 3.1 (in x-axis, 0% red: diesel, 0% black: biodiesel).

The density of the blend oils slightly dropped but still out of the standards of diesel fuel (JIS 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 max number of (ASTM975, 4.1 in max. for No. 2-D) (EN590 2013, 4.5 in max.). There was a bit drop in the oxygen content but decreased in half of the raw pyrolysis oil in the water content. The heat value was recovered up to 77% of that of diesel even for the 50% fraction of the raw bio-oil which was half of the diesel.

Table 3.1 Properties of the test oils

Chemical					
%wt/wt	Diesel [JIS2]	Bio diesel	10% Blend	30% Blend	50% Blend
C	85.04	83.21	75.57	74.15	73.90
H	13.55	11.38	11.25	11.14	11.00
N	0	0.1	0.1	0	0.1
S	0	0	0	0	0
O	0	4.88	14.01	15.21	16.22
Physical					
Density [kg/cm ³] at 15 degC	0.830	0.861	0.873	0.903	0.939
Kinematic viscosity [mm ² /s] at 40 degC	2.744	3.800	3.911	4.157	4.222
Water content [%wt/wt]	0	0	6.37	7.53	8.20
Low Heat value [MJ/kg]	45.0	41.0	36.4	35.6	35.1

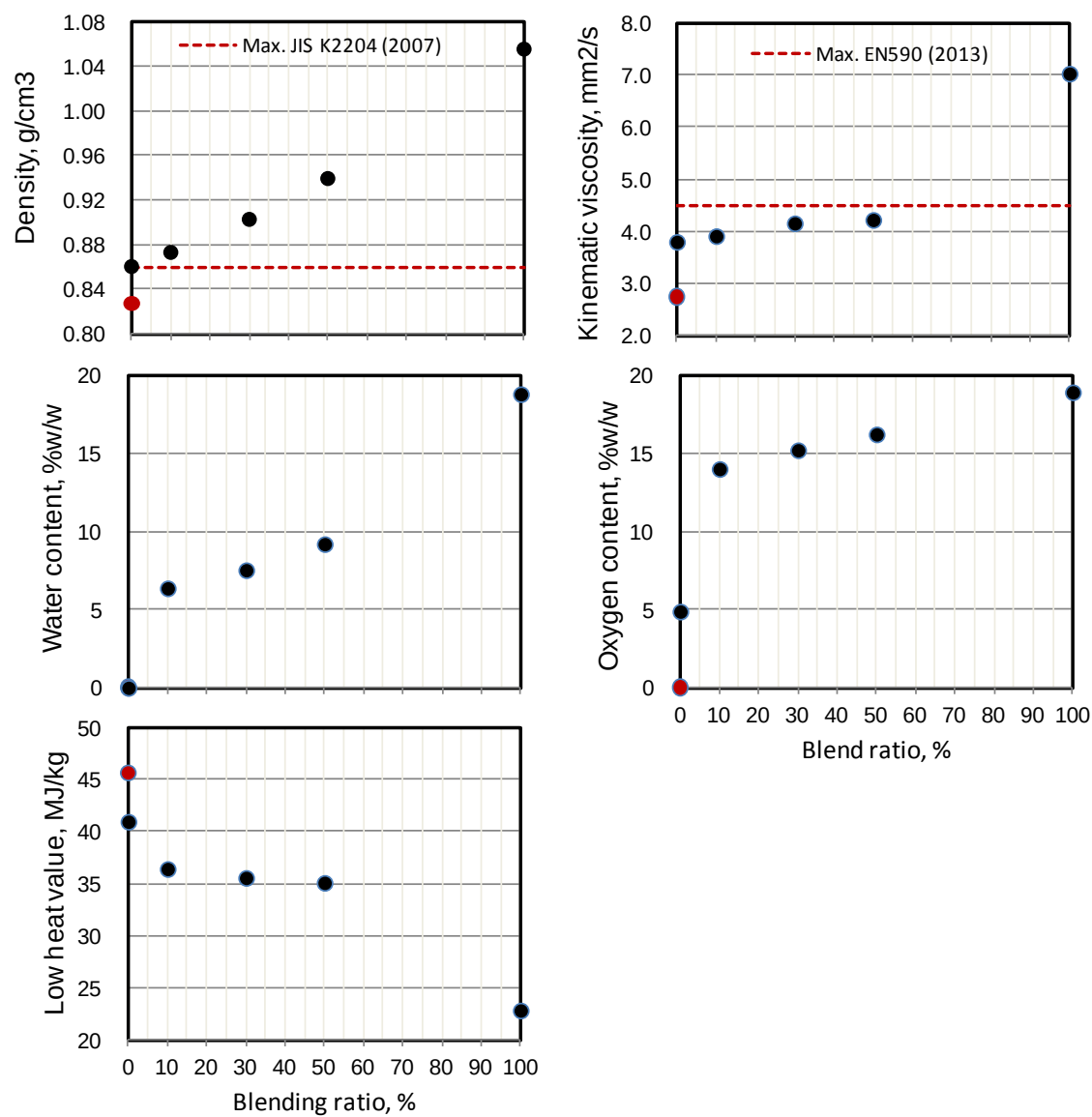


Figure 3.1 Property variance of the blend oils (0% red: diesel, 0% black: biodiesel).

3.1.2 Test Engine

For the engine test, a single cylinder direct fuel injection diesel engine was used and Table 3 shows the engine specification [3-1][3-2]. The engine is very small and turns out a low engine output. The combustion chamber shape was a re-entrant type. The fuel injection timing was 17.5 ± 0.5 deg. BTDC CA and the injection fuel pressure was 20 MPa, built by the mechanical fuel delivery pump for all tests.

Figure 3.2 shows the experimental equipment for the engine test. It is consisted of a diesel engine, measurement devices and a dynamometer which controls 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). The total hydrocarbon (THC) and the smoke concentration 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 a NDIR type exhaust gas analyzer (Iwatadengyo Co., Ltd FAST-3100). The intake air temperature and the exhaust gas temperature were measured by a K-type thermocouple with the diameter of 1.0 mm. The fuel consumption was measured by using the electric 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.

Table 3.1 Test engine specification

Engine specification	
Cylinder number	1
Bore X Stroke	70mm X 57mm
Displacement volume	219cm ³
Compression ratio	20.6
Aspiration type	Natural
Rated power	3kW/3600rpm
Combustion chamber	Re-entrant
Injection pressure	20MPa
Injection timing	17.5 ± 0.5 deg. BTDC
Number of injection hole	4
Diameter of injection hole	0.22mm
Diesel fuel spray angle	95 deg

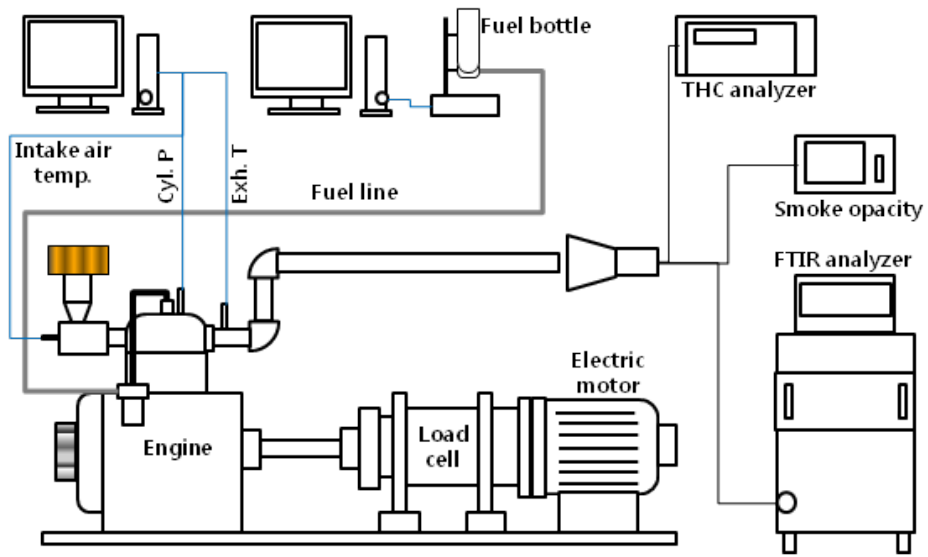


Figure 3.2 Experimental apparatus

At first, the test run fueled by the diesel was conducted in order to check whether the test conditions such as engine performance and 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.

3.1.3 Test engine operation points

As previously reported in the section 1.5 of chapter 1, the engine speed and the torque were specified based the EPA regulation. 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 results, R25 (25% load at the rated speed) 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 3.2.

Table 3.2 Specified operation point of engine test

Operation point	Engine speed	Torque	Power
-	rpm	Nm	kW
Idle	1500	0	0
I50	2450	3.65	0.94
I75		5.47	1.40
I100		7.3	1.87
R10	3500	0.86	0.32
R50		4.3	1.58
R75		6.45	2.36
R100		8.6	3.15

3.2 Results and discussion

3.2.1 Full load performance

Figure 3.4 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 the black in the 5% the x-axis and three blend oils, the 10%, 30% and 50% blending ratios. For all the full load tests, the throttle position was fully opened as mentioned in the section 3.1.3. The maximum reachable torques were almost the same excepting for the case of the 10% blend and the 50% blend for the case of 2450rpm. The standard deviation of the engine torque went slightly worse for the 50% blend in the case of 2450rpm than other test cases but the engine was stably operated even for the 50% blend.

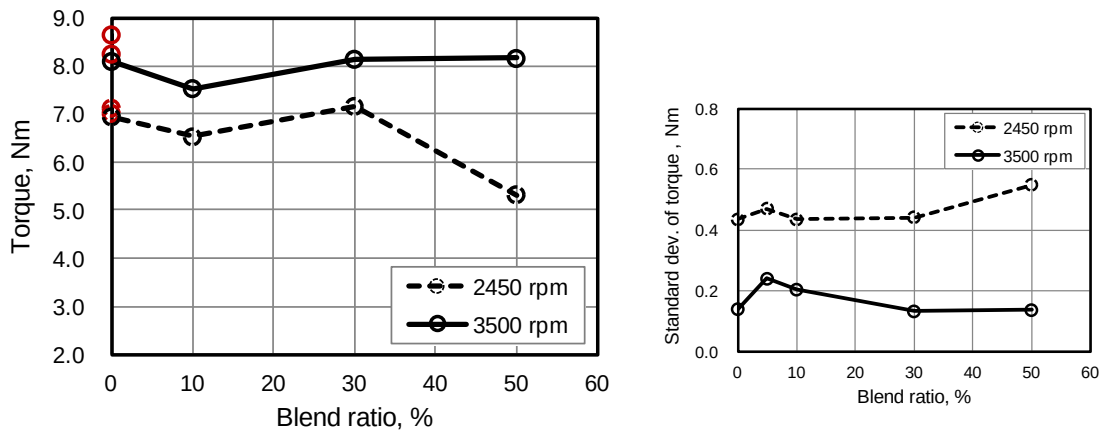


Figure 3.4 The maximum reachable torque (left) and the standard deviation of the torque (right) as a function of the blend ratio (0% red : diesel, 0% black: biodiesel)

For the analysis of the torque decrease, the injection amount per the single combustion is plotted in Figure 3.5. The injection amount increased with the elevation of the blend ratio due to the increase of the density. In Figure 3.6, 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 2450rpm. This means that the torques 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 and another is that the injection amount deteriorated due to the oil property which was caused by the blending.

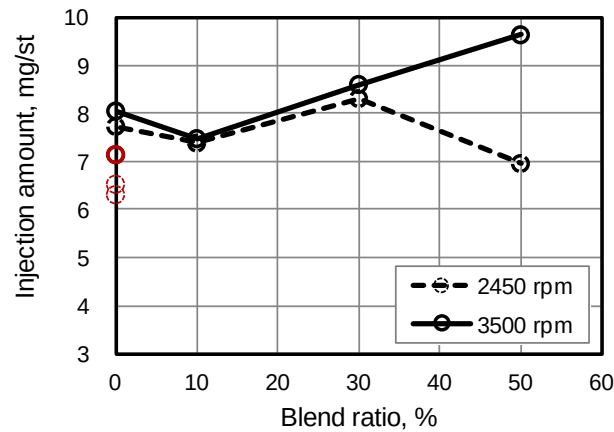


Figure 3.5 Injection amount per single combustion as a function of the blend ratio
(0% red : diesel, 0% black: biodiesel)

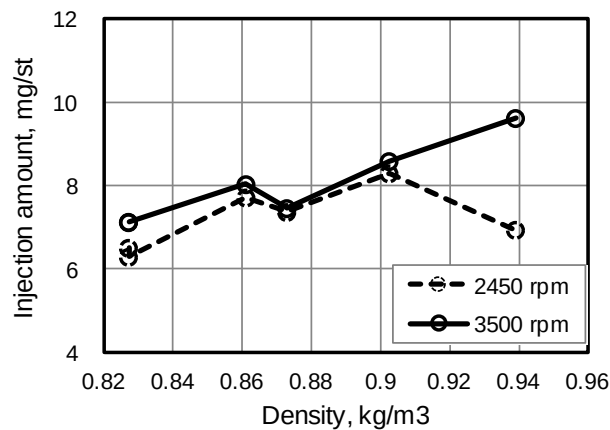


Figure 3.6 Injection amount per single combustion as a function of the density of the oils

For more detail investigation, combustion analysis was done. Figure 3.7 and 3.8 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 Figure 3.9. In overall, the ignition timing became delayed and the fraction of the premixed combustion phase increased with 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 the former

reason, mechanical linkage loosening. For the torque of the blend 50% in the case of 2450rpm, the ignition delay seems to be more 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 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 Figure 3.1. But the total amount of heat value per single combustion was compensated by the increased injection amount due to the increased injection rate driven by the higher density. Figure 3.10 shows the simple calculation by multiplying the injection amount shown in Figure 3.5 with the heat value listed in Table 3.1.

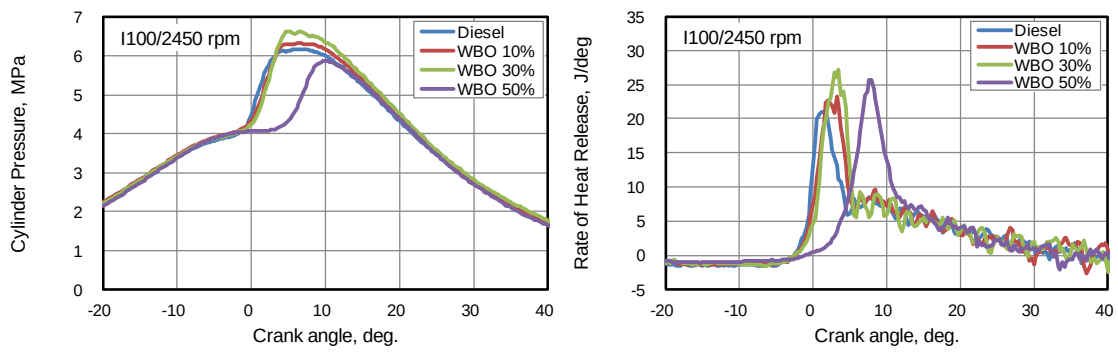


Figure 3.7 Cylinder pressure and rate of heat release at I100, 2450rpm

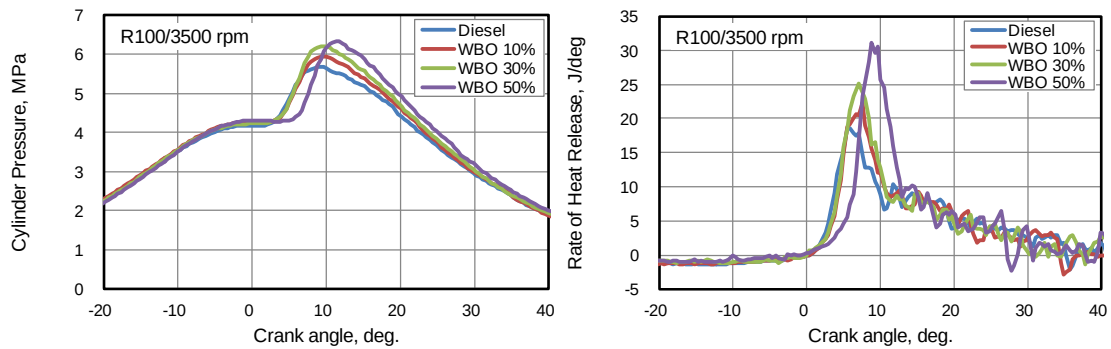


Figure 3.8 Cylinder pressure and rate of heat release at R100, 3500rpm

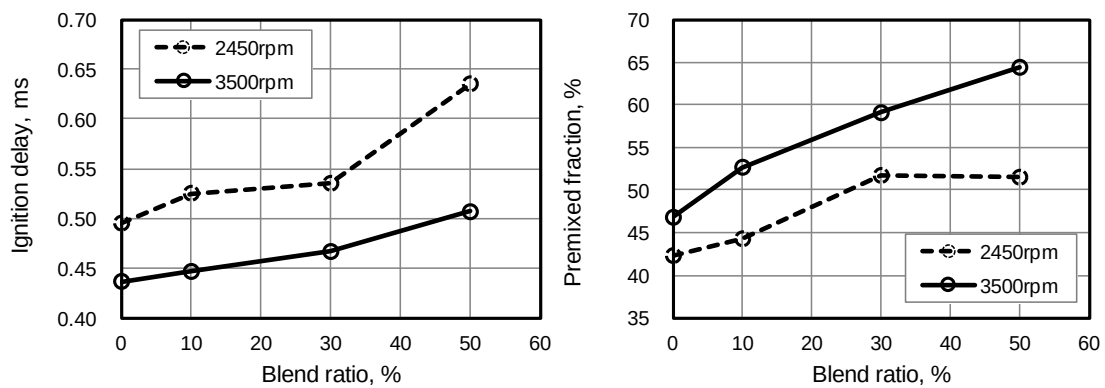


Figure 3.9 Ignition delay and fraction of premixed combustion phase as a function of the density of the oils (0: 100% diesel)

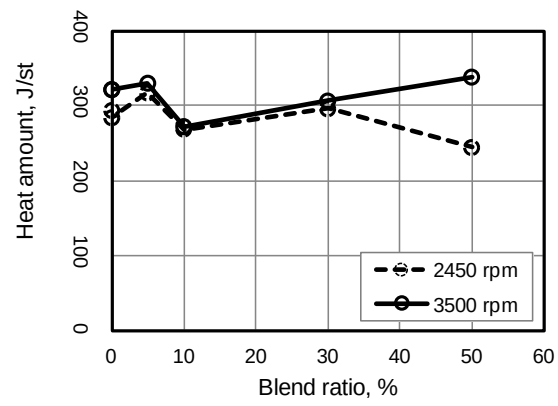


Figure 3.10 Heat amount per single combustion as a function of the blend ratio
(0: 100% diesel, 5%: biodiesel)

3.2.2 Exhaust emissions

3.2.2.1 Ignition delay

For the combustion analysis, in Figure 3.11, the ignition delay time and the fraction of the premixed combustion phase were summarized based on the combustion related data which are illustrated in Figure 3.13. The ignition timing was delayed with the increment of the blend ratio [3-3][3-4][3-5][3-6][3-7][3-8][3-9] and it means that the higher the blend ratio is, the longer the time is required[9]. The longer ignition delay time led to enlarge the fraction of the premixed combustion phase with the blend ratio. The tendency was more significant for the 50% blend. The ignition delay became slightly shorten at 3500rpm because of the elevation of the gas temperature at the end of the compression stroke as depicted in Figure 3.12 due to the higher compression intensity by the increased engine speed.

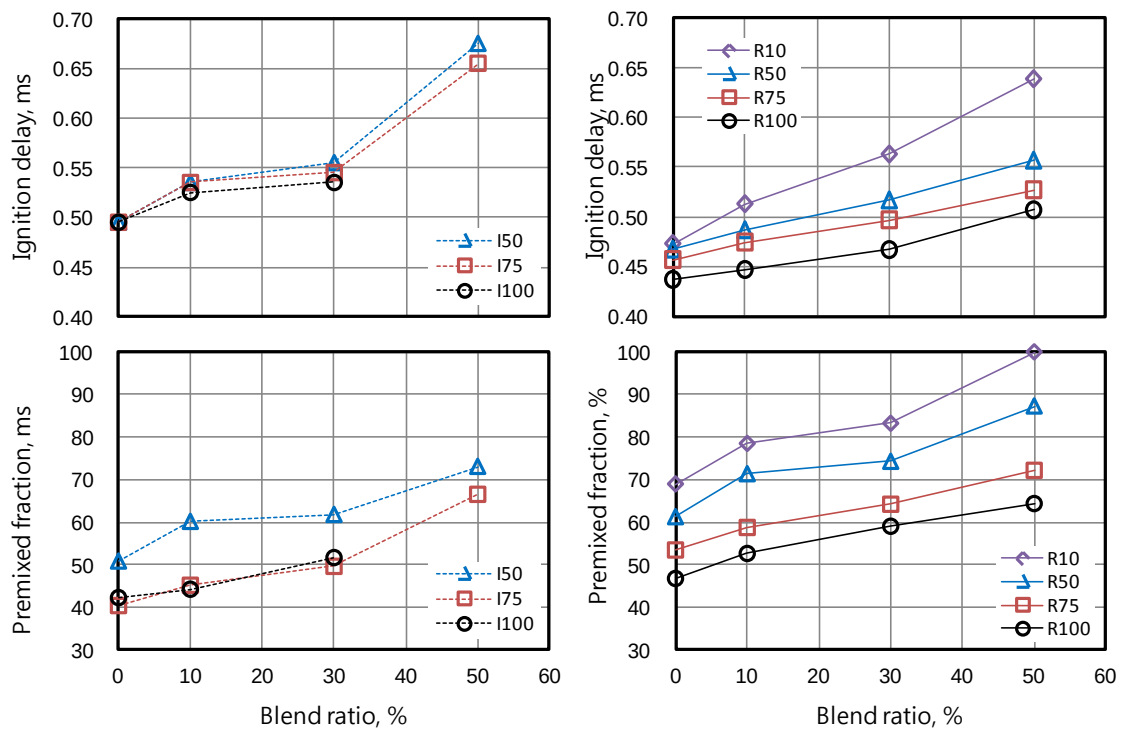


Figure 3.11 Ignition delay and fraction of premixed combustion phase as a function of the blend ratio at 2450rpm (0: 100% diesel)

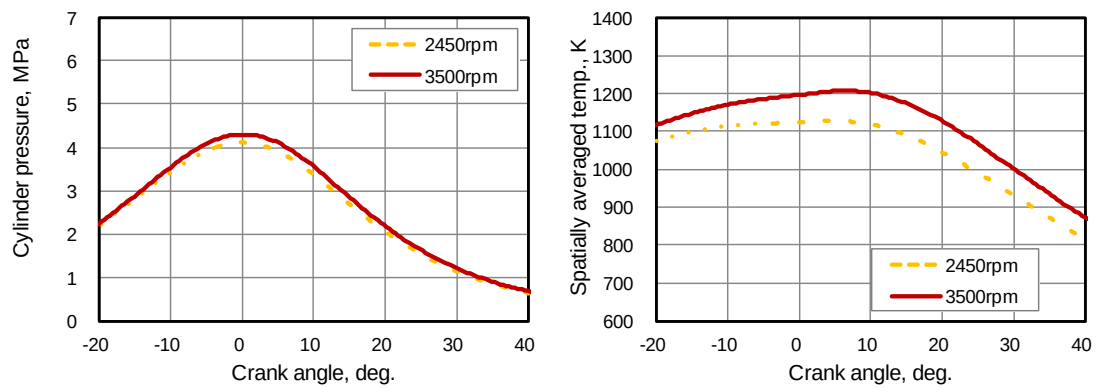


Figure 3.12 Cylinder pressure and spatially averaged temperature at motoring

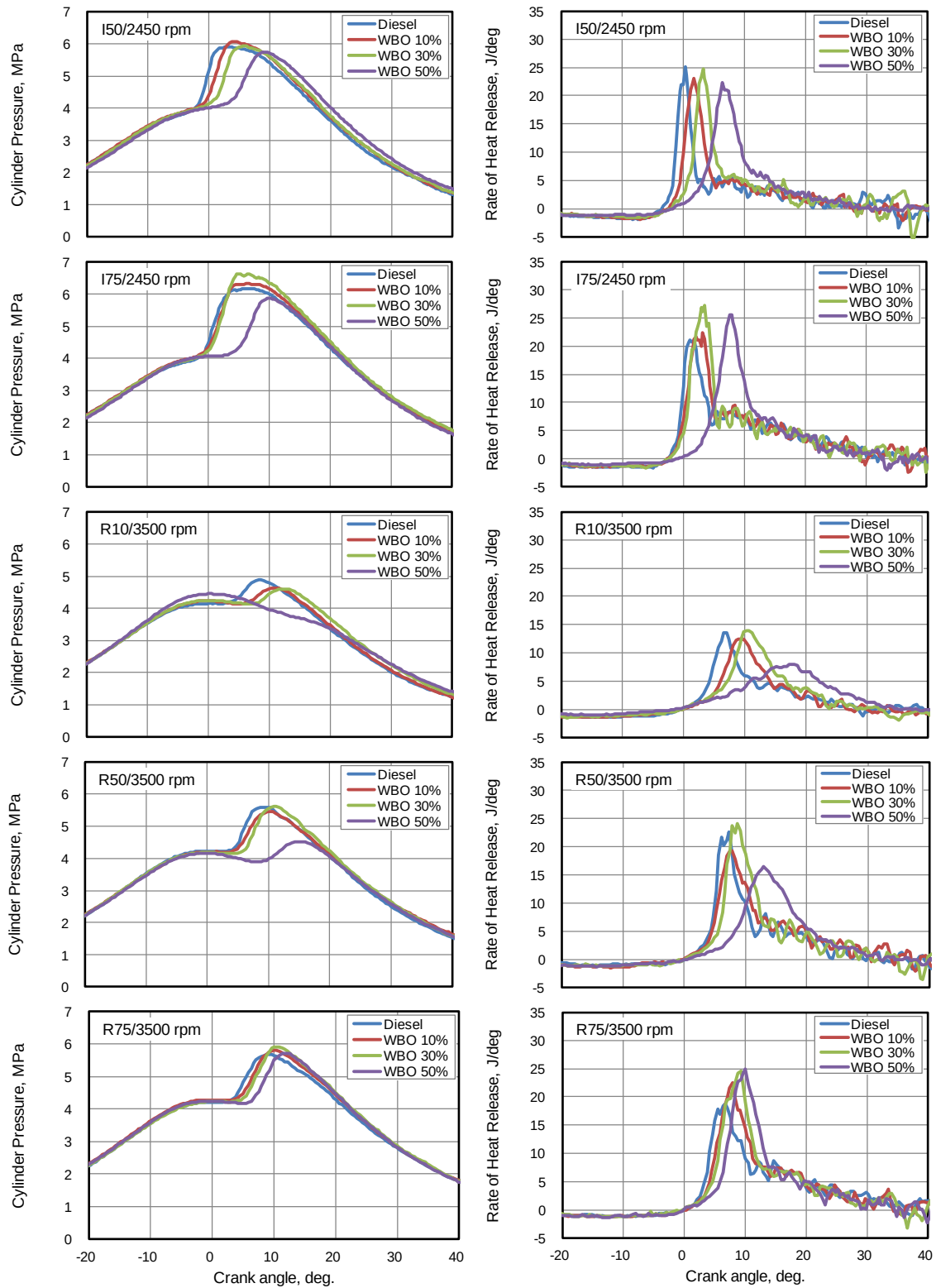


Figure 3.13 Cylinder pressure and rate of heat release

3.2.2.2 NOx emission

In Figure 3.14, NOx emission is plotted in the forms of the concentration, the mass flow rate and the specific emission at 2450rpm and at 3500rpm.

With the rise of the blend ratio, the NOx emission decreased for the case of 2450rpm and this tendency was clear for R10 but diminished with the load increase at 3500rpm [3-3][3-4][3-10]. NOx emission tends to increase with higher premixed combustion rate which leads to the elevation of the flame temperature [3-11]. But the NOx emission went to the opposite direction or showed constant as shown in the left side picture of Figure 3.15 (Normalized ones for the diesel as 100%). The NOx emission might be affected by the water content in the blend oils as shown in the right side picture [3-12][3-13][3-14][3-15][3-16][3-17]. For the case of 3500rpm, this NOx reduction by the increase of the water content was not observed.

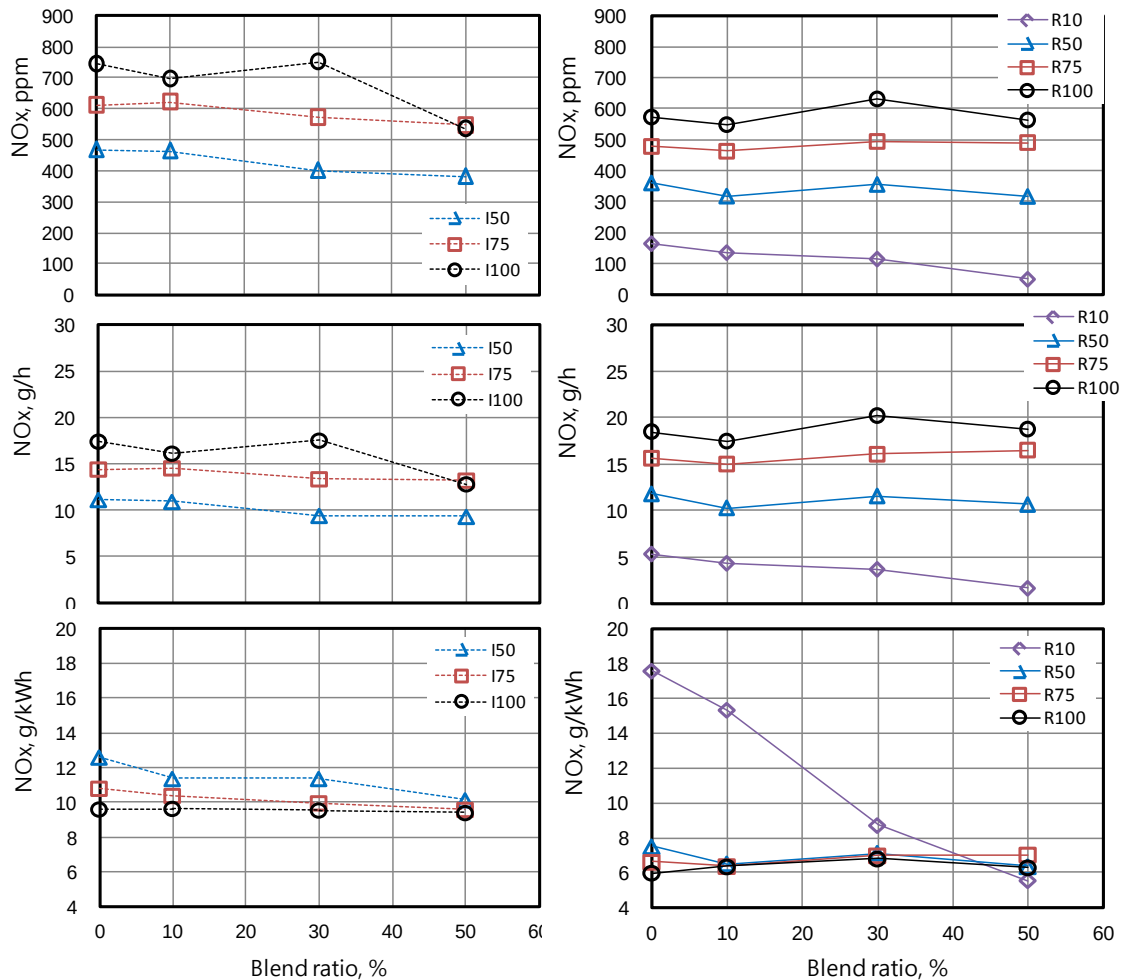


Figure 3.14 NOx emission in the forms of the concentration [ppm], the mass flow rate [g/h] and the specific emission [g/kWh]: left at 2450rpm, right at 3500rpm (0: 100% diesel)

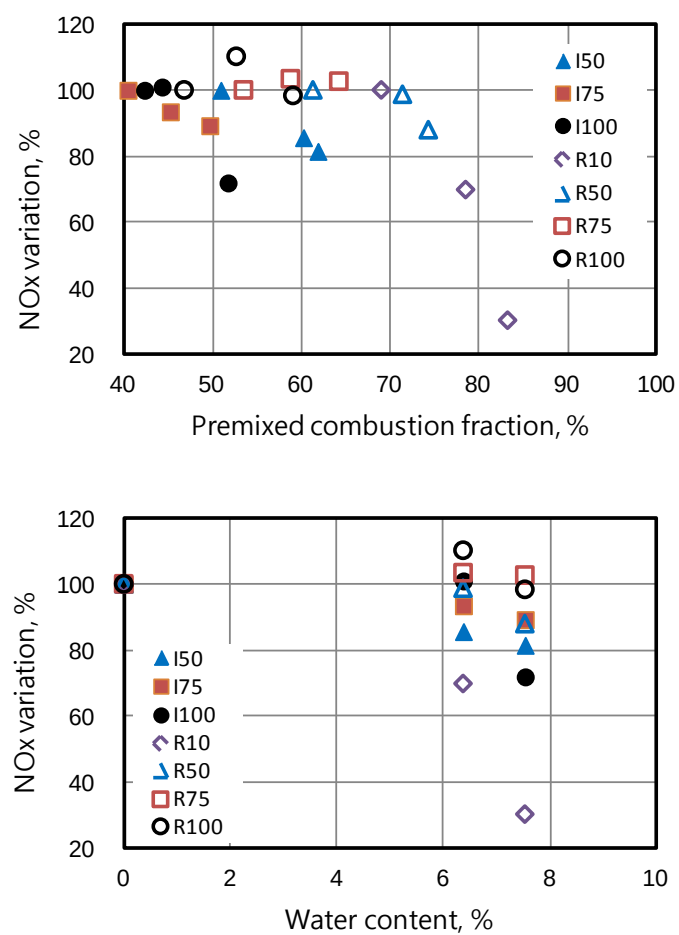


Figure 3.15 NOx emission variation as a function of the fraction of the premixed combustion (top) and the water content in the oils (bottom)

3.2.2.3 CO emission

Figure 3.16 shows the CO emission as a function of the blend ratio. The increase of the blend ratio resulted in the higher CO emission for both engine speeds. Especially, in R10, the CO emission turned out double higher for the 30% blend than that for diesel and around 4 times higher for the 50% blend due to the misfire which was led by the enlarged ignition delay. Figure 3.17 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 radical can promote the 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 [3-18]. 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 [3-19][3-20].

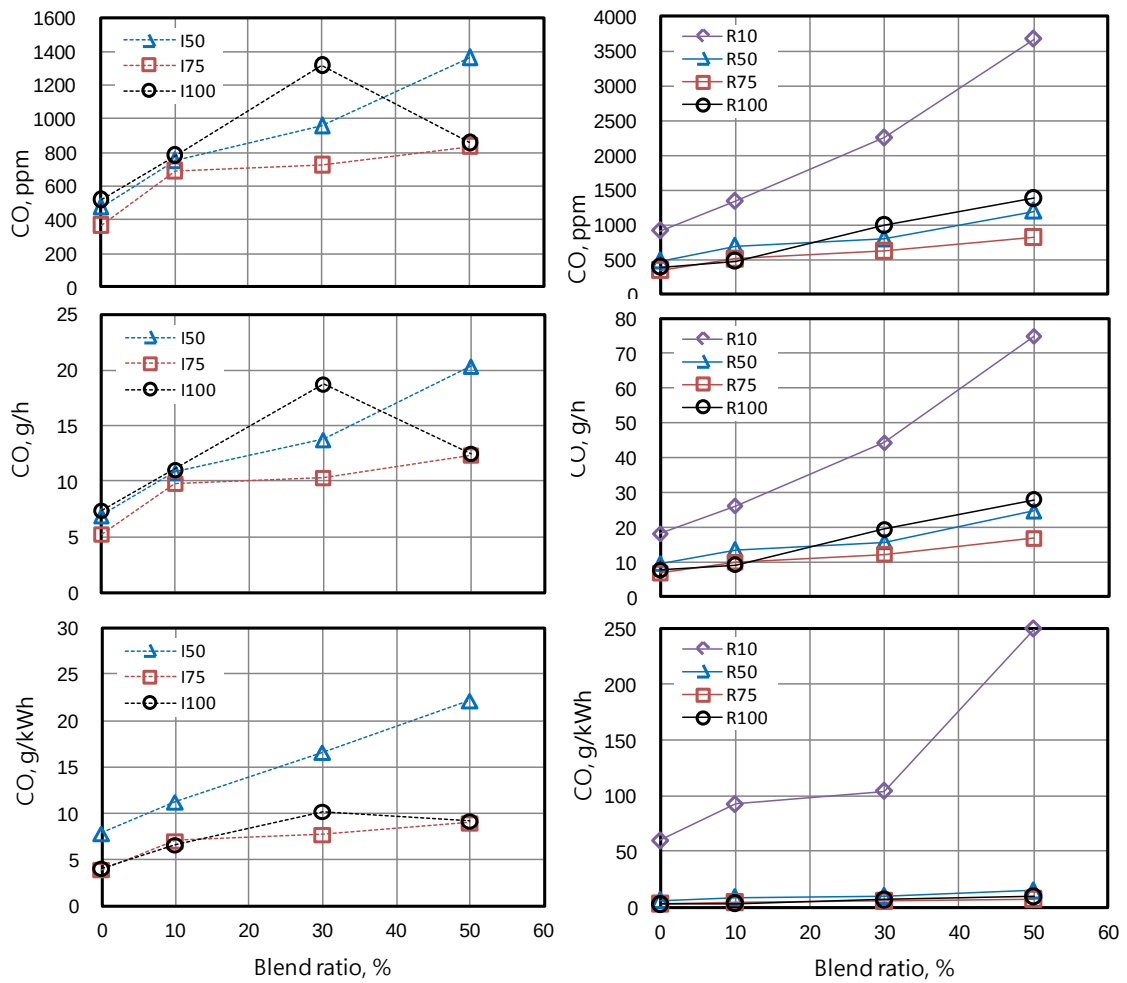


Figure 3.16 CO emission in the form of the concentration [ppm], the mass flow rate [g/h] and the specific emission g/kWh]: left at 2450rpm, right at 3500rpm (0: 100% diesel)

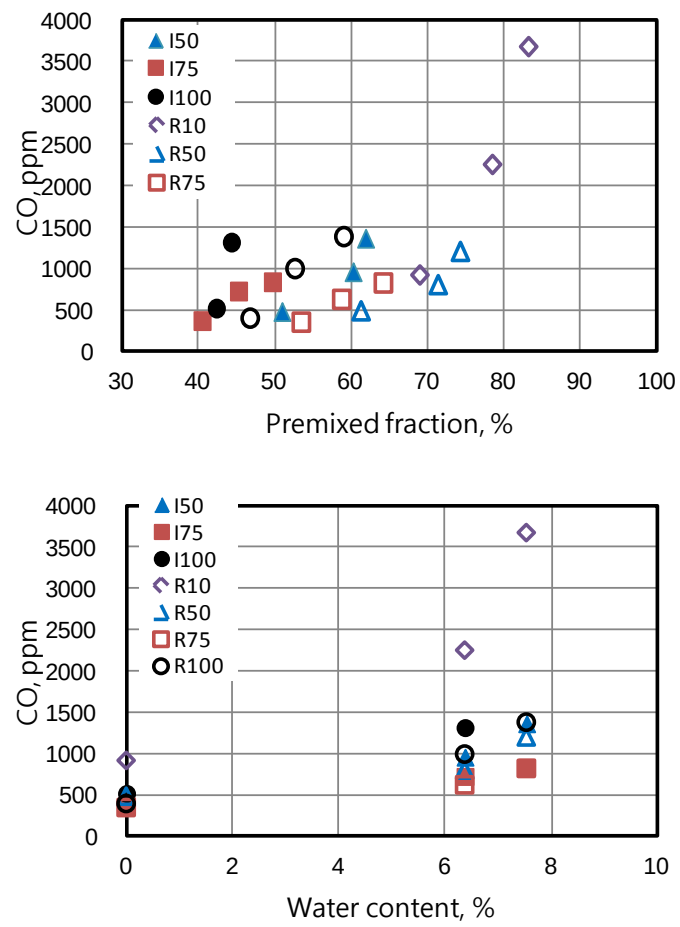


Figure 3.17 CO emission as a function of the fraction of the premixed combustion (top) and the water content in the oils (bottom)

3.2.2.4 Smoke emission

The smoke emission is illustrated in Figure 3.18, where for the 10% blend, the number of smoke is too much 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 calculated based on the injection amount, the air mass amount and the stoichiometric air-fuel ratio which the oxygen amount in the bio-oils 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 as mentioned in the previous section. And this made the lambda of the blend oils to be gradually reduced as depicted in Figure 3.19. In Figure 3.20, the smoke opacity was re-plotted as functions of the lambda and the kinematic viscosity and showed a certain correlation especially in a high load. In addition to the effectiveness of the lambda, the spray condition might affect on the smoke opacity. The viscosity and the surface extension of the biomass pyrolysis oil are higher than those of the diesel and thus lead to the poorer atomization [3-21][3-22]. 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, 0.28mm hole diameter, 210bar opening pressure) and the break-up length and the cone angle were smaller for the blend oil comparing to the diesel fuel. This resulted in the worsen THC emission and there was no smoke data. The Sauter mean diameter (SMD) of the raw biomass pyrolysis oil was greater than the diesel fuel [3-23]. The SMD observed became higher with the increase of the kinematic viscosity [3-24]. 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.

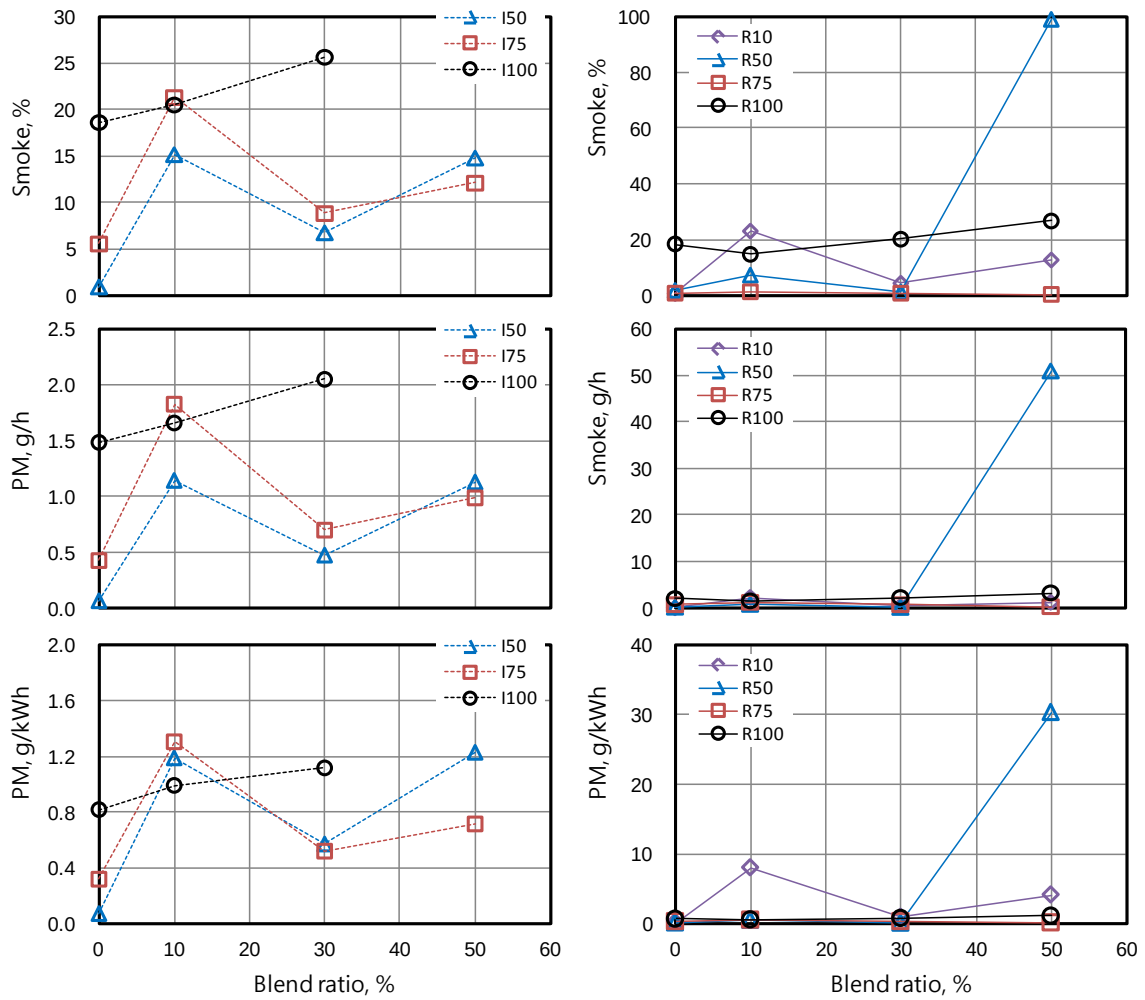


Figure 3.18 Smoke emission in the form of the opacity [%], the mass flow rate [g/h] and the specific emission [g/kWh]: left at 2450rpm, right at 3500rpm (0: 100% diesel)

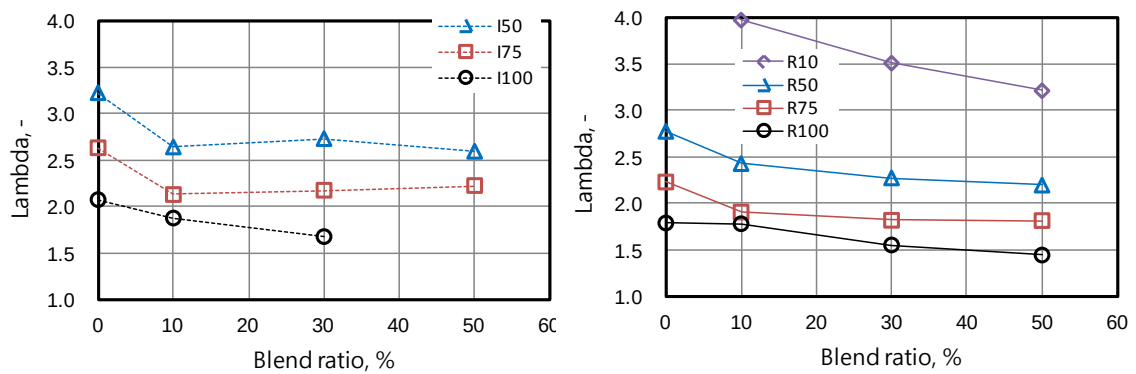


Figure 3.19 Lambda : left at 2450rpm, right at 3500rpm (0: 100% diesel)

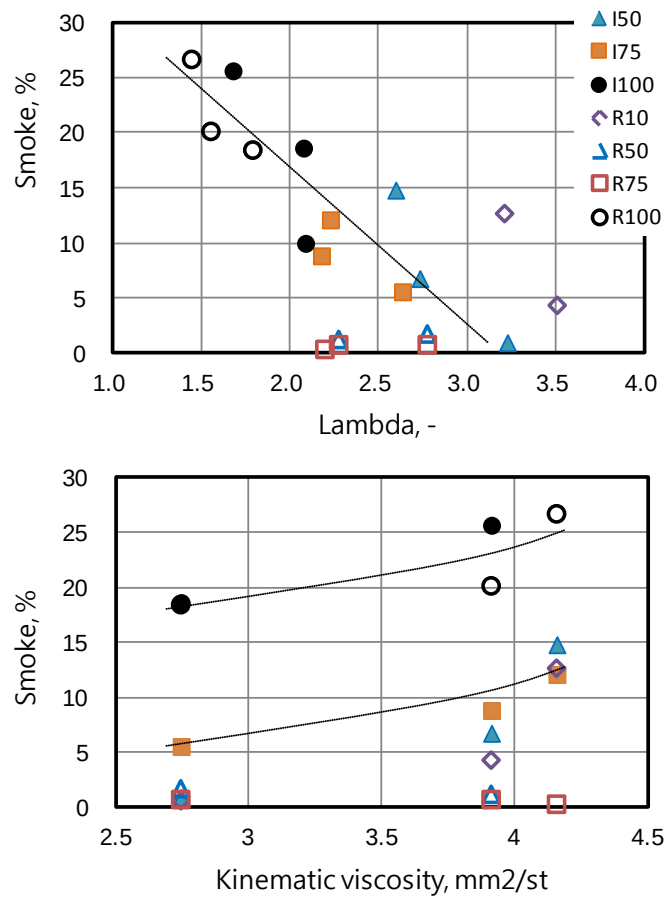


Figure 3.20 Smoke emission as a function of the fraction of the premixed combustion (left) and the kinematic viscosity of the oils (right)

3.2.2.5 THC emission

The THC emission is depicted in Figure 3.21. The THC emission increased up to the 30% blend ratio at 2450rpm. At 3500rpm, the THC emission was higher for the blend oil in the range of low loads. 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 is 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.

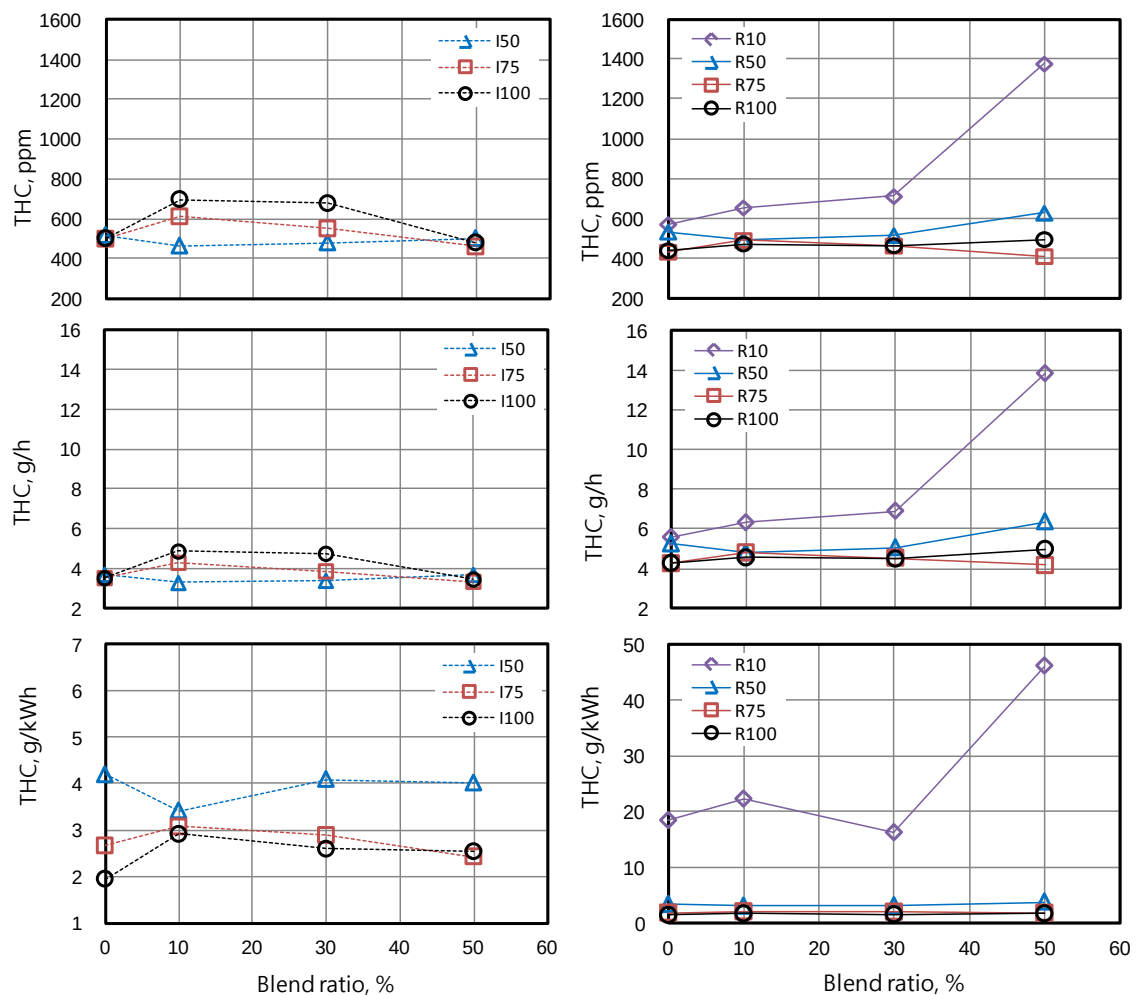


Figure 3.21 THC emission in the form of the concentration [ppm], the mass flow rate [g/h] and the specific emission [g/kWh]: left at 2450rpm, right at 3500rpm (0: 100% diesel)

3.2.2.6 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.

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 lower than those of the 10% and 30% blends and went slightly over the limit. The weighting factor which was shown in the section 1.6 of Chapter1 is around double of C1-8 mode in part load, and as shown in Figure 3.14, the NO_x specific emission of the diesel in R10 was pretty higher than those of the 10% and 30% blends. The test engine was not new one and had been run with various non regulated fuels such as biomass oils and other pyrolysis oils. All emission measurement devices were not fully pre-conditioned for the engine test like an 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 D5-mode became worse than the diesel due to the explosive THC emission caused by the misfire as shown in Figure 3.21.

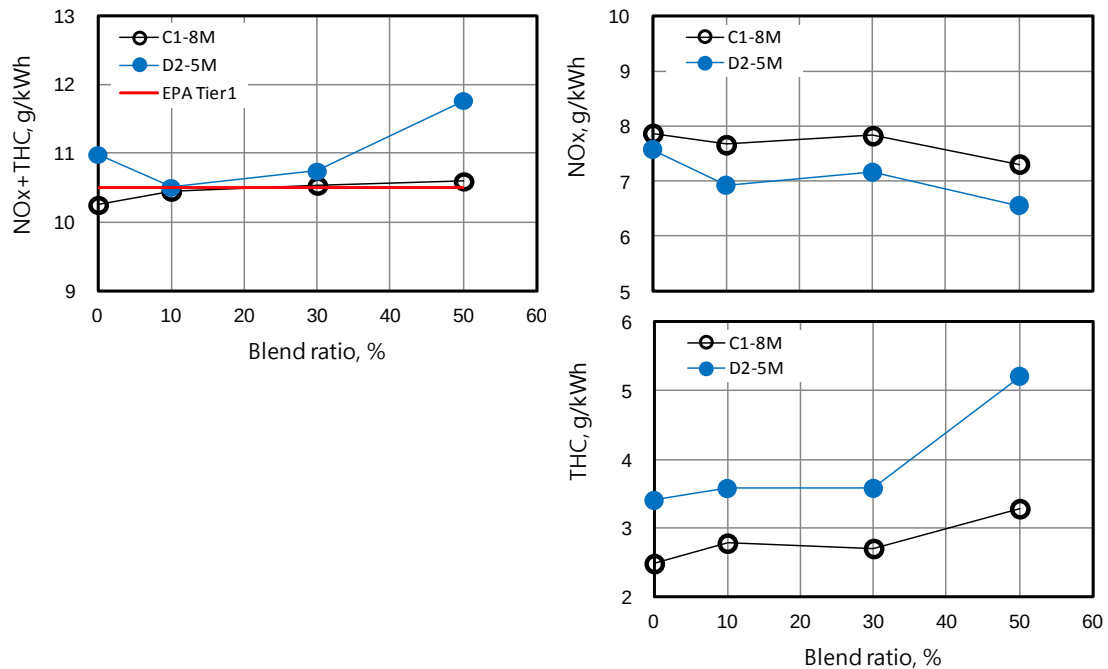


Figure 3.22 NO_x+THC mode emission for the diesel fuel (0%) and the blend oils

Figure 3.23 illustrates the result of the mode CO and PM exhaust emission. The CO emission result positioned around the regulation limitation (8g/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 much worsen lambda at full loads and the misfire at R10, the PM emission of the 50% blend went far over the limitation.

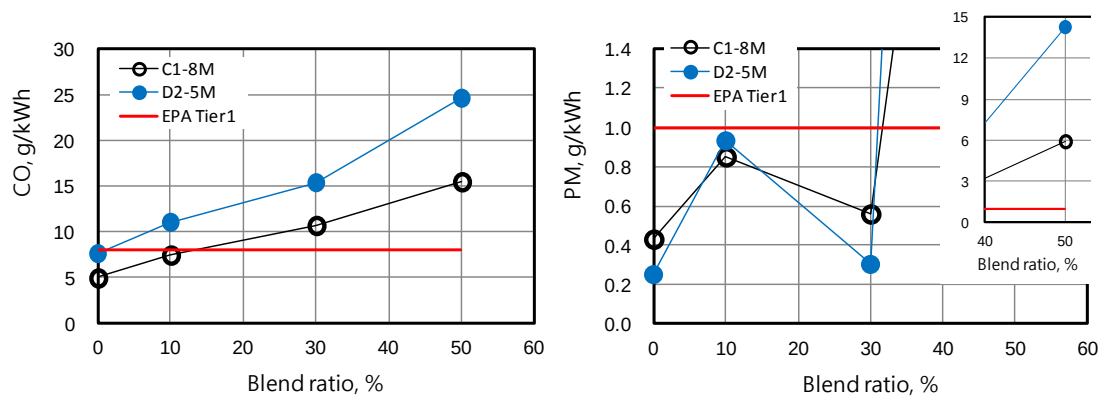


Figure 3.23 CO (left) and PM (right) mode emission for the diesel fuel (0%) and the blend oils

3.2.2.7 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 [3-20]. In this report, toluene (C_7H_8), acetaldehyde (C_2H_4O) 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 Figure 3.24. 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. But in almost all operation points up to the 30% blend, the portion was around 10% (except for R10 where the operation is pretty scarce in the real world) and was not nearly differed with that of biodiesel, whereas for the 50% blend, the toxic portion rose double to three times depending on the load at the rated engine.

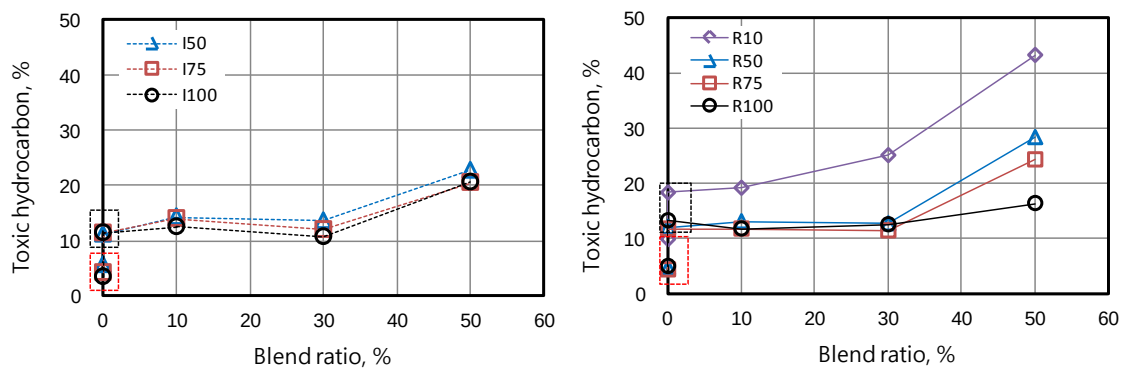


Figure 3.24 Comparison of the portion of the toxic hydrocarbon in the THC
(0: 100% diesel, 5%: biodiesel)

3.2.3 Specific fuel consumption

Figure 3.25 depicts the specific fuel consumption (SFC) and the brake thermal efficiency. The specific fuel consumption increased with the increase of the blend ratio due to the higher density and the lower heat 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.

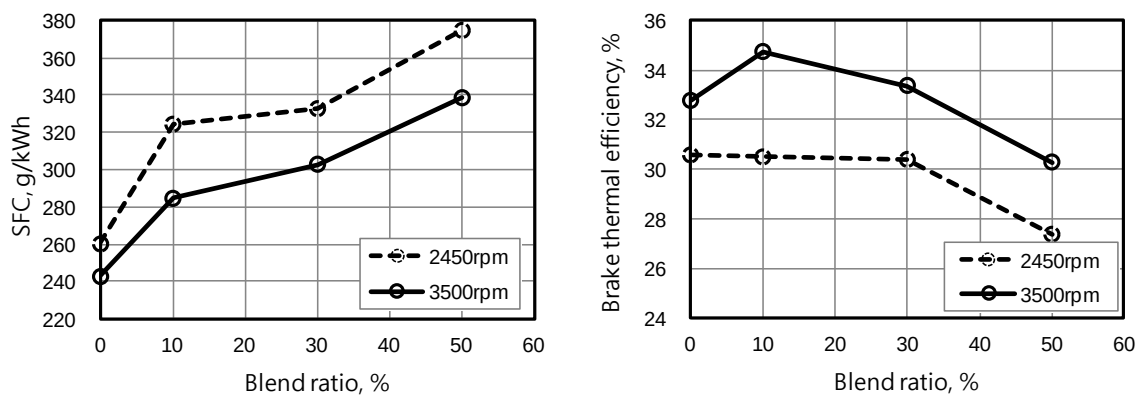


Figure 3.25 Specific fuel consumption (right) and brake thermal efficiency (left) as a function of the blend ratio (0%: diesel)

The possible full load operation point for the electricity generation use according to the engine is illustrated in Figure 3.26. 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 2450 rpm that might be caused by the decreased injection rate due to the higher blend ratio.

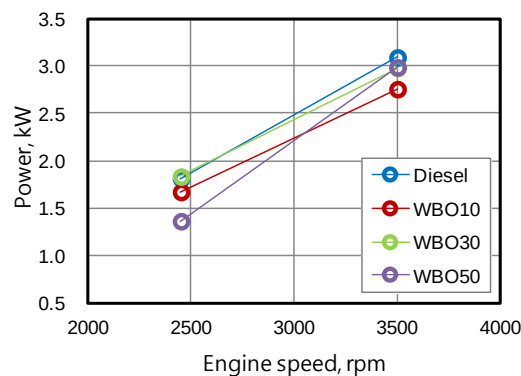


Figure 3.26 Possible engine operation points for the electricity generation

3.3 Conclusion

The pyrolysis oil was produced from waste biomass in a pilot scale pyrolysis plant. The raw pyrolysis 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 fuels with diesel fuel.

Up to 30% blending ratio, the engine operation compatible to the diesel was observed with regard to the full load engine output, the exhaust emissions and the specific fuel consumption.

The maximum engine power was shown to be the same level.

Mode exhaust emissions, C1-8 mode(Non-road mobile) and D2-5mode(Gen-set), resulted as:

The NO_x+THC emission was placed around the regulation limit and it might be acceptable considering the experimental conditions

The CO emission was positioned around the regulation limitation (8g/kWh) only for the C1-8 mode of the 10% blend and became worse around double the limit for the 30% blend at the D2-5mode. The CO emission in the low load at 3500rpm should be careful with regard to the variation of the composition of the bio-oil.

The PM emission (opacity conversion to mass) was slightly higher than that of diesel but approximately half magnitude of the legal limit.

The specific fuel consumption increased with the increase of the blend ratio due to the higher density and the lower heat value.

For the 50% blending ratio, the maximum engine power at 3500rpm was compatible to the diesel. Nevertheless, all the exhaust emissions went far over the limitations. The toxic hydrocarbon ratio reached about 4 times larger than that of the diesel.

Therefore, It is found that the present bio-oil can be utilized in the diesel engine. The blend ratio with the biodiesel is appropriate up to around 30%.

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Chapter 4

Engine performance test of the waste plastic pyrolysis oil in a direct injection diesel engine

In this chapter, plastics from municipal wastes were converted into oil through the pyrolysis and the catalytic reforming process in a commercial facility. Compared with diesel fuel, the raw pyrolysis oil showed lower kinematic viscosity than the minimum level of diesel fuel and almost the same heating value. Its carbon class differed from diesel, gasoline and kerosene and is mainly composed of naphthenes and olefins which have poor self-ignition quality. A single cylinder direct injection diesel engine was used for the test to show the compatibility of the pyrolysis oil to diesel fuel. The pyrolysis oil was blended with diesel fuel with different mixing ratios. The full load performance, the exhaust emission and the thermal efficiency were investigated from the view point of the compatibility to diesel based on the US EPA regulation mode.

Through the evaluation, the effectiveness of the pyrolysis oil adaption to diesel engine will be examined and the acceptable blend ratio will be additionally defined.

4.1 Materials and methods

4.1.1 Test fuels

The raw pyrolysis oil was mixed with diesel fuel and the mixing ratio was 20, 40, 60 and 80% where the number was the volume percentage of the raw pyrolysis oil to the diesel fuel. In the engine experiment, low mixing ratio blend oil was tried first considering the poor self-ignition quality. By the blending, the physical property of the raw pyrolysis oil was somewhat improved as shown in Figure 4.1. The density of the blend oils was still positioned below the standards of diesel fuel (EN590 2013, 0.82 in min.). The kinematic viscosity of the blend oils increased and was placed above the min number of (ASTM975, 1.3 in min. for No. 1-D). The heat value per unit mass did not differ between the test oils and the heat value per unit volume of blend oils was lower than that of diesel fuel due to the lower density of the pyrolysis oil as shown in Figure 4.2.

Figure 4.1 Density and kinematic viscosity as a function of the blend ratio (0%: diesel)

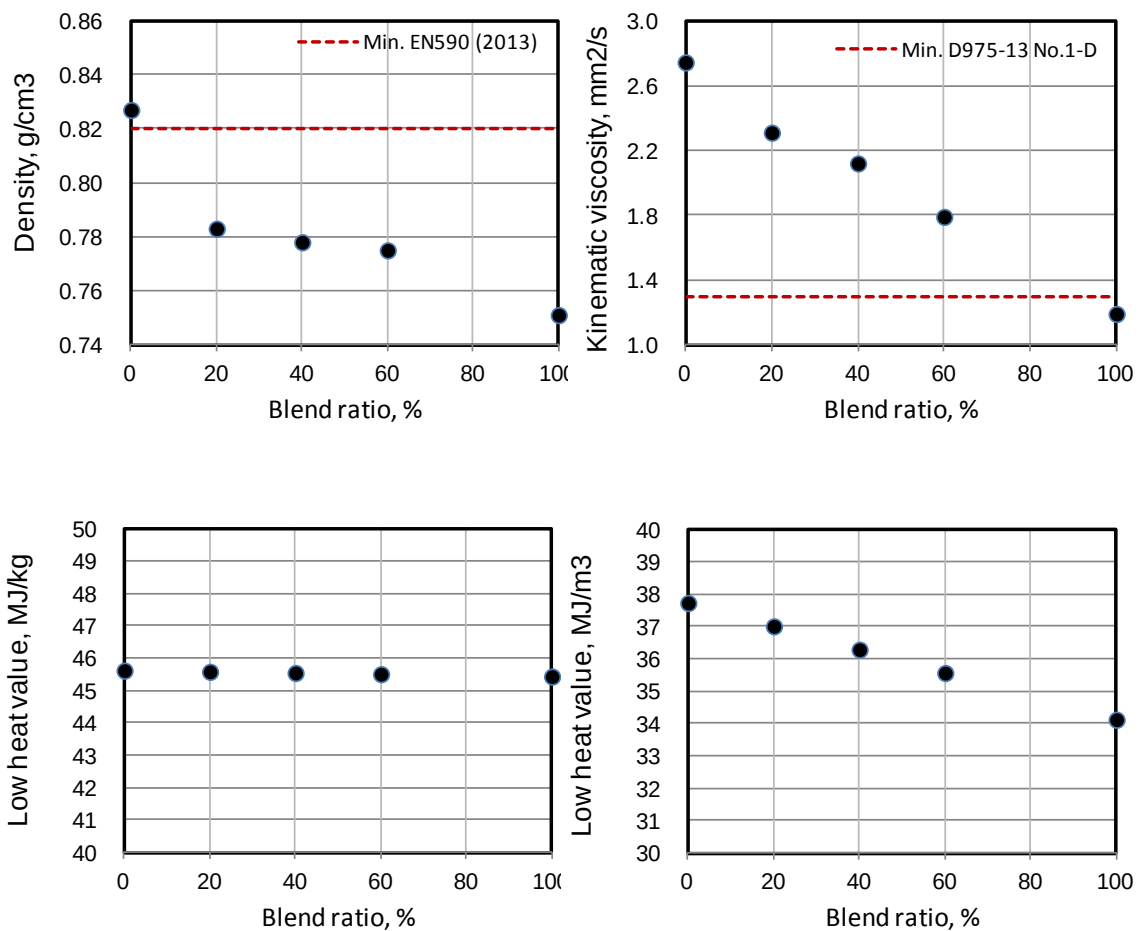


Figure 4.2 Low heat value as a function of the blend ratio (0%: diesel)

4.1.2 Test Engine

For the engine test, the single cylinder direct fuel injection diesel engine which was introduced in the chapter 3 was used

4.1.3 Test engine operation points

As previously reported in the chapter 3, the same test procedure was applied to this engine experiment

4.2 Results and discussion

4.2.1 Full load performance

Full load performance was evaluated upon 100% diesel which is expressed as the blend ratio 0% point on the horizontal axis and the two blend oils, the 20% and 40% blend ratios. For all the full load tests, the throttle position was fully opened as mentioned above. The maximum reachable torque is shown in Figure 4.3 indicating that it was drastically decreased with the increase of the blend ratio.

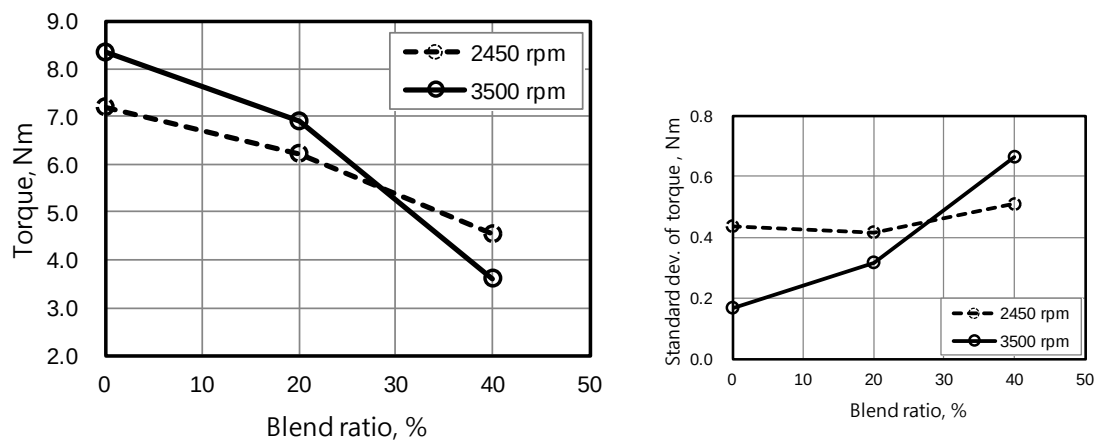


Figure 4.3 Maximum reachable torque (left) and standard deviation of the torque (right) as a function of the blend ratio (0: 100% diesel)

Up to the 20% blend ratio, comparable engine running was achieved even though the full load output was declined about 13% at 2450rpm and 17% at 3500rpm. The running stability was observed in the same level for the intermediate engine speed and became worse at the rated speed but it was acceptable level because the engine operation was continued without any troubles. For the 40% blend oil, the maximum torque dropped around 37% at the intermediate speed and 56% at the rated speed which corresponds to the R50 output of diesel. The engine operation stability which is expressed by the standard deviation of the torque depicted in Figure 4.3 (right one) became worse with increasing the blend ratio, and in the case of the 40% blend oil at 3500rpm, the engine could keep running only for a few minutes.

The heat value of the raw pyrolysis oil is almost identical with the diesel fuel and this means there is no change in the heat value by blending the pyrolysis oil with diesel fuel. Nevertheless, the torque of the blend oils dropped and some analysis is followed.

For the case of I100 operation point, the injection quantity (5.8mg/st) decreased (-8%) for

the blend 20% and 4.6mg/st (-27%) for the blend 40% comparing to 6.3mg/st of diesel fuel with increasing the blend ratio as shown in Figure 4.4. It is thought that the physical property of the raw oil such as lower density and kinematic viscosity affected the injection amount [4-1] [4-2]. The density drops, 5.3% for the blend 20% and 5.9% for the blend 40% as shown in Figure 4.1, might not cover all the injection amount drops. In the event of the injection, the rate of injection might be additionally reduced by the lower kinematic viscosity [4-3][4-4]. The relationship between the injected amount and the torque can be simply calculated by dividing the engine torque by the injection quantity. The fuel mass to torque conversion factor is around 1.1 Nm/[mg/st] for the diesel fuel as shown in Figure 4.4. For the 20% blend oil, 6.8Nm can be estimated by using the conversion factor and the injection mass of 5.8mg/st which was fully delivered by the pump but the actual torque gained was 6.2Nm (8% lower). For the 40% blend oil, the calculated torque is 5.2Nm and the actual was 4.5Nm (27% lower).

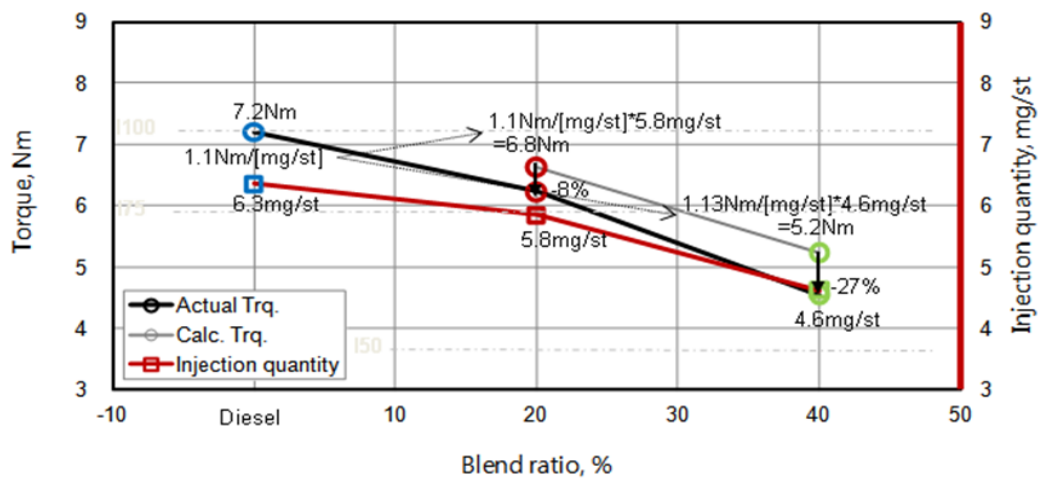


Figure 4.4 Torque and injection quantity as a function of the blend ratio at I100, 2450rpm

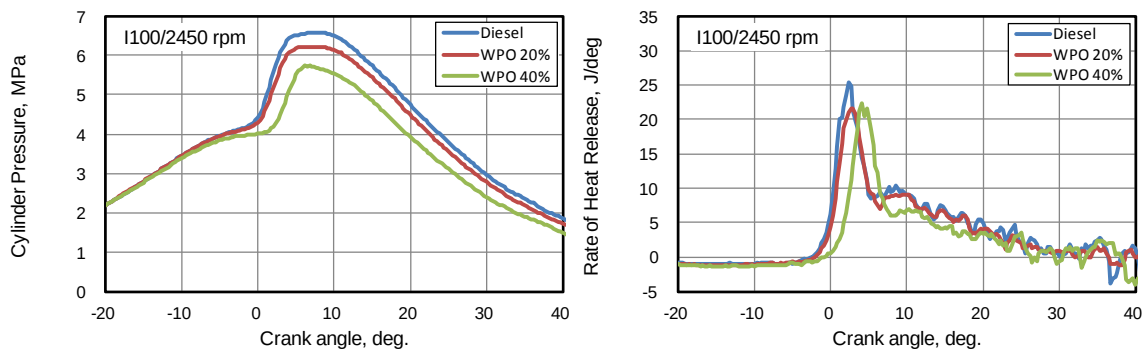


Figure 4.5 Cylinder pressure and the rate of heat release at I100, 2450rpm

For both blend oils, the actual torque output was less than the estimated ones and this went worse with increasing the blend ratio. This can be explained based on the combustion analysis as depicted in Figure 4.5. In the premixed combustion where the fuel-air mixture which is accumulated during the ignition delay defined as duration between the injection timing and the first rise of the heat release rate rapidly explodes, the cylinder pressure rise was slower due to the lower injection rate caused by the physical oil property as described above. The ignition delay became longer with increasing the blend ratio. The enlarged ignition delay is caused by the chemical property of the raw pyrolysis oil. For the 20% blend ratio, even though the ignition timing was slightly delayed, the amount of the heat release in the premixed combustion phase was lower than that for the diesel and the phase was slightly shifted backward. As a result, the torque dropped more than the estimated one. This might be caused by mainly the lower injection rate due to the physical property of the raw pyrolysis oil. For the 40% blend ratio, the ignition delay was significantly enlarged and the premixed heat release phase was moved more than that of the 20% blend. Thus the larger torque down than the calculated one was caused mainly by the ignition delay due to the chemical property of the raw pyrolysis oil.

The identical analysis can be basically propagated for the case of R100 operation point. The injection quantity of the diesel fuel increased more than that of I100 due to the increased pressure build-up driven by the higher engine speed and the injection quantity of 6.9mg/st produced the torque of 8.3Nm and the torque conversion factor was 1.2Nm/[mg/st] as illustrated in Figure 4.6. The estimated torque was 7.6Nm for the 20% blend and 5.4Nm for the 40% blend. The actual torque deteriorated by -9% and -35%, respectively. Those numbers are even larger especially for the 40% blend than those of the I100. For the 20% blend, the amount of the heat release in the premixed combustion phase decreased slightly than that of I100 as shown in Figure 4.7. For the 40% blend, the heat release phase delayed more by greatly enlarged ignition timing due to the higher piston speed and this led to the magnitude of the actual torque drop to the calculation to be larger than the case of I100. The ignition delay will be summarized later including part loads.

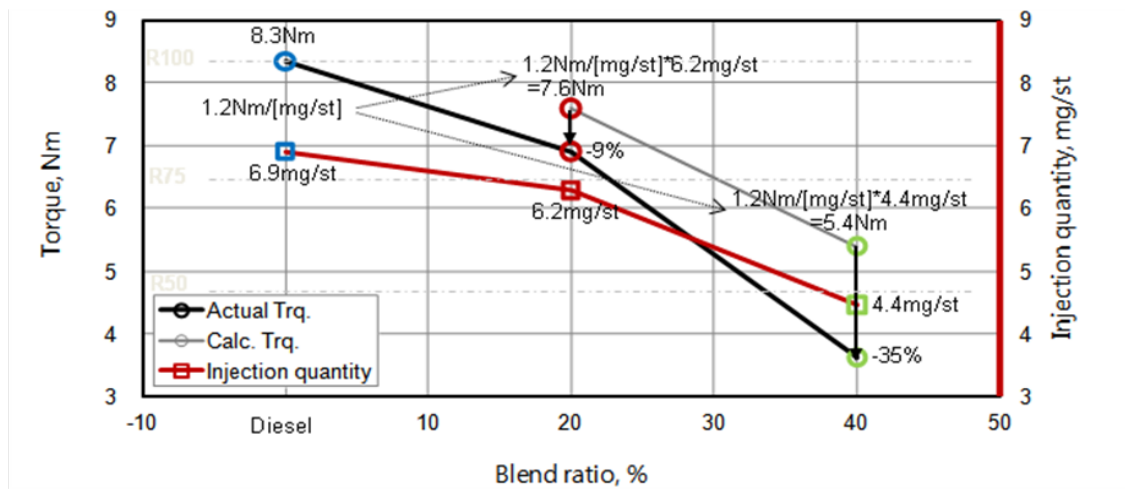


Figure 4.6 Torque and injection quantity as a function of the blend ratio at R100, 3500rpm

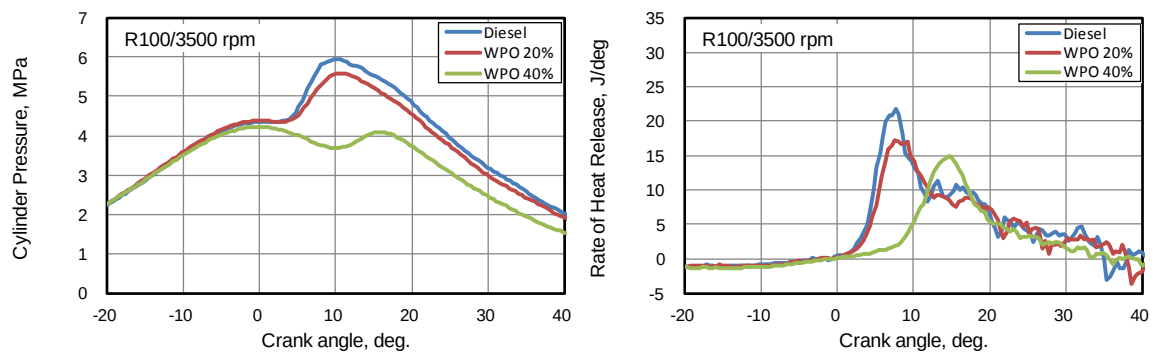


Figure 4.7 Cylinder pressure and rate of heat release at I100, 2450rpm

4.2.2 Exhaust emissions

4.2.2.1 Ignition delay

In Figure 4.8, the ignition delay time and the fraction of the premixed combustion phase was summarized based on the combustion data which are depicted in Figure 4.9. At 2450rpm, for the 20% blend ratio, even with the longer ignition delay, the fraction of premixed combustion was slightly lower than that of diesel and this was caused by the lower injection rate as described previously. For the 40% blend, the ignition timing delayed more and the premixed combustion fraction increased more than the 20% blend [4-5]. The similar tendency was observed either at 3500rpm. The ignition delay became slightly shorten at 3500rpm as discussed in the Chapter 3.

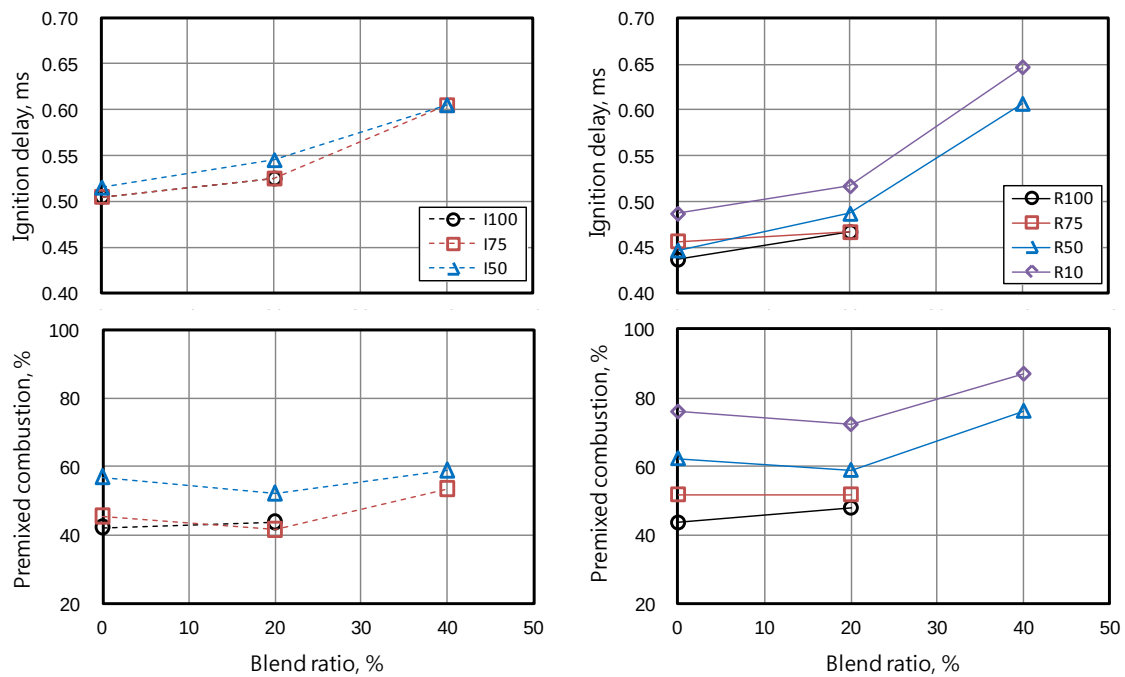


Figure 4.8 Ignition delay and fraction of premixed combustion phase as a function of the blend ratio (0: 100% diesel)

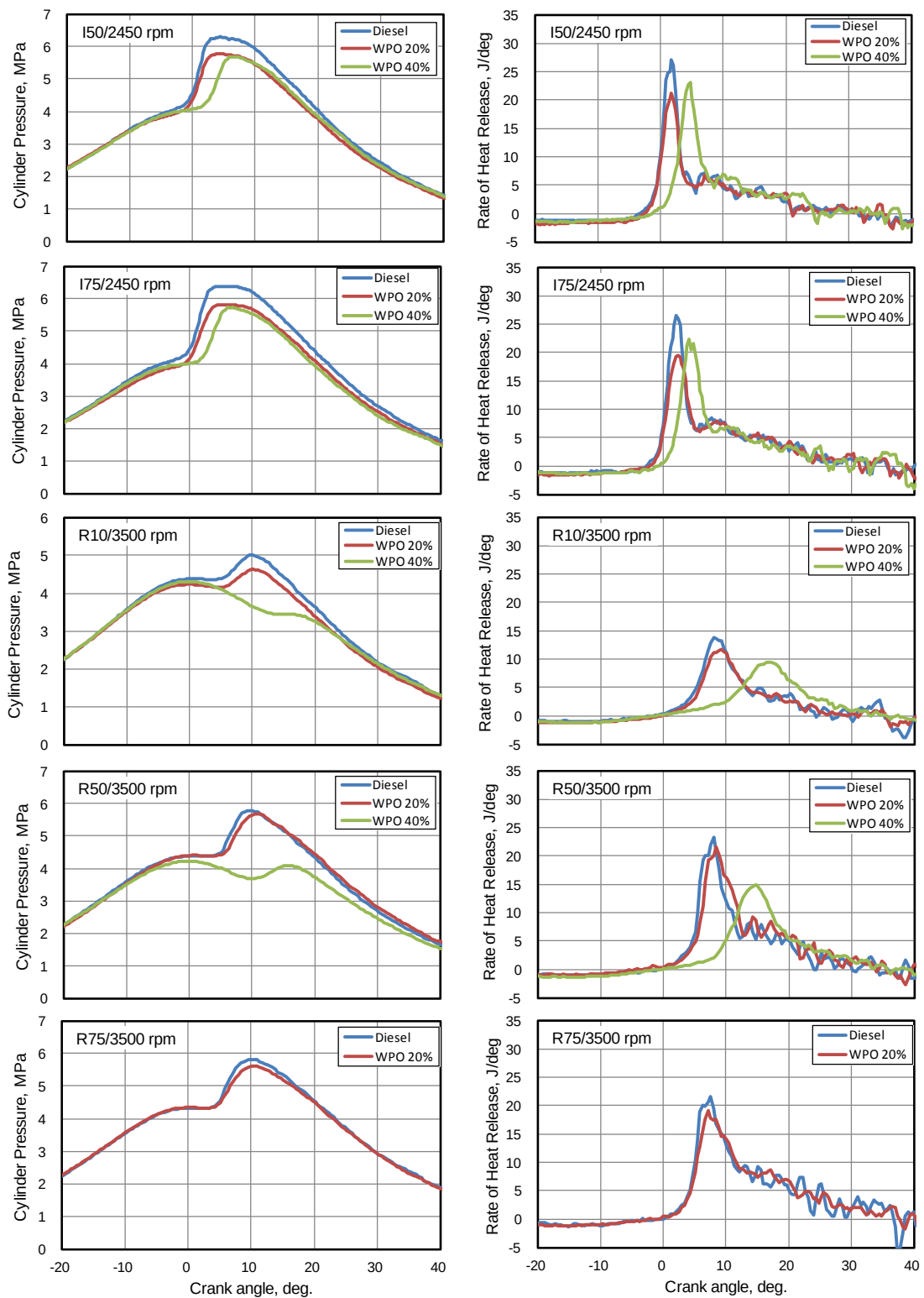


Figure 4.9 Cylinder pressure and rate of heat release

4.2.2.2 NOx emission

Figure 4.10 shows the result of the NOx emissions at 2450rpm and 3500rpm in the form of the concentration [ppm], the mass flow rate [g/h] and the specific emission [g/kWh]. For the 20% blend ratio, the NOx emission was not changed up to the 50% engine load and tended to slightly decrease from the 75% engine load and it might be contributed by the milder heat release rate due to the lower injection rate. For the 40% blend, the ignition timing delayed more and the premixed combustion fraction increased more than the 20% blend as shown in Figure 4.8, but the combustion occurred in more expanded space according to the downward of the piston and this resulted in the less NOx emission summarized on Figure 4.11. At the R10 point of the 40% blend, the NOx emission decreased around double due to the much longer ignition delay as shown in Figure 4.9.

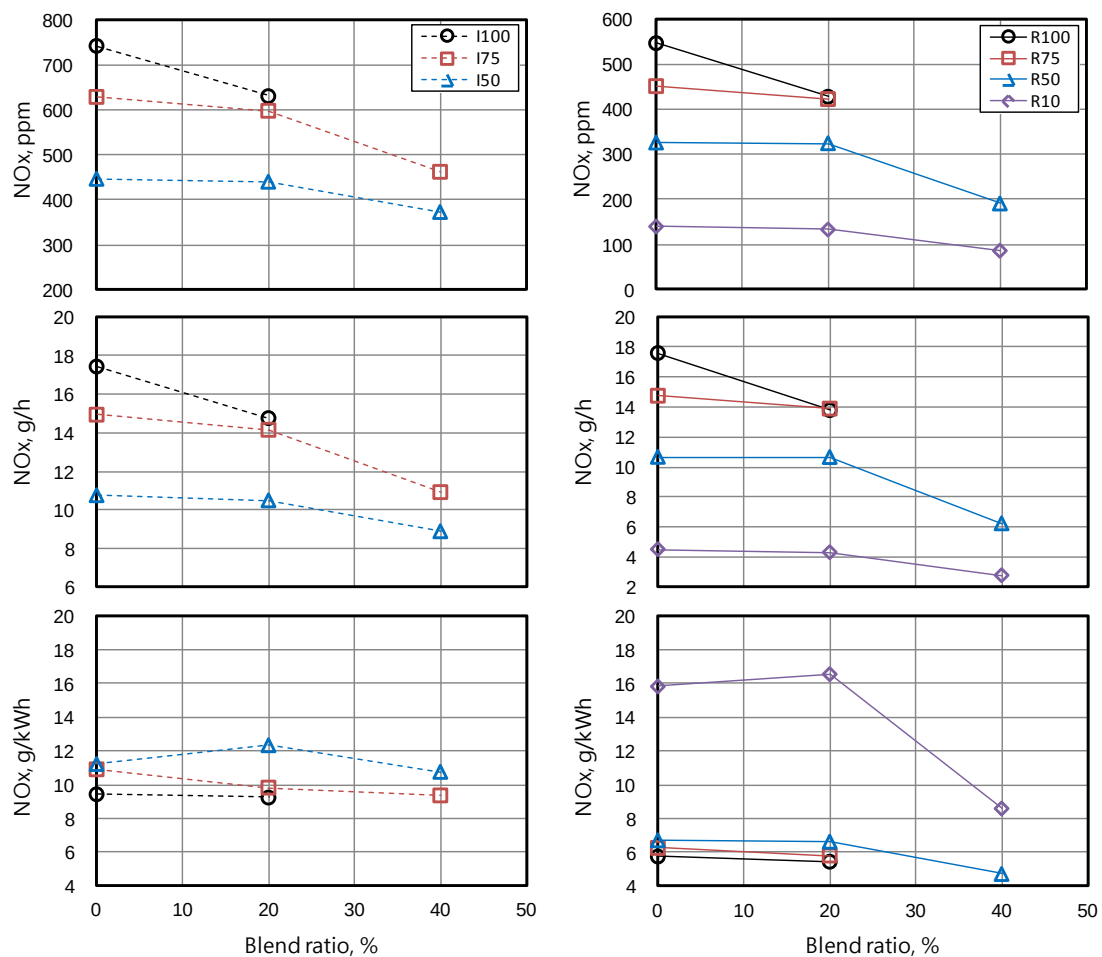


Figure 4.10 NOx emission in the form of the concentration [ppm], the mass flow rate [g/h] and the specific emission [g/kWh]: left at 2450rpm, right at 3500rpm (0: 100% diesel)

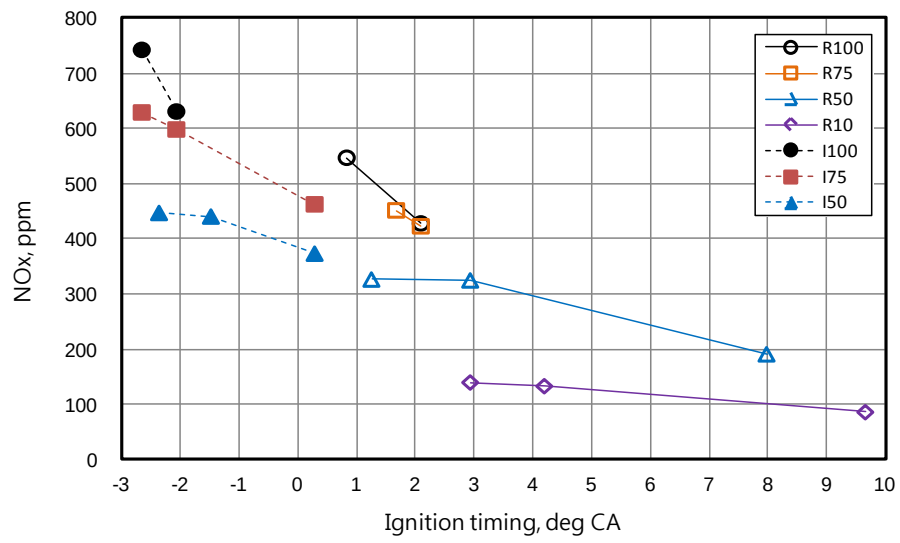


Figure 4.11 NOx emission as a function of the ignition timing

4.2.2.3 CO emission

Figure 4.12 depicts CO emission as function of the blend ratio. The CO emission decreased for the 20% blend and increased again for the 40% blend. This tendency almost tracked with the fraction of the premixed combustion except the I100. The more the premixed combustion was, the higher the CO emission was, which is summarized in Figure 4.13. The tendency was also observed in the Chapter 3 but the correlation looks stronger and it might be that the raw pyrolysis oil is originated from crude oil like the diesel fuel and there is no other factor to reduce the combustion temperature not like the bio-oil. At 3500rpm, for the 40% blend, the ignition delay time was not different from those at 2450rpm but the ignition timing was much more retarded in the crank angle according to the increase of the engine speed and this made the engine operation to be unstable and the CO emission to be increased.

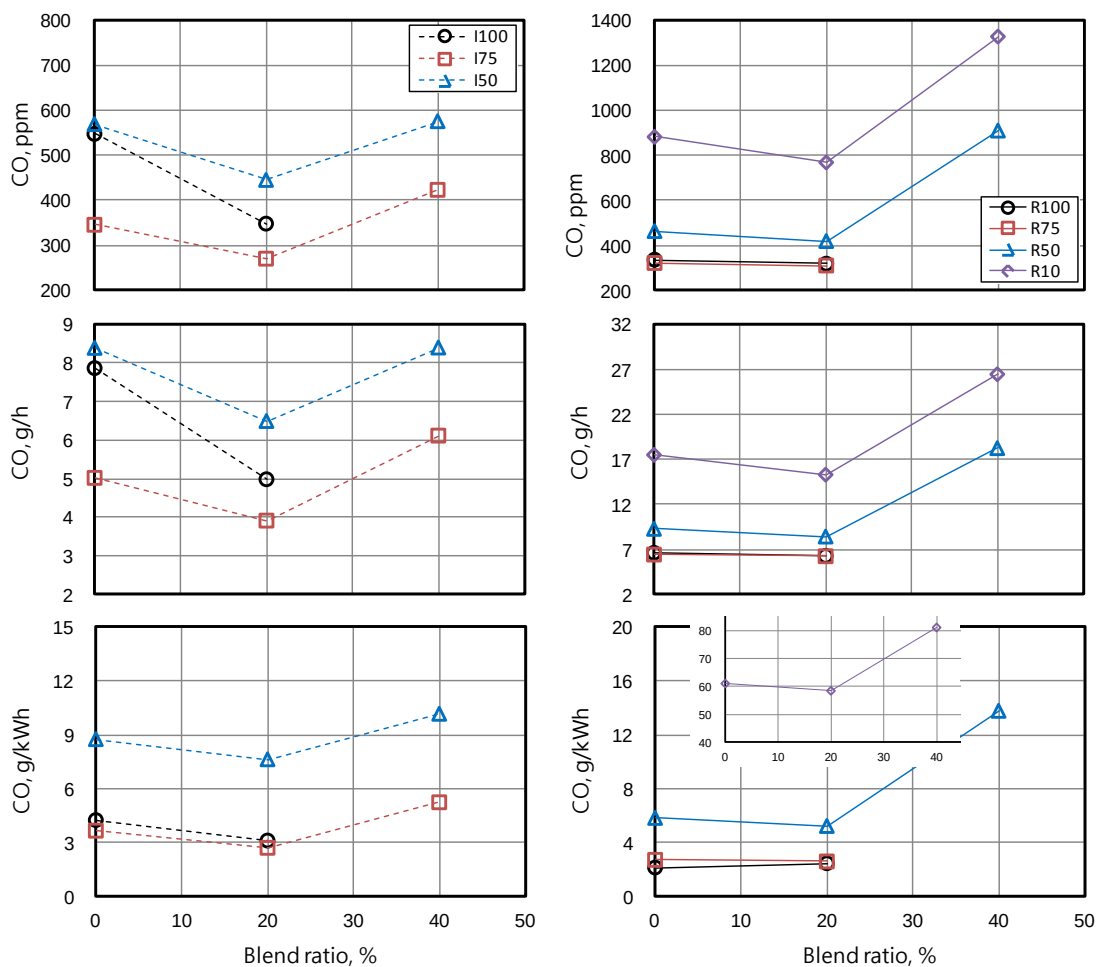


Figure 4.12 CO emission in the form of the concentration [ppm], the mass flow rate [g/h] and the specific emission [g/kWh]: left at 2450rpm, right at 3500rpm (0: 100% diesel)

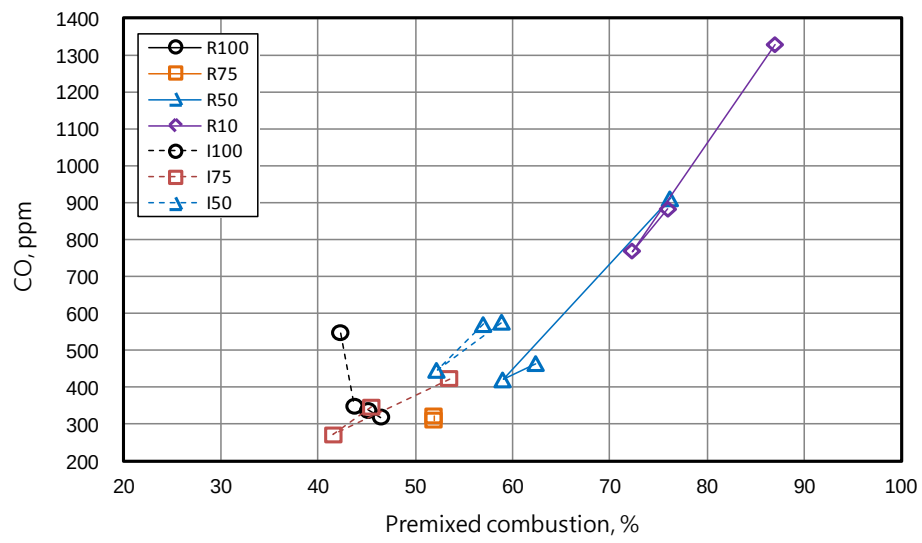


Figure 4.13 CO emission as a function of the fraction of the premixed combustion fraction

4.2.2.4 Smoke emission

The smoke emission was compared just at the R100 as shown in Figure 4.14. The smoke opacity of the 20% blend became slightly higher (about 0.5%). But the number might be negligible if it is taking into account of the precision of the opacity meter. In addition to that, the lambda of the 20% blend was higher due to the less injection amount of the blend oil than that of the diesel fuel and it was shown in the Chapter 3 that the higher the lambda is, the less the smoke is. The specific PM emission for both oils was far less than the EPA regulation, 1g/kWh.

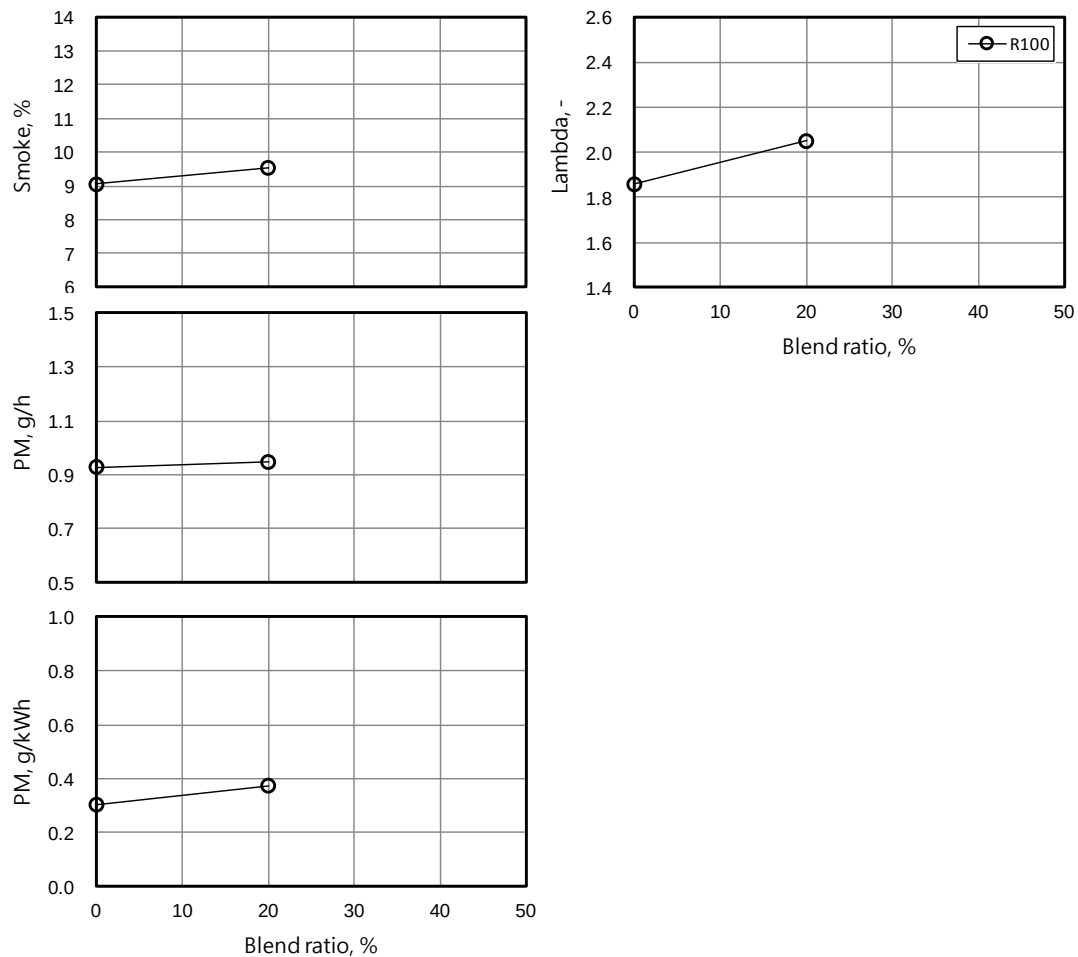


Figure 4.14 Smoke emission in the form of the opacity [%], the mass flow rate [g/h] and the specific emission [g/kWh] at R100 and Lambda (0: 100% diesel)

4.2.2.5 THC emission

Figure 4.15 illustrates the THC emission. The tendency was almost likely to that of the CO emission. The THC emission decreased in both engine speeds for the 20% blend [4-5]. For the 40% blend, the THC emission increased again and the level was less or same at 2450rpm but became higher at 3500rpm than those of the diesel due to the same reason which was describe for the CO emission. It might be the positive effectiveness of the lower viscosity on the condition of the spray atomization [4-6][4-7].

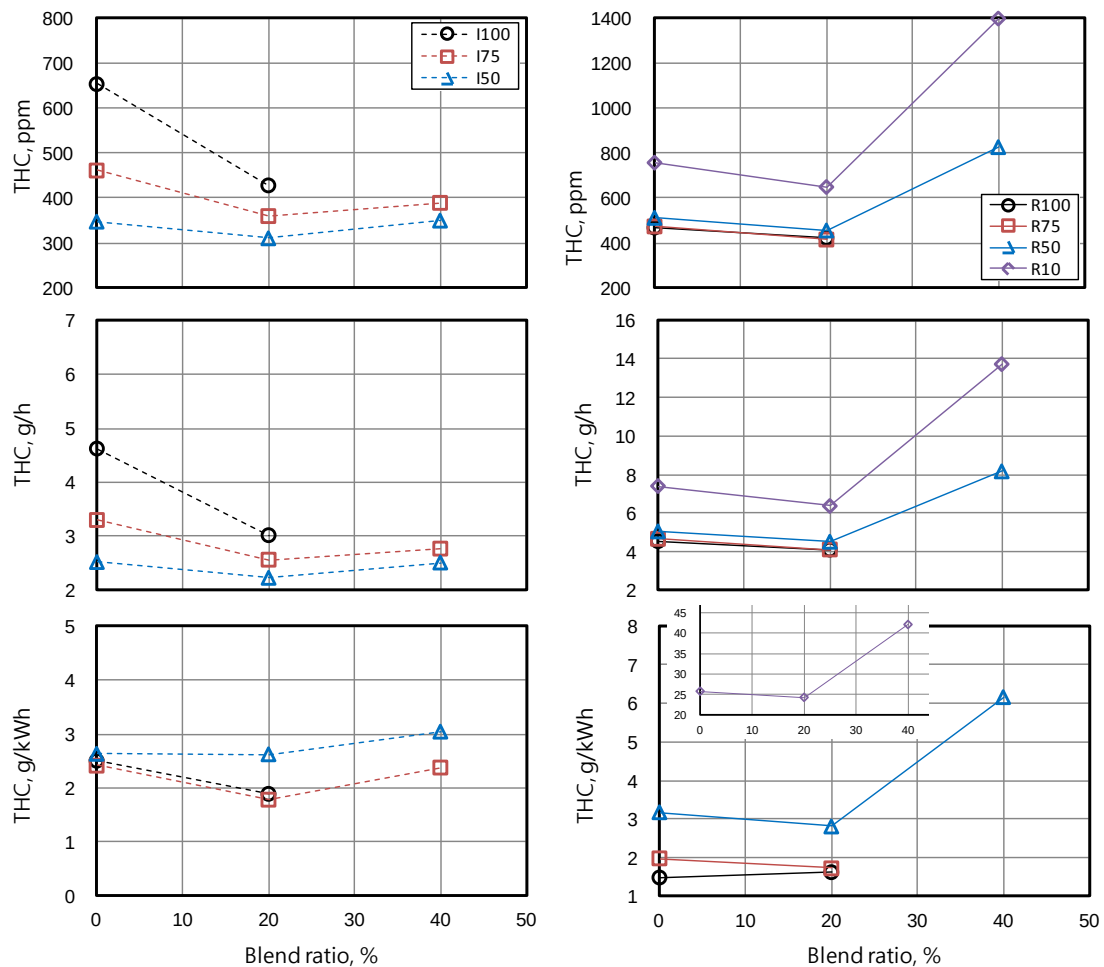


Figure 4.15 THC emission in the form of the concentration [ppm], the mass flow rate [g/h] and the specific emission [g/kWh]: left at 2450rpm, right at 3500rpm (0: 100% diesel)

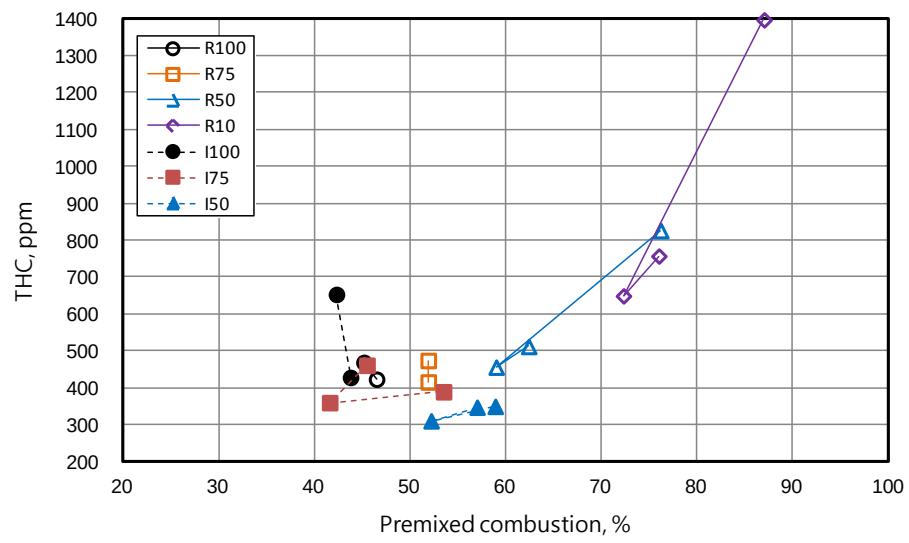


Figure 4.16 THC emission as a function of the fraction of the premixed combustion fraction

4.2.2.6 Exhaust emissions of EPA mode

Figure 4.17 shows the exhaust emissions with regard to C1-8 mode and D2-5 mode. The 40% blend oil was eliminated from this evaluation because the engine operation at the designated test points was not possible in I100, R75 and R100. Additionally, the operation in R10 and R50 was unstable due to the misfire which resulted in pretty high CO and THC emissions. The NO+THC was lower for the 20% blend than that for the diesel because of the milder heat release led by the physical property of the pyrolysis oil. In both EPA modes, the 20% blend marked marginally below the EPA limitation. The CO emissions in C1-8 and D2-5 were lower level for the 20% blend oil than those for the diesel fuel. The absolute number of the blend oil was less enough than the EPA regulation.

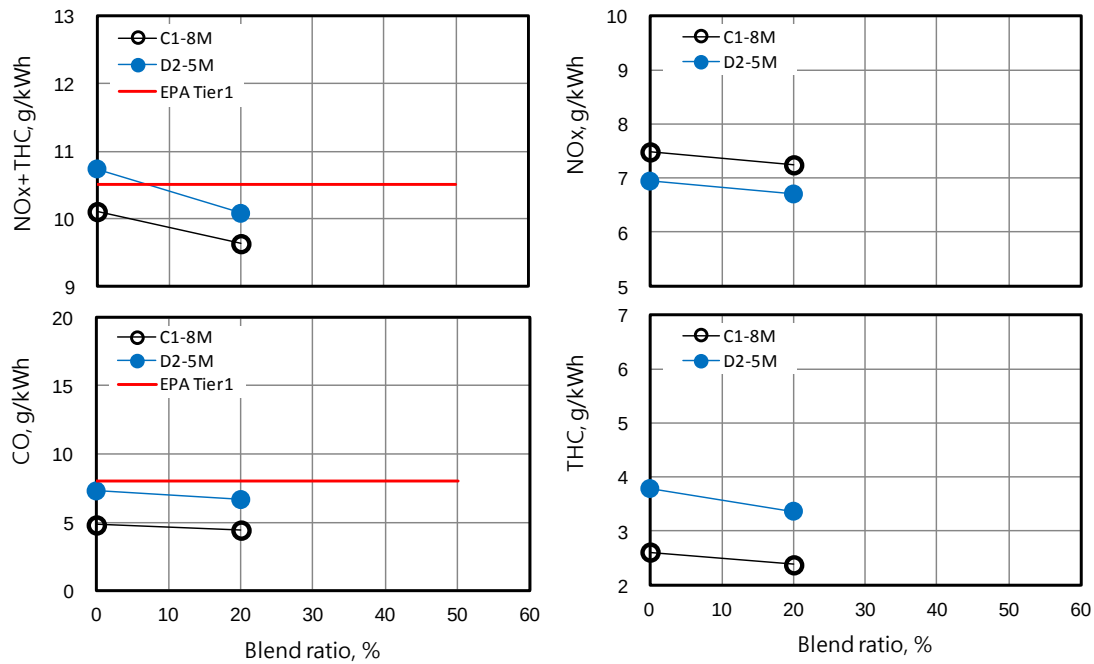


Figure 4.17 NOx+THC and CO mode emission for the diesel fuel (0%) and the blend oil

4.2.2.7 Toxic hydrocarbon ratio

The toxic hydrocarbon ratio was expressed as the ratio of the three toxic hydrocarbons content in the total hydrocarbon emission and is compared in Figure 4.18. The toxic hydrocarbon ratio was slightly lower for the 20% blend than for the diesel fuel in all engine operation points.

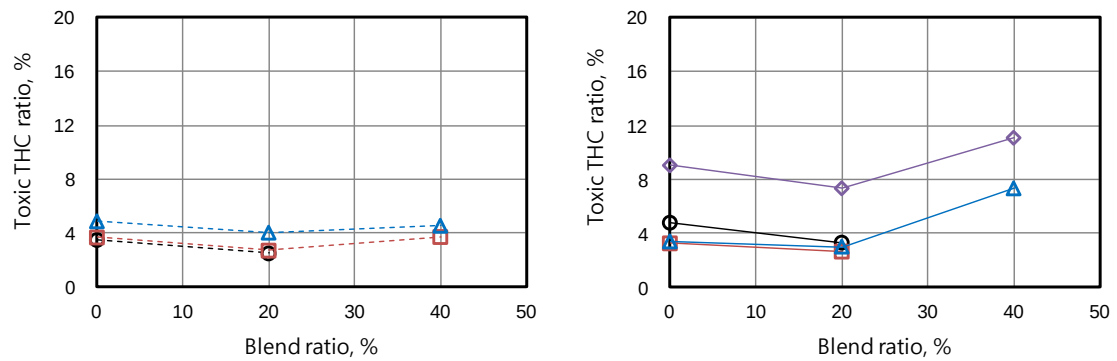


Figure 4.18 Comparison of the toxic hydrocarbon ratio (0% : diesel)

4.2.3 Specific fuel consumption

The specific fuel consumption (SFC) and the brake thermal efficiency at full load operation point are plotted in Figure 4.19. The thermal efficiency decreased with the increase of the blend ratio because of the less injection rate due to the physical property of the 20% blend and the retarded ignition timing caused by the chemical property of the 40% blend. The magnitude of the deterioration was slightly higher at 3500rpm than at 2450rpm due to the increased piston speed. For the 40% blend, at 3500rpm, the engine operation was too unstable to keep the operation for prolonged time but at 2450rpm, the engine could run normally. Thus, in order to increase the useful amount of the pyrolysis oil, the 40% blend at 2450rpm as shown in Figure 4.20 can be one choice for the electricity generation along with the de-rated engine power by one third and the lower thermal efficiency and the acceptable exhaust emission.

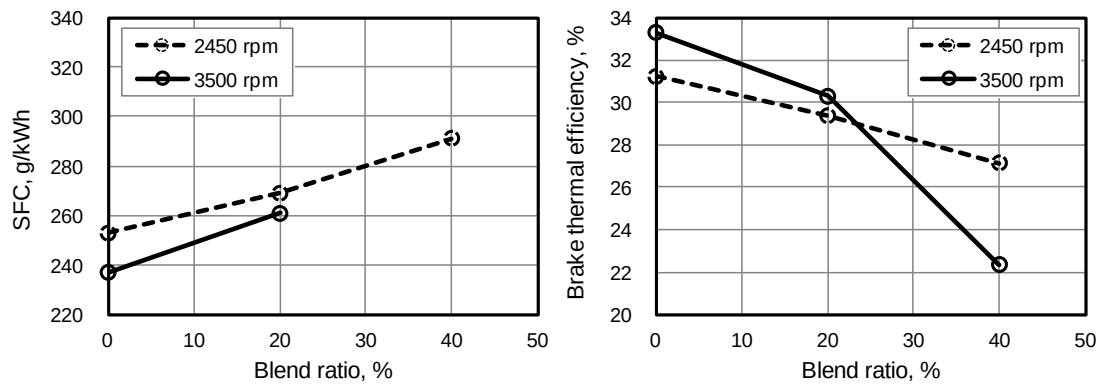


Figure 4.19 SFC and brake thermal efficiency as a function of the blend ratio (0%: diesel)

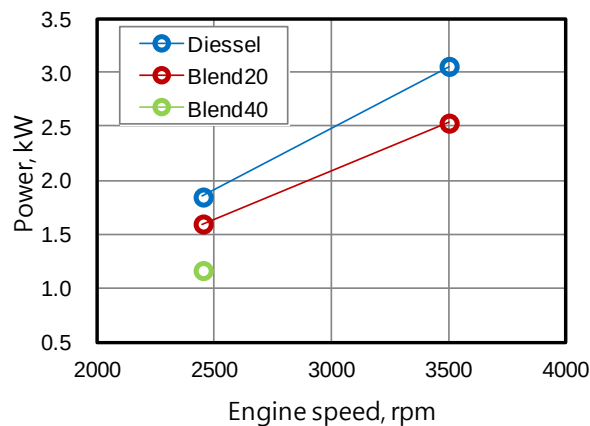


Figure 4.20 Potential engine operation points for the electricity generation

4.3 Conclusion

The pyrolysis oil was produced from real household waste plastics in a commercial pyrolysis plant. The raw pyrolysis oil was blended with diesel fuel in 20% and 40% volumetric ratio because of its low kinematic viscosity and poor self-ignition quality. A small single cylinder air cooled direct injection diesel engine was used for the combustion experiment to compare with the operation by diesel fuel.

With 20% blend ratio, the compatible engine operation was observed with regard to the full load engine output, the exhaust emissions and the thermal efficiency.

The maximum engine power dropped around 13% at the intermediate engine speed and 17% at the rated engine speed.

The exhaust emissions during the operation under the US EPA standard test modes, the C1-8 mode for non-road vehicles and the D2-5 mode for generation sets, resulted in :

The NO_x+THC and CO emissions were lower than those of the diesel fuel and marked marginally below both the EPA standards.

The PM was evaluated just at the rated power point and the number was much less than the regulation.

The toxic hydrocarbon which was defined as the content of toluene, formaldehyde and acetaldehyde in the THC emission was as low as diesel.

The thermal efficiency at full load was lower by maximum 3% in absolute. The combustion was mainly characterized by the oil physical properties such as the density and the kinematic viscosity and the impact of the chemical property was minor.

For 40% blend ratio, the maximum engine power dropped 37% at the intermediate engine speed and 57% at the rated engine speed where the engine operation was possible to be kept just for a few minutes. If the de-rated power by around one third is acceptable, the engine operation at 2450rpm can be one choice for the sake of decreasing the blend fraction of the diesel fuel. The chemical property played major role in the combustion characterized with the long ignition delay.

Thus, it is found that the present pyrolysis oil can be used in the diesel engine. The blend ratio with the diesel fuel is proper approximately 20% in volume. Additionally, around 40% blend ratio is possible at a limited engine speed such as below 2450rpm.

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Chapter 5

Engine long-run performance tests

In the Chapters 3 and 4, the appropriate blend ratio was defined taking into account of the full load performance, the exhaust emissions, the thermal efficiency and the engine operation stability. In this chapter, the acceptable blend ratio will be verified on the reliable operation through a long operation test. The engine operational test carried out during 8 hours continuously at the rated engine output point. In actual, the 8 hours is not sufficient duration for the assessment about the life time endurance of the pyrolysis oils but it can be a one indication for the endurance assessment. The 30% blend oil for the waste wood pyrolysis oil and the 20% blend oil for the waste plastic pyrolysis oil were chosen for this test. The test was done on the engine which was used for the Chapters 3 and 4.

5.1 Test procedure

Three types of oils were used: diesel, the 20% % blend oil of the waste plastic pyrolysis oil with the diesel which was decided in the Chapter 4 and the 30% blend of the bio-oil with the biodiesel which was obtained in the Chapter 3.

Before conducting the engine run with the pyrolysis oil, the engine operation fueled by the diesel was carried out in order to confirm whether the condition of the engine is normal or not. All the related measurement data were recorded every 30 minutes.

Photos were taken for the bottom surface of the engine head, the upper surface of the engine piston, the nozzle tip of injector before starting and after finishing the experiment. After finishing the test, the deposit which was adhered to the engine head surface were carefully removed and weighed after one night cooling down. For the weighing the deposit, the engine head was polished before each engine test as shown in Figure 5.1.



Figure 5.1 The engine head (bottom view, after the cleaning) before the engine test

5.2 Results and discussion

5.2.1 The 30% blend oil of the waste biomass pyrolysis oil

Figure 5.2 shows the time history of the engine power and the injection amount (mass per one stroke, mg/st). The power output of the diesel slightly dropped just after half hour operation and gradually decreased to 2.75kW (0.25kW power down) since 6 hours and thereafter stabilized. Contrary to that of the diesel, for the case of the 30% blend, about 0.5kW went down soon after 30 minutes and was kept constant but decreased again after 7 hours. Both of the power history trend were related to the reduction of the injection amount as shown in the right one in Figure 5.3. In addition to that, for the 30% blend, the engine power at 8 hour dropped more to around 2.4 kW even though the injection amount did not change from that at 6 hour and this is caused by the lower P_{max} (the maximum cylinder pressure) of 6.2 MPa and the retarded P_{max} producing timing of 12.5 deg CA ATDC (After Top Dead Center).

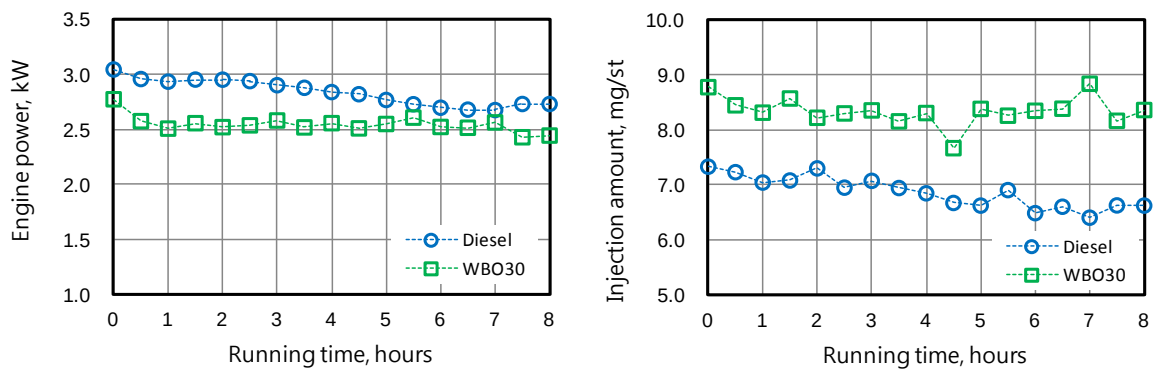


Figure 5.2 Time history of the engine power output (right) and the injection amount (left)

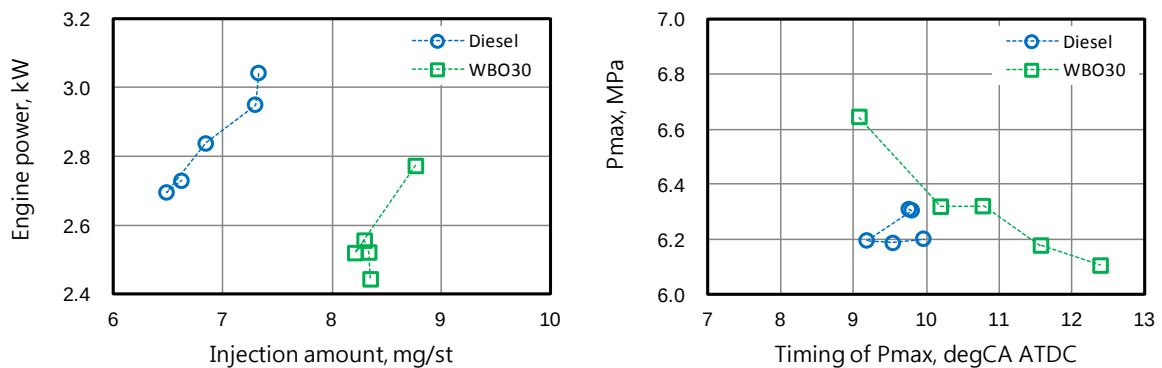


Figure 5.3 Engine power as a function of the injection amount (right) and P_{max} as a function of P_{max} producing timing (left)

For the 30% blend, since the first half hour operation, the combustion was not happened for a certain time in seconds. The cylinder pressure rise by the combustion was not happened and the cylinder pressure was the same to that of motoring because the engine was motored in force by the electric motor. In addition to that, the indication of the fuel metering was not changed. There might be some problem in the fuel delivery pump and hence failure to feed the fuel into the injector. But since several seconds, the injection of the fuel was recovered and the combustion occurred normally. The recovery might be introduced by the forced motoring by the electric motor. Due to the reversibility, the test was kept to go through the 8 hours. The failure mode intermittently and irregularly happened as shown in Figure 5.4. The duration of the failure and the interval randomly dispersed from 1 second to 40 seconds and 2 times to 5 times in 1 hour respectively. During the first 3 hours, the data was missed because it was late to be aware of recording the time stamping. There might be a relationship between the first 0.5 kW power-drop and the trouble. After finishing all the tests, the remainder of the 30% blend oil in the fuel delivery system was flushed with the diesel fuel via running by the electric motor but the injection was not happened.

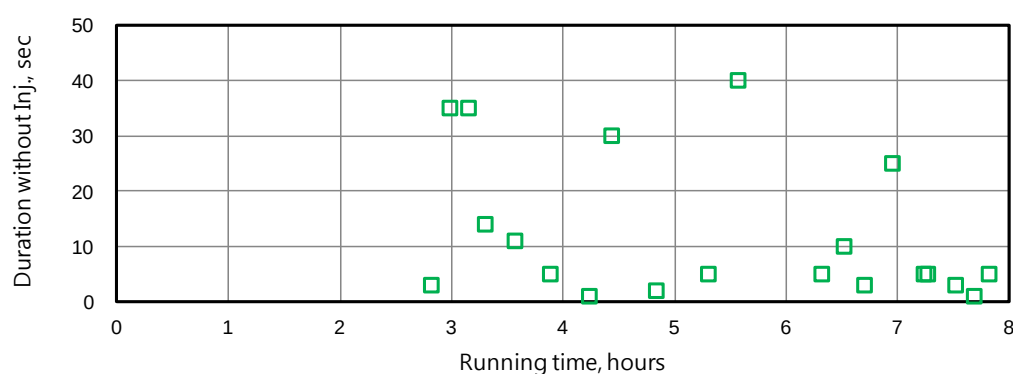


Figure 5.4 Time history of the duration with injection

Figure 5.5 shows the time history of the exhaust emissions. For the diesel, the NO_x emission slowly decreased by the time passing and it is caused by the lower injection rate which was reflected by the lower injection amount as shown in Figure 5.2. There had been very little change in the heat release rate between the first 2 hours and the last 2 hours as illustrated in Figure 5.5. All other emissions like THC, CO and smoke showed a little fluctuation during the test but did not show a significant difference from the initial to the end.

NO_x emission gradually decreased like that of the diesel but the cause looked different because the recorded injection amount was kept constant after one hour (even there was a fluctuation which might be caused by the fuel metering error due to the intermittent injection stop). The ignition timing as summarized in Figure 5.8 was retarded by the time passing and it was around 5 deg. CA difference from the start to the end of the test. Due to the reduced injection rate as shown in Figure 5.2, the rate of heat release became mild and as a result, this contributed the NO_x decrease. The THC, CO and smoke opacity rapidly increased around 5 hours later.

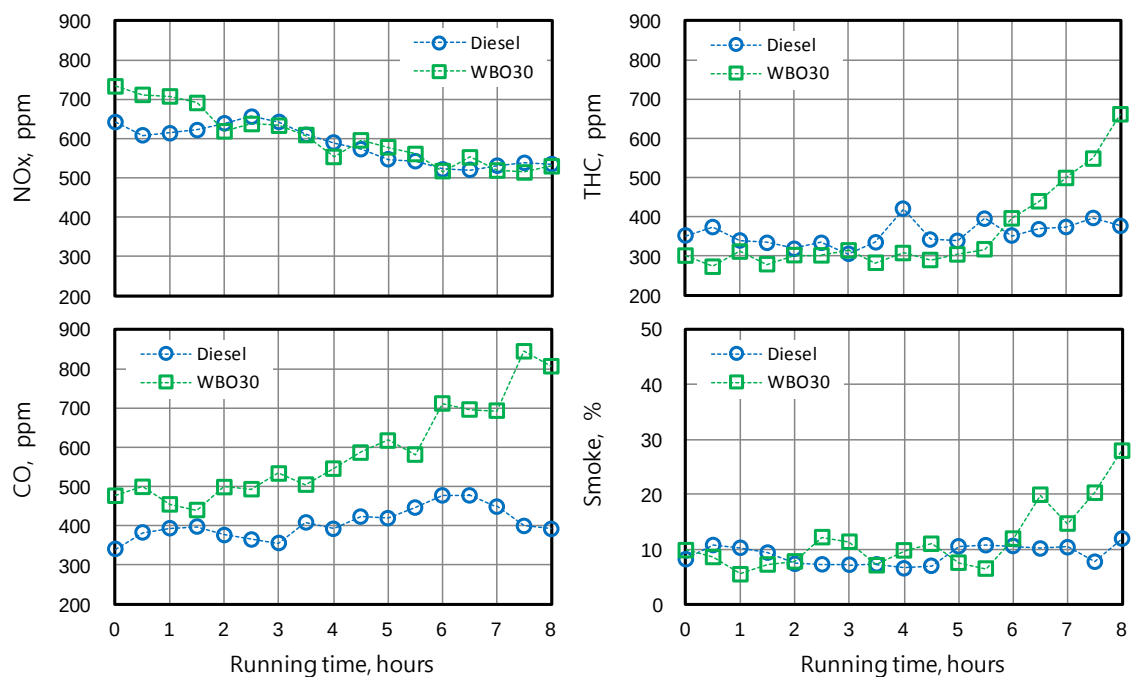


Figure 5.5 Time history of the exhaust emissions

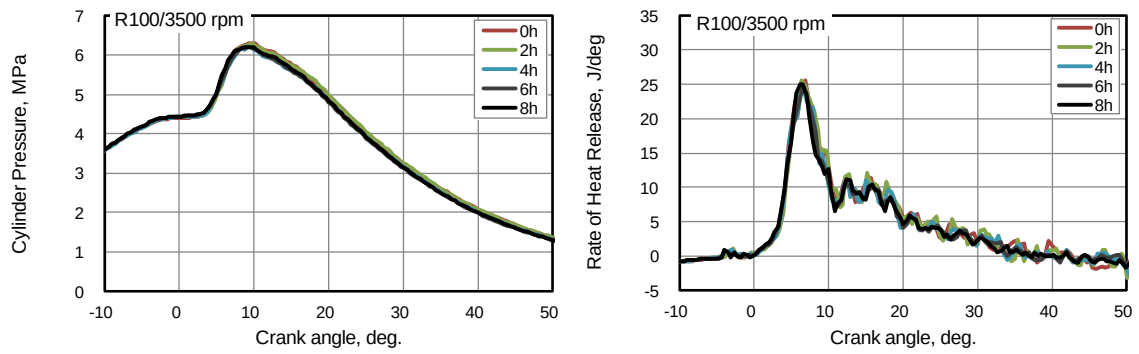


Figure 5.6 Time history of the cylinder pressure and the rate of heat release in case of the diesel operation

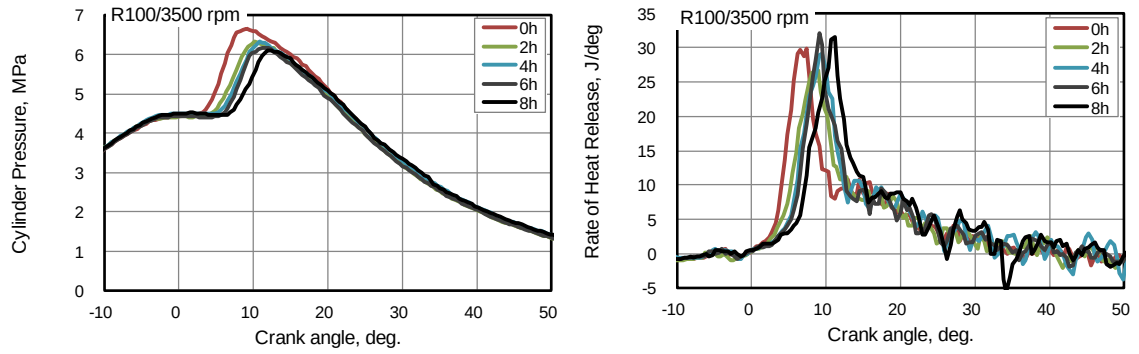


Figure 5.7 Time history of the cylinder pressure and the rate of heat release in case of the 30% blend oil

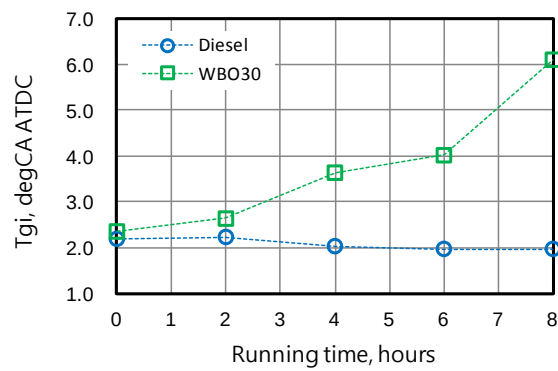


Figure 5.8 Time history of the ignition timing

The deposit on the bottom surface of the engine head was weighed after the tests and the mass is shown in Figure 5.9. The 30% blend resulted in around two times higher than that of the diesel. The number of the mass does not have absolute meaning itself but somehow qualitative comparison. Analysis with regard to the double deposit mass will be followed below based on the photos.

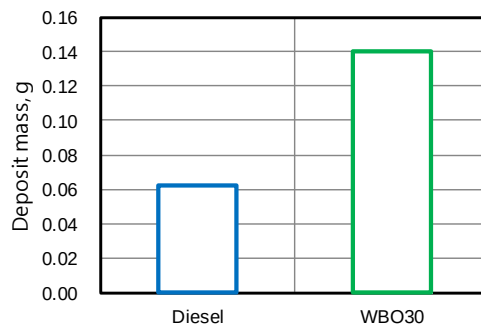


Figure 5.9 Deposit mass adhered to the bottom surface of the engine head after the test

Figure 5.10 shows the outside appearance of the nozzle tip after the engine tests for both fuels. For the 30% blend, the carbon deposit was built on the outlet of the nozzle hole whereas any deposits were not found for the case of the diesel [5-3][5-4]. Figure 5.11 shows the upper view of the engine piston after the tests. For the diesel, the spray trace is shown on the outer side of the piston surface. The number 1 spray looks totally separated from other three sprays which were close to each other. The three sprays (number 2, 3 and 4) seemed to be interfered on the outside of each spray but kept its spray pattern like the dot line.

For the 30% blend, even though the number 1 spray was individually segregated, other three sprays seemed to be severely interfered like a belt line as shown in the photo. The color was bright gray for all the sprays while the diesel was black colored. Inside the piston bowl, several shining particles were visible and these seemed to be inorganic solid matter. In Figure 5.12, the bottom view of the engine head for the 30% blend is depicted after the test. The pillar shape deposit was largely built on the surface starting from the outlet of the number 1 nozzle hole. Even though the magnitude differed, the deposit was found on the surface for other three sprays while there were no deposits for the case of diesel. From the analysis, the poor spray quality must be playing a large role to the worse exhaust emissions such as the smoke and the THC.



Figure 5.10 Nozzle tip after the tests: right (the diesel) and left (the 30% blend)

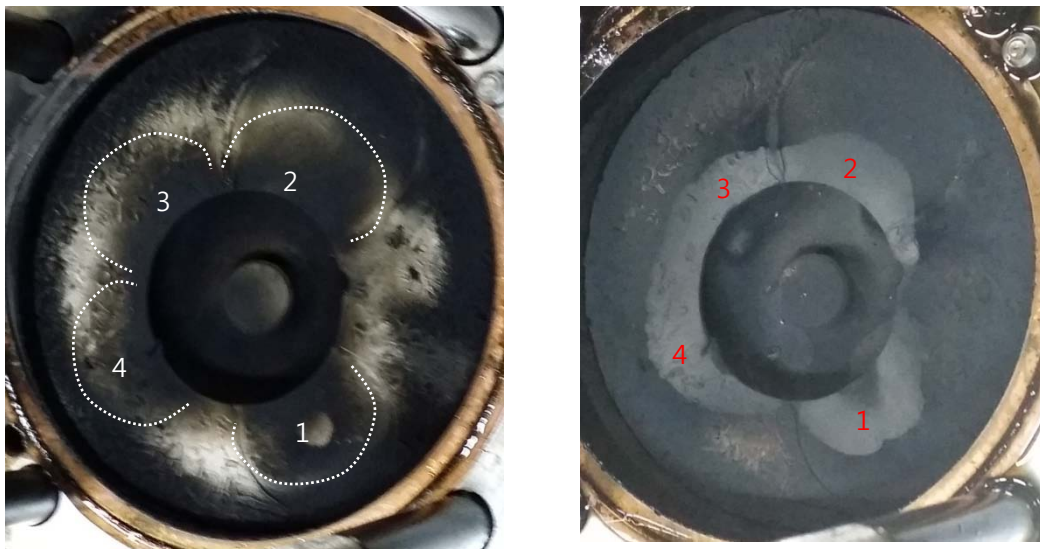


Figure 5.11 Upper view of the piston after the engine tests: right (the diesel) and left (the 30% blend)

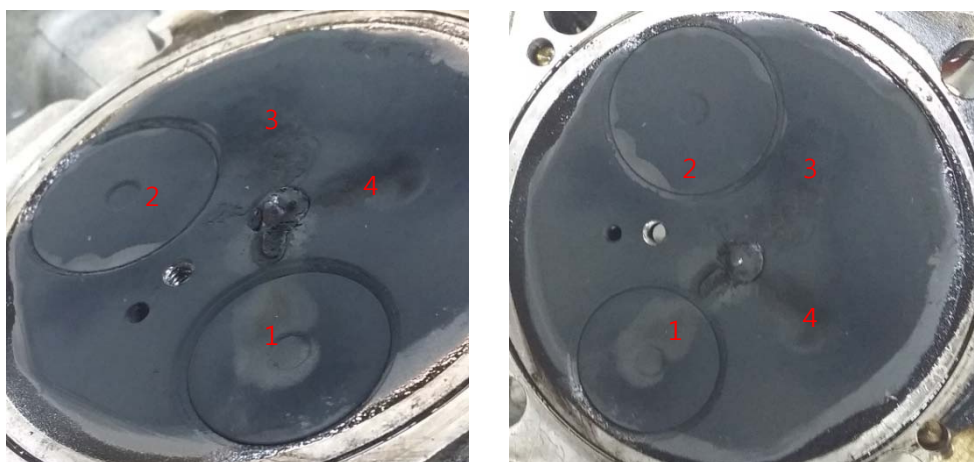


Figure 5.12 Bottom view of the engine head after the engine tests: the 30% blend

Figure 5.13 shows the appearance of the fuel vessel and the fuel tube line after the 8 hours engine running. The tar was stick to the bottom surface inside the vessel and the heavy tar like the thin black line inside the tube was found. Mueller reviewed about the stability of biomass pyrolysis oil as the two primary modes of instability of the pyrolysis oil are hydrophilic/hydrophobic phase separation of the pyrolysis oil micro-emulsion and liquid-phase chemical reactions. Pyrolysis oil is not created in thermodynamic equilibrium and as a result reactions during storage tend to move the composition toward to thermodynamic equilibrium [5-1]. Even with the aid of the surfactant like methanol, the unstable compositions in chemical reaction still remained in the blend oil and the phase separation proceeded and polymerized. As a result, the tar turned out as reported in the Chapter 3. The instability of the blend oil might affect the injection timing delay with the time passage which is described in Figure 5.8. Such reaction might lead to increase in the viscosity [5-2], and as a result, as shown Figure 5.2, the injection amount of the 30% blend trended to be constant in overall with time while that of the diesel slowly decreased with the time passage.



Figure 5.13 Fuel vessel and fuel tube line after the engine tests: the 30% blend

5.2.2 The 20% blend oil of the waste plastic pyrolysis oil

Due to the irreversible fuel delivery failure, the delivery pump and the injector were replaced with the new ones and additionally, several mechanical moving parts also exchanged with the new ones.

Because of the restriction of the available time to the engine test, the long-run test was not conducted for the diesel fuel and carried out only for the 20% blend oil of the plastic waste pyrolysis oil. Figure 5.14 shows the time history of the engine power and the injection amount (mass per one stroke, mg/st). The data which was evaluated in the previous chapter is plotted just as reference. The injection amount of the 20% blend was higher than that in the Chapter 4 but the power was lower. It might be resulted from the replacement of the parts; the new pump contributed to the former and the new moving parts affected on the latter. The engine run fueled by the 20% blend oil had been finalized through the 8 hours without any troubles.

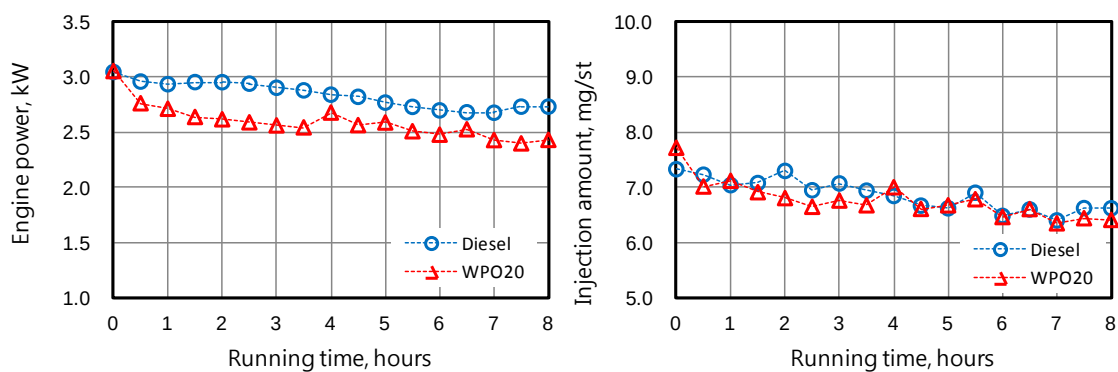


Figure 5.14 Time history of the engine power output (right) and the injection amount (left)
(The data of the diesel was taken over from the previous chapter)

The time history of the exhaust emission is illustrated in Figure 5.15. The cylinder pressure and the rate of the heat release are depicted in 2 hours interval in Figure 5.16.

The NO_x emission slightly decreased as time passed due to the milder heat release which is expressed as the reduction of the premixed combustion fraction and the lower peak of the heat release rate which is resulted from the injection rate decrease as shown in Figure 5.14. All other emissions did not show a certain tendency with the time passage.

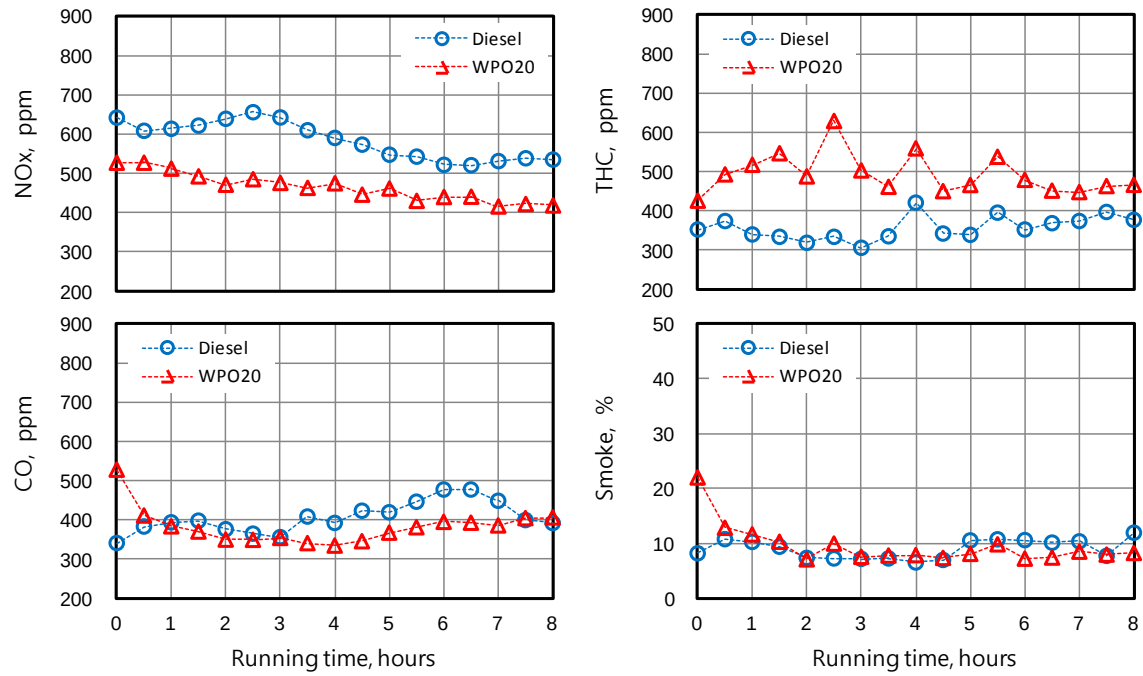


Figure 5.15 Time history of the exhaust emissions
(The data of the diesel was taken over from the previous chapter)

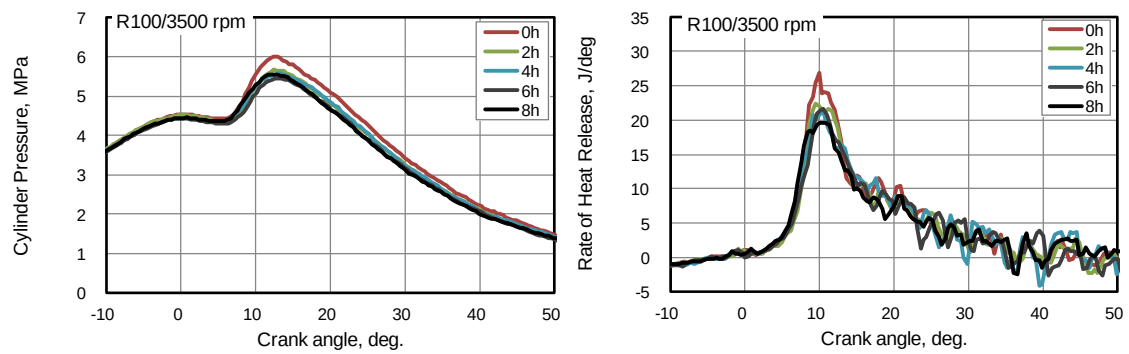


Figure 5.16 Time history of the cylinder pressure and the rate of the heat release in the case of the 20% blend

In Figure 5.17, the appearance photos of each part after the 8 hours test are listed. The deposit which was found in the long operation fueled with the 30% blend of the bio-oil was not observed. It looks almost same with those of the test with diesel. This corresponds to the result of the exhaust emission which was evaluated already.



Figure 5.17 Nozzle tip, bottom view of the engine head and upper view of the piston after the engine test of the 20% blend oil

5.3 Conclusion

The defined blend ratio in the Chapters 3 and 4 were evaluated for the potential of long operation through 8 hours continuous operation in the engine. The evaluation was focused on the potential of continuous operation at the rated engine output point and the exhaust emission trace with the time passage and it concluded as below.

For the 30% blend oil of the bio-oil with the bio diesel:

The fuel injection system was encountered with the irreversible failure after finishing the 8 hours operation and it is confirmed by the impossibility of the fuel injection with the diesel fuel. The fuel injection stop was occurred after the first 30 minutes operation and it intermittently observed through the whole test duration.

The injection timing was gradually delayed with the time passage and it might be brought by the chemical instability of the biomass pyrolysis oil. This might affect the exhaust emissions.

The NO_x emission decreased little by little through the whole time but all other exhaust emissions (CO, THC, smoke) sharply rose nearly 5 hours later.

Abnormal deposit inside the engine such as the surface of the nozzle tip, the engine head and the piston was observed.

For the 20% blend oil of the waste plastic pyrolysis oil with the diesel:

The engine operation was successfully finished during the 8 hours.

The NO_x emission gradually declined due to the injection rate decrement which was almost identical tendency with that of the diesel. All the other exhaust emissions such as CO, THC and smoke were observed in the unchanged level through the duration.

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Chapter 6

Summary and Conclusions

6.1 Summary

The goal of this work was to give practical information in adaption of the pyrolysis oils (bio-oil, waste plastic oil) into a diesel engine for mainly the generation of electricity. It was pursued by that all the pyrolysis oils which was fueled to the diesel engine was come from the real world plants and the oils was upgraded through very simple process of blending with commercial diesel and biodiesel fuel.

For the bio-oil, because of the first small volume of the oil delivery from the plant (pilot production) the combustion test was conducted in the RCM fueled by the blend oil of the bio-oil with the diesel fuel and then the bio-oil was blend with biodiesel in order to mainly prevent from the phase segregation. The waste plastic oil was just blended with the diesel fuels without a special pretreatment.

The most appropriate blend ratio was defined through the evaluation of the result from the operation test in a diesel engine. The engine operation points were selected based on the EPA regulation standard. The evaluation was carried out for the full load power output, the exhaust emissions (the EPA standard) and the specific fuel consumption.

In order to adapt the blend oils as real world utilization, the engine operation fueled with the blend oils must be sustained for a long time. According to this, the defined blend ratio was given to the blend oils and the engine operation test was conducted for 8 hours continuously. The 8 hours was not sufficient for the purpose but might be useful to get a small indication.

Two different types of pyrolysis oil were used and evaluated independently in each chapter. Before summarizing the conclusions, the overall graphical view was summarized as shown in Figure 6.1, 6.2 and 6.3.

In Figure 6.2 physical property was compared. The density and the kinematic viscosity increased with the blend ratio for the bio-oil blend while they decreased for the plastic oil blend. The LHV of the raw plastic oil was almost identical to the diesel fuel and constant even with blending but for the bio-oil the LHV (Low Heat Value) declined with the blend

ratio elevation. The water content increased according to blending for the bio-oil in contrary to the plastic oil of the water absence.

Figure 6.2 shows on the injection amount, the engine power, the SFC (Specific Fuel Consumption) and the ignition delay at R100 (rated engine power point). The injection duration was kept constant for all the oil test cases at R100 by fixing the throttle position to that of diesel. The injection amount of the bio-oil increased and had linear correlation with the oil density and due to this the engine power at R100 was kept identical to that of diesel fuel even though the LHV decreased with blend ratio. The reduction of the injection in plastic oil directly resulted in severely engine power decline and additionally the longer ignition delay made negative impact on the engine power. The SFC had mainly dependency on the oil density for the bio-oil and the ignition delay for the plastic oil respectively. The ignition delay became longer with the increased blend ratio and it is because of the water content for the bio-oil and the chemical component for the plastic oil. For the plastic oil, the ignition delay became drastically longer over 20% blend and this resulted in the unstable combustion addition to the power drop as mentioned above.

When it comes to increase oil blend ratio, the bio-oil should be careful for the increase in NO_x+THC (especially THC), PM and CO, the plastic oil does not have concern for exhaust emissions which shows the reduction of NO+THC (mainly due to THC reduction) and the same level for CO and PM.

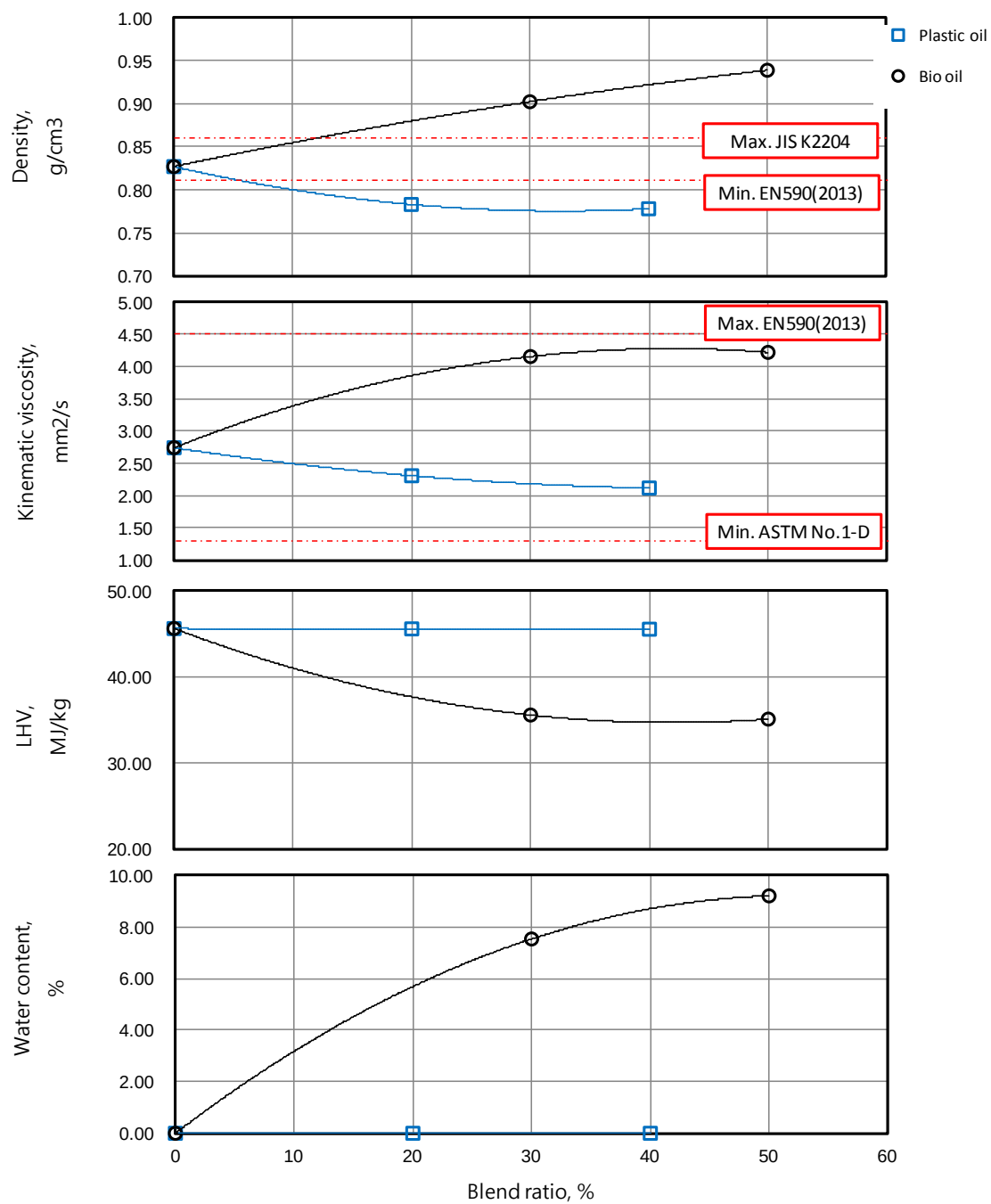


Figure 6.1 Density, Kinematic viscosity, low heat value and water content as a function of the blend ratio (0%: diesel)

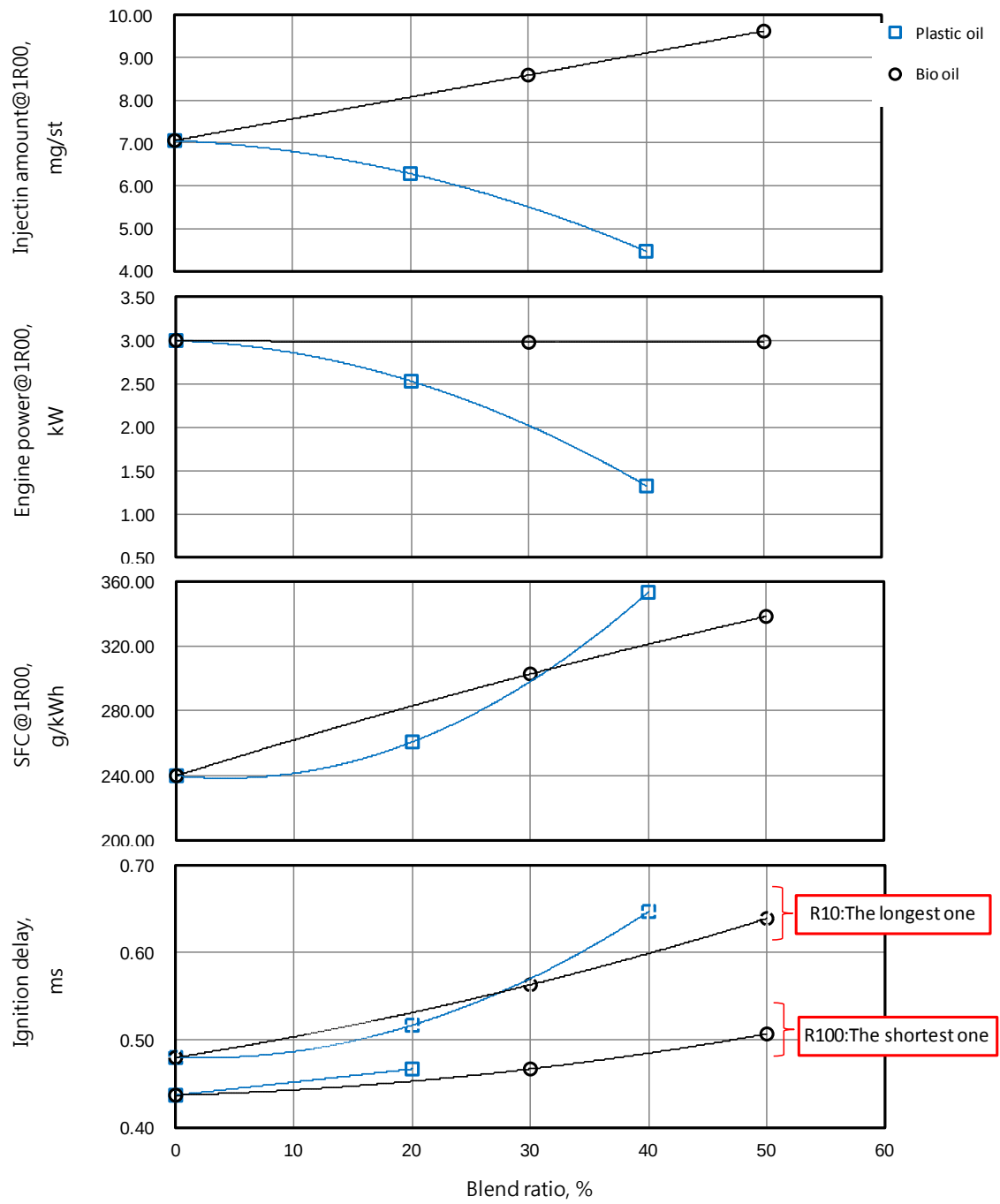


Figure 6.2 Injection amount/engine power/SFC at R100 and ignition delay (Solid: R100, the shortest delay and Dot: R10, the longest delay) as a function of the blend ratio (0%: diesel)

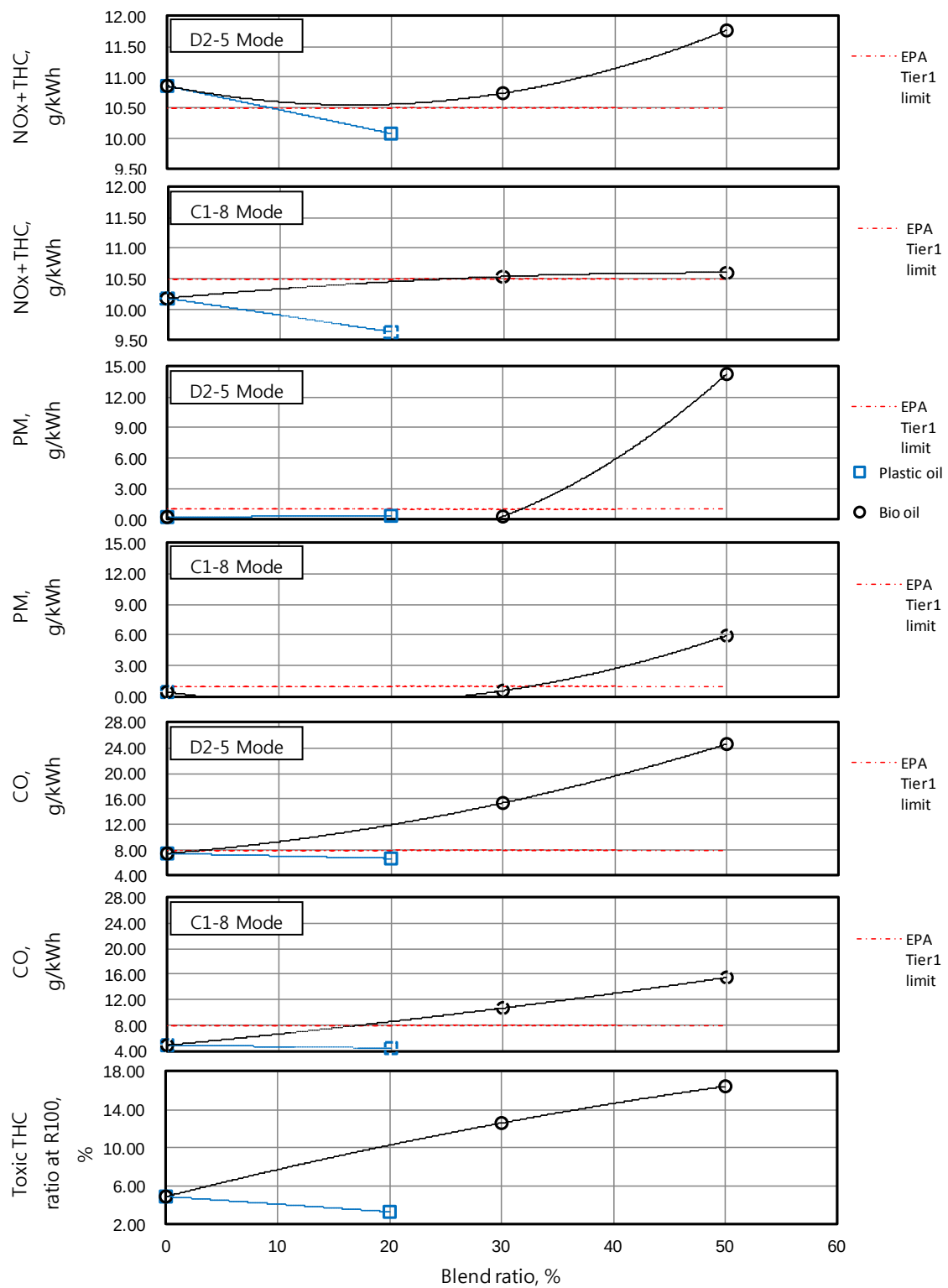


Figure 6.3 EPA mode exhaust emissions (NOx+THC, CO, PM) and toxic THC ratio at R100 as a function of the blend ratio (0%: diesel)

Why the smoke was increased for the engine test than for the RCM test ?.

Figure 6.3 shows the normalized comparison for NO_x and smoke emission between the blend of biodiesel/bio-oil (70%/30% in volume) in the engine test and the blend of diesel/bio-oil (75/30% in volume) in the RCM. In the engine test, the smoke for the diesel/bio-oil was higher than that of the diesel fuel while the smoke for the diesel/bio-oil blend was drastically decreased than that of diesel fuel.

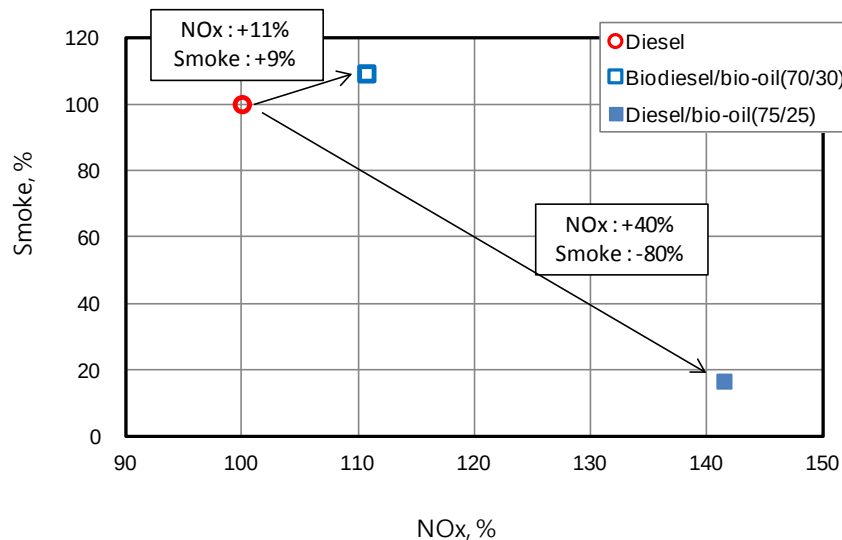


Figure 6.4 Normalized comparison for NO_x and smoke emission between the blend of biodiesel/bio-oil (70%/30% in volume) in the engine test and the blend of diesel/bio-oil (75/30% in volume) in the RCM test

1. Air excess ratio(λ)

There is strong correlation between the λ and the smoke as shown in Figure 3.20. The λ of the biodiesel/bio-oil (1.6) became worse around -10% due to the higher density than that of the diesel fuel (1.8). In case of the RCM, the λ of the diesel/bio-oil blend (2.0) was -3% lower than the diesel fuels (2.0).

2. Spray atomization difference depending on the injection pressure and the kinematic viscosity

Figure 6.4 shows the effect of the kinematic viscosity on Sauter mean drop diameter as a function of the injection pressure. The Sauter mean diameter (SMD) in 20MPa injection pressure (at the engine test) shows the higher difference rather than that in 100MPa injection pressure (128MPa at the RCM test).

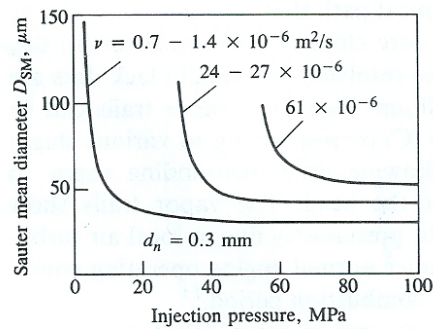


Figure 6.5 Effect of kinematic viscosity on Sauter mean drop diameter as a function of the injection pressure [4-7]

3. Liquid penetration of spray and its impingement onto the combustion chamber

The liquid penetration of the blend oils is greater than that of the diesel fuel due to its higher kinematic viscosity [5-4][5-18]. The longer liquid spray (No. 1 as shown in Figure 6.5) impinges the combustion chamber wall and split into two directions (up side and down side), the up side liquid spray reaches the bottom surface of the engine where the temperature is lower and rapidly cooled down and the combustion became worse. It can be evident from the picture which is shown in Figure 5.12 (carbon deposits on the engine head) in Chapter 5. On contrary to that, in the RCM, the liquid spray might not be contacted onto the combustion chamber because of the sufficient distance (100mm) where the spray can be typically developed.

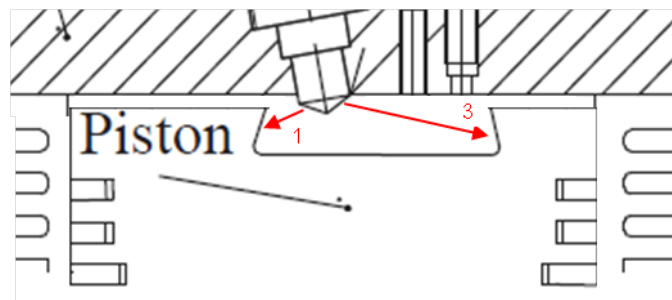


Figure 6.6 Schematic of the combustion chamber and injection nozzle (actual scale and the piston position at the injection timing)

Despite the positive effect of micro-explosion due to water and of oxygen presence in fuel on smoke emission (the very lower smoke emission in the RCM test), the smoke emission of the blend oil in the engine test became worse than that of diesel fuels.

Why the ignition delay of the bio-oil was shorter than that of diesel at the RCM test comparing with the result at the engine test ?.

At the RCM test the ignition delay of the bio-oil resulted in shorter than that of the diesel. But the result was contrary at the engine test. For this some analysis and inference had been introduced.

Figure 6.6 shows cylinder pressures from the RCM test. There was cylinder pressure deviation at the piston top position (at the time of 45ms) between the diesel and the blend oil (blend25) and during the test for the certain oil. It might be resulted from the experimental error in adjusting the test conditions.

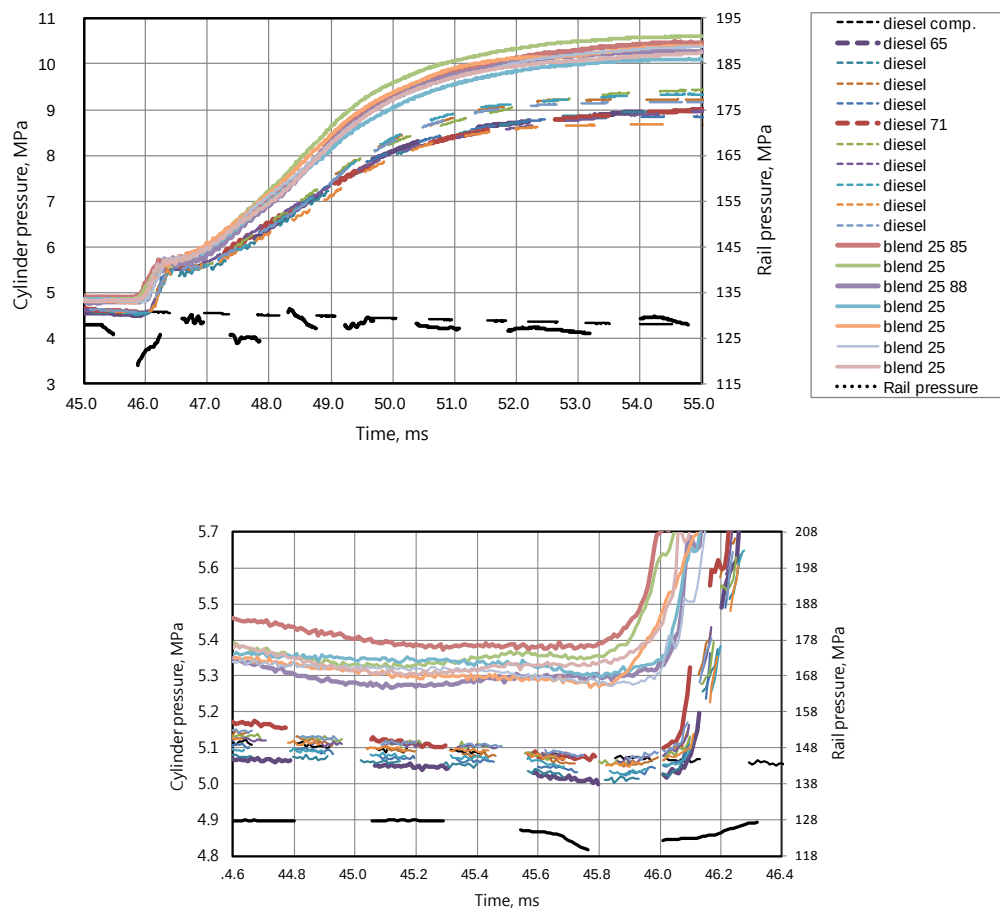


Figure 6.7 Cylinder pressure deviations during the test of the diesel and the 25% blend oil (blend25) at the RCM test

The maximum and minimum ignition delay for the oils is plotted according to the cylinder pressure analysis in Figure 6.7, the solid mark was analyzed one from the cylinder pressure and the dot line was just extrapolated. The ignition delay decreased according to the elevation of the cylinder pressure for the two oils as shown in the top one of Figure 6.8. But the sensitivity is larger for the blend oil than for the diesel and the ignition delay becomes longer at same cylinder pressure for the blend oil than the diesel. This trend is almost qualitatively identical to that in the engine test shown at the chapter 3.2.2.1. In bottom one of Figure 6.8, the ignition delay well correlates with the mean cylinder temperature and it shows the same tendency, longer ignition delay at the same mean cylinder temperature. It is interesting that at higher mean cylinder temperature the ignition delay become equal and it can be a hint to improve performance deterioration caused by the larger ignition delay.

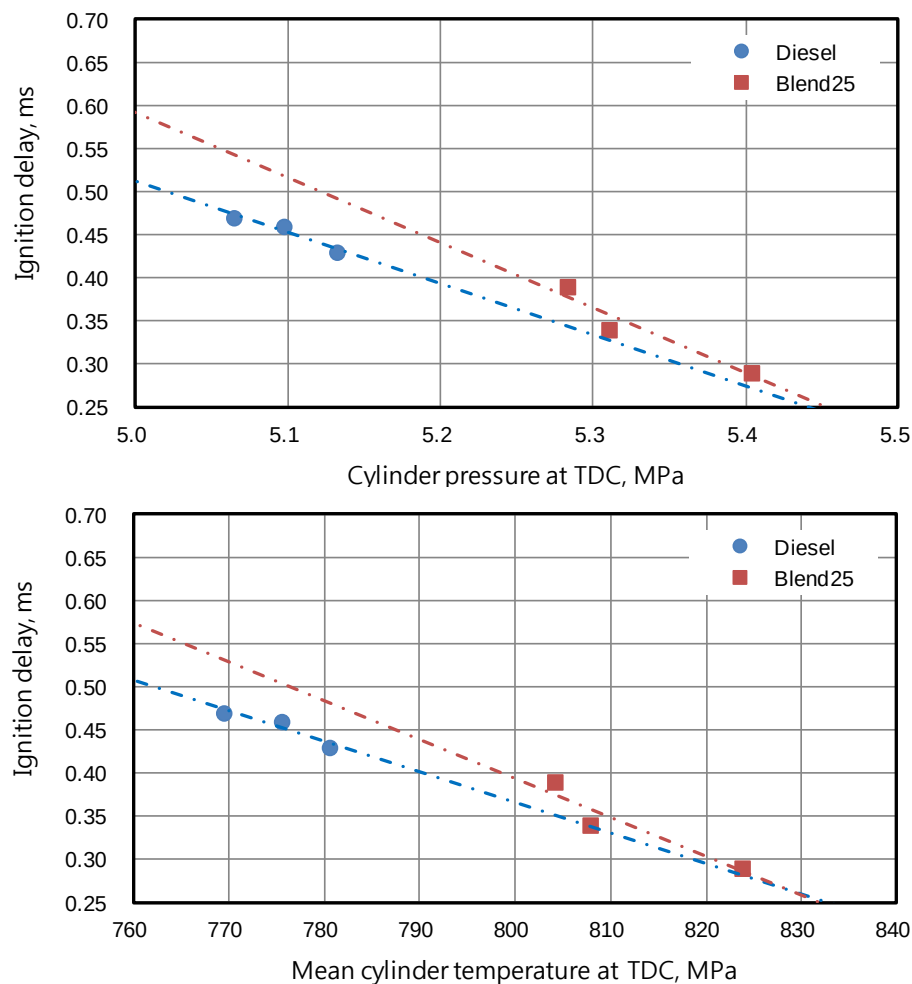


Figure 6.8 Ignition delay depending on the cylinder pressure (top) and the mean cylinder temperature (bottom) at TDC

Further summary

What are concerns to increase the blend ratio further or to improve engine performance? and what can be measured?

When we try to increase the blend ratio, the concerns can be very simply summarized as shown in Table 6.1. The concerns differs between the bio-oil and the plastic oil

Table 6.1 Summary of concerns

	Bio-oil	Plastic oil
Power output	Even	Worse
NOx	Worse	Even or better
THC	Worse	Even or better
CO	Worse	Even or better
Smoke	Worse	Even or better
Fuel consumption	Worse	Worse
Injection system	Worse	Even or worse

Bio-oil

Characteristic changes with the blend ratio: higher density and kinematic viscosity, longer ignition delay, higher injection rate, higher water amount

What has to be focused to measure with each concern.

THC: Better air-fuel mixing

CO: Reduced pre-mixed combustion, better air-fuel mixing

Smoke: Better air-fuel mixing, higher overall lambda, improved spray impingement

Fuel consumption: Advanced Pmax producing

Injection system: Preventing the bio-oil from polymerization

What can be reasonably measures?

Advancement of injection timing: CO may reduce in some extent due to the lower pre-mixed combustion fraction but mainly fuel consumption improvement (power increase) with the advanced Pmax producing but THC and Smoke remain as concern. The magnitude of advancement can be referenced from Figure 6.9.

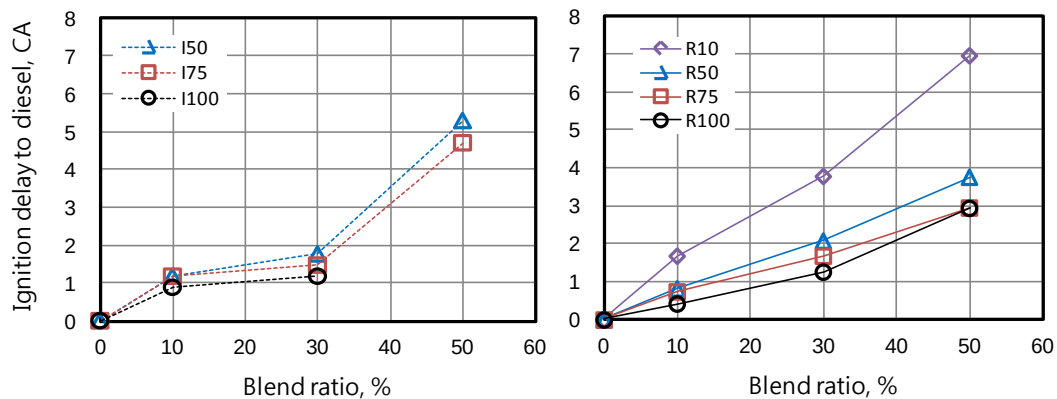


Figure 6.9 Relative ignition delay according to the blend ratio for the bio-oil (0%: diesel)

Induction air heating: CO reduces due to the lower pre-mixed combustion fraction, fuel consumption improves with the advanced Pmax producing caused by the shortened ignition delay but THC and Smoke (much worse by poorer volumetric efficiency and lambda decrease) remain as concern.

Fuel line heating: This makes the kinematic viscosity reduced and the spray atomization improved. As a result, THC, CO and smoke may be reduced. But the risk of the polymerization may be increased and the heating temperature must be carefully controlled. If the heating could contribute the ignition delay, CO would be additionally decreased and the engine power would be increased.

Modification of piston bowl shape: The entrance of piston bowl in side of no. 1 spray should be enlarged and the liquid spray can expand without impinging with piston wall. This can play a role to reduce mainly smoke production and additionally CO and THC can be affected positively.

Plastic oil

Characteristic changes with the blend ratio: lower density and kinematic viscosity, longer ignition delay, lower injection rate

What has to be focused to measure with each concern.

Power output: higher injection rate, advancement of Pmax producing timing

Fuel consumption: Same above

What can be reasonably measures?

Advancement of injection timing: The engine power can be increased with this measure but the improvement magnitude may be very low. The magnitude of advancement can be referenced from Figure 6.10.

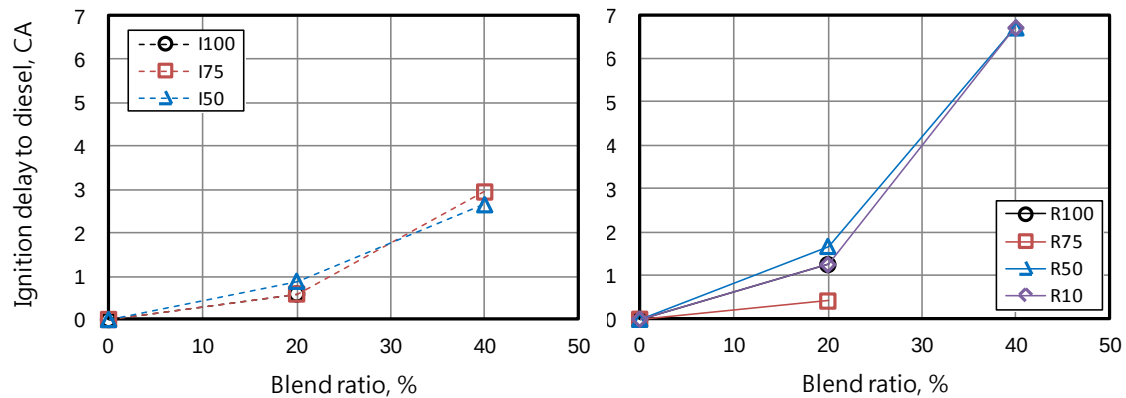


Figure 6.10 Relative ignition delay according to the blend ratio for the plastic oil (0%: diesel)

Induction air heating: It can contribute to shorten the ignition delay but the effectiveness on the engine power elevation may be very limited level. It mainly play a positive role to enlarge the blend ratio due to the shorten ignition delay.

Modification of injector nozzle hole in number or diameter: Increase of nozzle hole number or enlarge of nozzle hole diameter can make significant change in increasing injection rate and result in power increase. But the spray atomization tends to be poor (smoke increase) with the enlarged nozzle hole diameter and this should be careful in the modification.

Each measure can be chosen independently or in combination but must be carefully adapted with consideration of effort. The effort level (hardware build-up, evaluation and optimization) may be increased in the sequence of the fuel line heating, induction air heating, injection timing, piston bowl shape, nozzle hole diameter and nozzle hole number.

6.2 Conclusions

The results are summarized that

For the 25% blend oil of the bio-oil with the diesel in the RCM (High pressure fuel injection system):

1. For the given experimental conditions, the ignition delay of blend 25 was slightly shorter than that of diesel fuel. The combustion period was almost dominated by the mixing controlled combustion phase for the two fuels. During the mixing controlled combustion phase, blend 25 showed a higher mean temperature and a faster heat release rate contributing to complete combustion in almost the equal time even with more injection amount than that of diesel. This resulted in less smoke emission by 80% and higher apparent combustion efficiency while NO_x emission increased by 40% comparing to diesel fuel.

For the blend oil of the bio-oil with the bio diesel, it is found that the present bio-oil can be utilized in the diesel engine (single cylinder direct injection with 20MPa injection pressure, 0.21L, 3kW/3500rpm):

1. 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.
2. It should be careful of the increase of the kinematic viscosity according to the raw biomass and the pyrolysis process because it may affect the spray condition in negative direction resulting in higher exhaust emissions like smoke, CO and THC emission. Additionally, higher water content may attribute to longer ignition delay resulting in misfire at part loads.
3. Through the 8 hours engine run with the 30% blend oil, the fuel injection system of the engine was encountered with the irreversible failure. It is thought to be due the polymerization of the bio-oil which was observed in the fuel delivery line. It can be said that the 5% surfactant of the methanol was not sufficient. Abnormal carbon deposit inside the engine such as the surface of the nozzle tip, the engine head and the piston was observed.

For the blend oil of the waste plastic pyrolysis oil with the diesel fuel, it is found that the present bio-oil can be utilized in the diesel engine (single cylinder direct injection with 20MPa injection pressure, 0.21L, 3kW/3500rpm):

1. Up to around 20% blending ratio, the engine operation was compatible to the diesel. It resulted that the full load engine output was lower about 17%, the exhaust emissions (NO+THC, CO, smoke) were met marginally to the EPA Tier1 standard while and the 8 hours continuous operation was successfully completed.
2. The chemical property variation from the raw waste material and the pyrolysis should be kept in mind in terms of the ignition delay which can make misfire. But it is not easy and it may be one approach that the blend ratio is started lower and gradually increased.

The ignition delay was mainly resulted from the oil property and the water content might be the main reason (Chemical analysis was not able to be done and it will be a great theme for future study) for the bio blend oil and the chemical content was major contribution for the plastic blend oil. During the ignition delay, the amount of air-fuel mixture differed between the bio-oil and the plastic oil due to differentiation in the physical property such as the density and the kinematic viscosity. This might be one of reasons to make the engine performance (power and exhaust emissions) different in the both blend oils.