



Combustion, performance and emission characteristics of direct injection diesel engine fueled by Jatropha hydrogen peroxide emulsion

Kim-Bao Nguyen
Dan, Tomohisa
Asano, Ichiro

(Citation)

Energy, 74:301-308

(Issue Date)

2014-09-01

(Resource Type)

journal article

(Version)

Accepted Manuscript

(Rights)

©2014.

This manuscript version is made available under the CC-BY-NC-ND 4.0 license
<http://creativecommons.org/licenses/by-nc-nd/4.0/>

(URL)

<https://hdl.handle.net/20.500.14094/90003499>



Combustion, Performance and Emission Characteristics of Direct Injection Diesel Engine Fueled by Jatropha Hydrogen Peroxide Emulsion

Kim-Bao NGUYEN¹, Tomohisa DAN^{2,*}, Ichiro ASANO²

¹Graduate School of Maritime Sciences, Kobe University

5-1-1 Fukaeminami, Higashinada-ku, Kobe, 658-0022, Japan

²Faculty of Maritime Sciences, Kobe University

5-1-1 Fukaeminami, Higashinada-ku, Kobe, 658-0022, Japan

*Corresponding author:

Tomohisa DAN,

E-mail: dan@maritime.kobe-u.ac.jp

Phone: +81-78-431-6289

Address: 5-1-1 Fukaeminami, Higashinada-ku, Kobe, 658-0022, Japan.

Abstract

Recently, Jatropha, a fast growing tropical plant, has caught the attention of researchers because the oil produced from Jatropha is a non-edible. However, worse emissions, when using raw Jatropha oil in diesel engines, have been reported in the literature due to its higher viscosity and lower volatility. Therefore, it is necessary to find some way to reduce the emissions of diesel engines.

In the present research, experiments have been designed and performed to study the effects of Jatropha hydrogen peroxide emulsion on combustion, performance, and the emission characteristics of a diesel engine. Hydrogen peroxide in a solution of water at a concentration of 30% was used for making the emulsions with mixing mass ratios of 5%, 10%, and 15%. A single cylinder, four-stroke, high speed, direct injection diesel engine was used for the experiments. The acquired data were analyzed for various combustion parameters such as in-cylinder pressure, ignition delay, ignition duration, heat release, for performance parameters and emissions of CO, CO₂, HC, NO_x, and PM as well. While running the engine on the Jatropha hydrogen peroxide emulsion fuel, improvements in performance and emissions were found and the optimum mixing ratio of the solution with hydrogen peroxide was 15%.

Keywords: Jatropha oil, Jatropha hydrogen peroxide emulsion, combustion, performance, emission

1. Introduction

Recently, the use of diesel engines has been facing problems such as the fossil fuel crisis and air pollution. Harmful pollutants, namely Nitrogen oxides (NO_x) and Particulate matter (PM), from diesel engines are very difficult to reduce simultaneously. Additionally, the production of the global warming gas (CO₂) is unavoidable whatever the fuel when using diesel engines. Vegetable oils, that are biodegradable, that have low aromatic and sulfur content, have recently gained attention as a promising alternative fuel for a greener future. Short-term tests have indicated that most vegetable oils are capable of being used directly in existing diesel engines with little or no modification. However, in long-term tests, some issues were reported such as piston ring sticking, injector coking, severe engine deposits, gum formation and oil thickening [1-3]. This is primarily attributed to the high viscosity and poor volatility due to their large molecular weight and bulky molecular structure. These physical properties of vegetable oils lead to poor atomization, inadequate mixing with air during injection and atomization process, consequently contribute to incomplete combustion when using vegetable oil as fuel for diesel engines. This results in an increase in particulate, CO, and HC emissions, but lowers NO_x emissions compared to those when using diesel oil [4-8]. Among vegetable oils, *Jatropha curcas* has been of interest recently, because it is a non-edible oil, thus there is no issue of the food-fuel conflict [7].

Jatropha oil was identified as a leading candidate for an alternative fuel for commercialization among various non-edible vegetable oils [9]. *Jatropha* plant can be grown in arid conditions on almost any kind of ground, and does not suffer excessively from droughts or require concentrated irrigation. Therefore, unlike other common biofuel crops, it is very easy to cultivate, even on poor soil. High viscosity, low volatility, and low cetane number of the *Jatropha* oil have been reported in previous studies [10-12]. M. S. Kumar et al. [7] observed higher smoke, HC, CO and NO_x emissions of the engine operated with *Jatropha* oil. Lower NO_x and break thermal efficiency, and higher CO, HC of *Jatropha* oil compared to those of diesel were reported by B. S. Chauhan et al. [13] due to the high viscosity and low volatility. Higher smoke, CO, HC emissions of *Jatropha* oil were also observed in the studies of D. Agarwal et al. [11], J. N. Reddy et al. [14], and A. S. Ramadhas et al. [15]. The drawbacks of *Jatropha* oil could be overcome by preheating [13] or esterification and blending with diesel [16, 17], increased injection pressure, or perhaps a dual fuel engine.

The benefits of the water emulsified fuels to the combustion and emissions of compression ignition engines have been tested, and proven in the literature [18-25, 26]. Recently, *Jatropha* methyl ester emulsified with wood pyrolysis oil was tested and its benefit was observed [27]. But, operation of diesel engine fueled with *Jatropha* hydrogen peroxide emulsion has not been reported as yet. Hydrogen peroxide (H₂O₂) is effective for soot oxidation due to the presence of the Hydroxyl radical (OH) dissociated during the combustion process as reported in the studies of B. Franz. et al. [28] and C. Born et al. [29]. K. S. Nagaprasad et al. [30] found higher engine break thermal efficiency with increased concentrations of H₂O₂, and D. S. -K. Ting et al. [31] reported a significant

reduced ignition delay with addition of H_2O_2 . The burning rate of methane fuel was enhanced owing to the presence of OH, O, and H radicals thus produced less CO emissions [31-32]. M. C. Mulenga et al. [33] reported that NO_x emissions were lower as a result of a lower average cycle temperature. Hence, it can be summarized that the addition of H_2O_2 to the engine combustion chamber improves the combustion and emissions. However, addition of H_2O_2 to Jatropha oil has not been tested until now.

Therefore, this research was designed and conducted to gather information on the engine performance, combustion and emissions and to identify the optimum mixing ratio of hydrogen peroxide solution to Jatropha oil.

2. Material and Experimental setup

In this study, test fuels include the neat Jatropha oil (JO), Jatropha hydrogen peroxide emulsions (JHE) with light oil (LO) as the baseline fuel. The properties of the test fuels are given in Table 1. Hydrogen peroxide in a solution of water at a concentration of 30% was used for making the emulsions in this research. To make the emulsion fuel, a mixing system was used that included tanks for Jatropha oil and chemical (H_2O_2 solution), a circulating pump, and a static mixer. The Jatropha oil mass was measured during each step of the experiments. Mixtures of Jatropha oil with solutions of hydrogen peroxide were created according to the Jatropha oil mass with mixing ratios of 5%, 10% and 15% (hereafter so-called JHE5%, JHE10%, and JHE15%). To keep emulsified fuels homogeneous and stable, three kinds of surfactants (Rheodol SP-L 10, Rheodol 440V, and Emulgen 103, Kao Chemicals Corp., Japan) were used. While the Jatropha oil was circulating through the mixing pump, the chemical was added through the chemical tank and the flow rate of the chemical was controlled by a needle valve for homogeneous mixed fuels. A schematic diagram of the mixing system is shown in Fig. 1a.

The experiments were conducted on a single cylinder, four-stroke, high speed, direct injection diesel engine (Yanmar Co., Ltd., Japan). To inject the fuel into the combustion chamber, a set of mechanical fuel injection system including a fuel pump and a mechanical injector was utilized. The specifications of the engine are given in Table 2. The engine was equipped with a piezoelectric pressure sensor and a rotary encoder for measurement of the in-cylinder pressure and crank angle. The pressure and crank angle signals are transmitted to a computer via a charge amplifier. These signals were used for monitoring cylinder pressure history and for calculation of heat release after the experiments. Load of the engine was set through a water-dynamometer coupled to the shaft of the engine. A set of chemical luminescence-type gas analyzers was used to measure the emissions emitted from the engine and a PM meter was used to collect PM at each step of the experiments. Standard error was used to evaluate the measurement accuracy. In this study, it is under 5% for the measurements.

Firstly, the engine was fed with the light oil and the results were recorded at each load of the engine. After this, we switched to the test fuels including neat Jatropha oil, and Jatropha hydrogen peroxide emulsions. During the experiments, the engine load was set at different values of 1.55 kW, 3.02 kW, 4.67 kW, and 6.20 kW with a constant speed of 2000 rpm. At each load, reading the emissions of CO, CO₂, HC, and NO_x was made through the exhaust gas analyzers, while, the Particulate matters (PM) were collected on the filter papers of the PM meter. A schematic diagram of the experimental set-up is illustrated in Fig. 1b.

3. Theoretical consideration

The apparent net heat release rate was calculated by using the first law of thermodynamics [34] from recorded pressure data with following formula:

$$\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta}$$

where γ is the ratio of specific heats, $\gamma = C_p/C_v$, P is instantaneous cylinder pressure, (Pa), V is instantaneous cylinder volume, (m³). The value of γ depends on temperature, T (K), of the gas in the cylinder and was calculated by equation [35]:

$$\gamma = 1.338 - 6.0 \times 10^{-5} * T + 1.0 \times 10^{-8} * T^2$$

Ignition delay was computed as a period between the start of injection (SOI) and the start of combustion (SOC). Combustion duration was calculated by interval between the SOC and the end of combustion (EOC). The SOC and EOC were determined from heat release history, and the SOI was determined according to the opening pressure of the injector and injection pressure history.

4. Results and discussion

4.1 Combustion characteristics

The combustion process of a diesel engine depends on the fuel-air mixing process and is related to fuel properties, such as viscosity, volatility, and the fuel injection system itself. With existing engines, the combustion is determined mainly by the properties of the fuel and operational parameters. In this study, the combustion characteristics are demonstrated mainly by a number of factors, namely combustion pressure, ignition delay, combustion duration, heat release rate, net cumulative heat release and center of combustion.

In-cylinder pressure of the engine is indicated in Fig. 2. It is clear that the peak pressures of the JHE fueled engine were higher than those of Jatropha oil or light oil in most cases. Moreover, the pressure of the JHE fuel was more advanced in comparison to those of the light oil and neat Jatropha oil except the case of JHE15% at engine load of 1.55 kW. This resulted from the physical characteristics of the JHE fuel such as high compressibility and the effect of H₂O₂ in the combustion process. At lower load, the

combustion was far from the top dead center (TDC), and the combustion phase was closer to the TDC when the engine load increased. This is due to the fact that the fuel was injected earlier at higher engine loads.

The start of combustion and the ignition delay are shown in Fig. 3a and 3b, respectively. At low load, the low combustion temperature resulted in longer ignition delay and combustion initiated later for the JHE15% fuel. The ignition delay was shorter and combustion started earlier for the JHE5%. At higher loads of the engine, for the JHE5% and JHE15% fuels, the ignition delays were shortened and the start of combustion was earlier in comparison to those of the neat Jatropha oil or light oil. This resulted from the advanced injection of the emulsion fuels and perhaps the presence of the hydrogen peroxide that enhanced the combustion of the emulsion fuel.

The center of combustion, at which 50% of heat release occurs, is indicated in Fig. 4a and the combustion duration is presented in Fig. 4b. Even though the start of combustion was earlier, it is clear that the combustion center for emulsion fuels tended to be toward the later stages of combustion, especially for the JHE5% and JHE15% fuels. This is evidence of much more heat release and longer combustion duration as shown in Fig. 4b. This may result from the oxidation of the soot by the OH radicals in the later combustion phase releasing more heat, so it prolongs the combustion.

Heat release rate (HRR) in the cylinder of the engine is presented in Fig. 5. The sharp and highest increment of the heat release rate is for the light oil. The fast combustion of the light oil results from the physical properties of fossil fuel, such as low viscosity, and high volatility, thus it is easier for atomizing and mixing with air. It is clear that, the combustion of the emulsion fuels initiated earlier, but the maximum values were lower in comparison to those of the light oil or the neat Jatropha oil. At 1.55 kW, 3.02 kW, 4.67 kW and 6.20 kW, the peaks of the HRR were 62.4, 66.2, 71.2, and 69 for the LO, respectively, while the JHE15% caused a relative reduction of 28.4%, 25.2%, 39.3%, and 33% when compared to the LO. For the other emulsion fuels, the peaks of the HRR were between 44.1 and 56.3 depending on the fuel and load conditions. This may result from the cooling effect of the water content in the emulsion fuels and their lower heating value. In the diffusion combustion phase, there is a large magnitude of oscillation in the heat release rate and the release of heat lasted into the late combustion phase with the JHE5% and JHE15% fuels. This can be attributed to the heat radiated from oxidation of soot by the OH radicals in the late combustion phase.

Net cumulative heat release (HR) is shown in Fig. 6. It is evident that the heat release was higher for the emulsion fuel, but the complete combustion of the emulsion fuel wasn't until the later stage. At 1.55 kW, the peaks of the HR were 586.9, and 568.4 for the LO and JO while the JHE5%, JHE10%, JHE15% increased 23.4%, 15.9%, and 22.9% when compared relatively to the JO. At 6.20 kW, it was 1174 for the JO, and climbed up to 1274.5, 1288.9, and 1322.5 with a relative increment of 8.6%, 9.8% and 12.6% for the JHE5%, JHE10%, JHE15%, respectively, when compared to the JO. The longer combustion of the emulsion fuel was explained

previously as a result of oxidation of soot particles by the OH radicals in the later stage. The longer heat release rate due to soot oxidation contributed to higher and later heat release of emulsified fuel. It was especially true for the JHE5% and JHE15%.

4.2 Performance characteristics

In this section, the performance parameters of the engine such as in-cylinder and exhaust gas temperatures, and break thermal efficiency will be introduced. Combustion temperature was calculated from the in-cylinder pressure history. Exhaust gas temperature was measured by the thermocouple and recorded by computer during the experiments. Break thermal efficiency was calculated as the ratio of the brake power to the energy supplied by the fuel consumption.

In-cylinder temperature is presented in Fig. 7. It can be seen that in-cylinder temperatures of the engine when fueled with the neat Jatropha oil and light oil were very close. However, there were relatively large differences for emulsion fuels especially in the late combustion stages. The JHE5% and JHE15% showed higher in-cylinder temperatures in most cases. In comparison with the JO, the JHE5% had an increment in in-cylinder temperatures of 112.2, 166.4, 247.9, and 204.8 °C, while the JHE15% induced 22.5, 43.9, 131.8, and 136.5 °C higher than those of the JO at 1.55, 3.02, 4.67, and 6.20 kW, respectively. This was a consequence of the heat release rate and the heat release histories. Higher heat releases of emulsion fuels in the later combustion stage caused higher in-cylinder temperatures.

Exhaust gas temperature is depicted in Fig. 8a. Lower temperatures were at lower loads and increased with an increase of engine load. This is because more fuel is needed to burn to produce higher power. Exhaust gas temperatures of the engine fueled with emulsified fuels were higher than those of the light oil or neat Jatropha oil, especially for the JHE5% and JHE15% at higher loads. At 1.55 kW, exhaust gas temperatures were 239.2, 251.4 °C for the LO, JO, while they climbed up to 269.2, 252.8, and 264 °C for the JHE5%, JHE10%, JHE15%, respectively. At 6.20 kW, it was 576.2 °C for the JO, and the JHE5%, JHE10%, JHE15% increased 9.6%, 2.5%, 4.2% rising to 631.6, 590.5, 600.2 °C, respectively. Higher heat release, and a longer-extended-combustion stage resulted in increasing the exhaust gas temperatures. Earlier completed combustion of the JHE10%, and high heat release in the premixed combustion phase lowered exhaust gas temperatures as compared to the JHE5% or JHE15%.

Fig. 8b shows the break thermal efficiency (BTE) of the engine. The BTE was the highest for light oil fuel due to its better combustion characteristics. For the JO, starting at 18.7% at a load of 1.55 kW, then BTE climbed up to around 26% at medium and higher loads. At loads of 1.55 kW, and 3.02 kW, when compared relatively to the JO, the BTE of the JHE10% were reduced 7.7% and 13.7%, while they increased 26.7% and 2.7% for the JHE15%. At higher loads, both emulsion fuels increased BTE, particularly for the JHE15% with an increment of 15.7% compared relatively to the JO. For the JHE10%, the reduction in BTE at lower loads

may be attributed to the cooling effect, and not enough hydrogen peroxide to improve combustion. To produce higher power, more fuel quantity must be injected into the combustion chamber, thus 15% of a solution of hydrogen peroxide was enough to enhance combustion and increase thermal efficiency as compared to the neat Jatropha oil and other emulsion fuels.

4.3 Emission characteristics

The emissions of the engine including CO₂, CO, HC, NO_x, particulate matter and soot are indicated in Fig. 9. Fig. 9a shows clearly that the emissions of CO₂ decreased with an increase in the engine power. This results from the better combustion condition at higher loads. Moreover, JHE5% and JHE15% reduced emissions of CO₂ when compared to the JO. The JHE5% had a reduction rate of 16.4%, 17.2%, 11.5%, and 8.7% corresponding to the engine power from 3.02 kW to 6.20 kW, respectively, when compared to the JO. For the JHE15%, the reduction rate was 16%, 14%, 7.4%, 6.6%. This can be attributed to enhanced heat release from oxidation of soot in the later combustion stage, thus higher thermal efficiency, and less fuel consumed while lowering CO₂ emissions.

CO emissions, resulting from incomplete combustion, versus engine power are illustrated in Fig. 9b. When compared to the JO, at low load, due to the inferior combustion condition and the cooling effect, for the JHE10%, the emissions of CO increased 14.5%, while the JHE5%, JHE15% gave a comparable result. At medium loads, temperatures in the cylinder were elevated, so the emulsion fuels were more completely combusted. At 3.02 kW and 4.67 kW, the reduction rate was up to 23.9% and 20.5% for the JHE5%, while the JHE15% reduced 14.9% and 35.6%, respectively. At 6.20 kW, much more fuel had to be injected into the combustion chamber, therefore, the local rich fuel combustion and the cooling effect of the JHE5% caused an increment of 37%. While the JHE10% diluted the fuel to avoid local rich fuel injection resulting in a reduction of CO of 4.6%.

Fig. 9c indicates HC emissions of the engine. The emissions of HC decreased with an increase in the engine power. This is due to the higher combustion temperatures at higher engine loads. The emulsified fuels produced much more HC emissions especially at low loads. At 1.55 kW, HC increased 82%, 116.8%, and 102.9% for the JHE5%, JHE10%, and JHE15%, respectively, when compared relatively to the JO. This may result from the higher density of the emulsified fuels that caused to deep entrancement of the sprays. Moreover, at low loads the cooling effect of the emulsion fuel was dominant due to the low combustion temperatures. At higher loads, the differences in emissions of HC between the Jatropha oil and the emulsified fuels became smaller due to the better combustion. At 6.20 kW, it is observed that the HC emissions of the JHE5%, JHE10%, and JHE15% had an increment of 1.8%, 2.2%, and 14.3%, respectively, when compared relatively to the JO.

NO_x emissions are depicted in Fig. 9d. NO_x emissions of the JHE10% were higher than the Jatropha oil in most cases due to the higher peak heat release rate. When compared relatively to the JO, the JHE10% increased 7%, 4.3% and 7.2% at 1.55 kW, 4.67 kW, and 6.20 kW, respectively. NO_x emissions form at high temperatures where the heat release rate may be at the maximum. For the JHE5% and JHE15%, the lower peaks of the heat release rates, and the cooling effect resulted in lower NO_x emissions in most cases when compared to the Jatropha oil. The JHE15% had a reduction rate of 3.7%, 16.5%, 0.8% at 1.55 kW, 3.02 kW, and 4.67 kW, respectively, while it had an increase of 4.8% at 6.20 kW. The JHE5% seems to be the best for reduction NO_x emissions.

Particulate matter and soot emissions are shown in Fig. 9e and Fig. 9f, respectively. It is clear that PM decreased with an increase of the engine power for most emulsified fuels. This results from the poor combustion conditions at low load, thus the less oxidation of soot. At medium loads, the combustion temperatures elevated, and the injected fuel was not rich, and as a result the oxidation of soot by the OH radicals was reason enough for reducing soot and PM. When compared relatively to the JO, at 3.02 kW, the JHE5% and JHE15% had reduction rates of 19.5%, and 28% for PM, while their reductions were 20.6% and 22.3% at 4.67 kW, respectively. At these load, the reduction of soot was 16.6% for the JHE5% and up to 38% for the JHE15%. At maximum power, the injected fuel became local rich enough to increase soot and thus the PM tended to increase slightly again. However, for the JHE10% and JHE15%, the reductions were observed. The reduction of PM was observed of 28.6%, and 26.6% for the JHE10%, and JHE15%, respectively. While the soot reductions were 46.4% and 27.6% for the JHE10% and JHE15%, respectively. Overall, we found that the JHE15% gave the lowest emissions of PM at all loads and for soot reduction at higher loads. This is attributed to the oxidation of soot by the OH radicals released during the combustion process.

5. Conclusion

A direct injection diesel engine operated with neat Jatropha oil, and a Jatropha hydrogen peroxide emulsion with mixing ratios of 5%, 10% and 15% were studied to investigate the effects of the JHE fuel on the combustion, performance, and emissions of the engine. In summary, the main features are as follows.

- 1- The engine has operated successfully on the Jatropha hydrogen peroxide emulsion up to the 15% mixing ratio. The combustion of the emulsified fuels shifted to the earlier stage and extended to the later stage. The peak of the heat release rate was reduced with emulsified fuels and there was a large magnitude of oscillation in the heat release rate in the diffusion combustion phase. The heat release rate in the late combustion stage for the emulsified fuels was higher than those of the Jatropha oil and light oil, and the cumulative heat release for the emulsified fuel was also higher.

- 2- In-cylinder and exhaust gas temperatures were higher especially for the JHE5% and JHE15% as compared to those of the Jatropa and light oil. The JHE15% improved break thermal efficiency of the engine in most cases as compared to other emulsion fuels and Jatropa oil.
- 3- The JHE5% and JHE15% reduced CO₂ and NO_x emissions, while, they also reduced CO emissions at low and medium loads. The emulsion fuels increased HC emissions. At medium and high power conditions, PM and soot emissions were reduced by the emulsion fuel, especially for the JHE15%. Overall, this study found that the optimum mixing ratio of hydrogen peroxide to improve combustion, performance and emissions in diesel engine was 15%.

Acknowledgement

We would like to acknowledge our Lab-members, Mr. NISHIMURA, Mr. YAMAMOTO, Mr. YAMAMURA, Mr. YOSHIDA, at the Internal Combustion Engineering Laboratory, Faculty of Maritime Sciences, Kobe University for their help in conducting the experiments.

References

- [1] A. K. Agarwal and L. M. Das. Biodiesel development and characterization for use as a fuel in compression ignition engines. J. Eng. Gas Turbines Power 2000, 123(2), p. 440-447.
- [2] R. Altın, S. Çetinkaya and H. S. Yücesu. The potential of using vegetable oil fuels as fuel for diesel engines. Energy Conversion and Management 2001, Volume 42, Issue 5, p. 529–538.
- [3] A. K. Agarwal, J. Bijwe and L. M. Das. Effect of biodiesel utilization of wear of vital parts in compression ignition engine. J. Eng. Gas Turbines Power 2003, 125(2), p. 604-611.
- [4] Hemmerlein, N., Korte, V., Richter, H., and Schröder, G.. Performance, exhaust emissions and durability of modern diesel engines running on rapeseed oil. SAE Technical Paper 910848, 1991.
- [5] C W Yu, S Bari, A Ameen. A comparison of combustion characteristics of waste cooking oil with diesel as fuel in a direct injection diesel engine. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering 2002, Vol. 216, no. 3, p. 237-243.
- [6] O.D. Hebbal, K.Vijayakumar Reddy, K. Rajagopal. Performance characteristics of a diesel engine with deccan hemp oil, Fuel 2006, Volume 85, Issues 14-15, p. 2187-94.
- [7] M. S. Kumar, A. Ramesh, B. Nagalingam. An experimental comparison of methods to use methanol and Jatropha oil in a compression ignition engine. Biomass and Bioenergy 2003, Volume 25, Issue 3, p. 309-318.
- [8] A. S. Huzayyin, A.H. Bawady, M.A. Rady, A. Dawood. Experimental evaluation of diesel engine performance and emission using blends of jojoba oil and diesel fuel. Energy Conversion and Management 2004, Volume 45, Issues 13-14, p. 2093-2112.
- [9] S.-Y. No. Inedible vegetable oils and their derivatives for alternative diesel fuels in CI engines: A review. Renewable and Sustainable Energy Reviews 2011, Volume 15, Issue 1, p. 131-149.
- [10] Lujaji, F., Bereczky, A., Janosi, L., Novak, Cs., Mbarawa, M.. Cetane number and thermal properties of vegetable oil, biodiesel, 1-butanol and diesel blends. Journal of Thermal Analysis and Calorimetry 2010, Volume 102, no. 3, p. 1175-1181.
- [11] D. Agarwal, A. K. Agarwal. Performance and emissions characteristics of Jatropha oil (preheated and blends) in a direct injection compression ignition engine. Applied Thermal Engineering 2007, Volume 27, Issue 13, p. 2314-2323.
- [12] K. Pramanik. Properties and use of jatropha curcas oil and diesel fuel blends in compression ignition engine. Renewable Energy 2003, Volume 28, Issue 2, p. 239-248.
- [13] B. S. Chauhan, N. Kumar, Y. D. Jun, K. B. Lee. Performance and emission study of preheated Jatropha oil on medium capacity diesel engine. Energy 2010, Volume 35, Issue 6, p. 2484-2492.

- [14] J. N. Reddy, A. Ramesh. Parametric studies for improving the performance of a Jatropha oil-fuelled compression ignition engine. *Renewable Energy* 2006, Volume 31, Issue 12, p. 1994-2016.
- [15] A. S. Ramadhas, S. Jayaraj, C. Muraleedharan. Use of vegetable oils as I.C. engine fuels-A review. *Renewable Energy* 2004, Volume 29, Issue 5, p. 727-742.
- [16] B. S. Chauhan, N. Kumar, H. M. Cho. A study on the performance and emission of a diesel engine fueled with Jatropha biodiesel oil and its blends. *Energy* 2012, p. 616-622.
- [17] M. Mofijur, H.H. Masjuki, M.A. Kalam, A.E. Atabani. Evaluation of biodiesel blending, engine performance and emissions characteristics of Jatropha curcas methyl ester: Malaysian perspective. *Energy* 2013, p. 879-887.
- [18] H.Z. Sheng, et. al. The droplet group microexplosions in water-in-oil emulsion sprays and their effects on diesel engine combustion, Symposium (International) on Combustion 1994, Volume 25, Issue 1, p. 175-181.
- [19] R. J. Crookes, F. Kiannejad, M. A. A. Nazha. Systematic assessment of combustion characteristics of biofuels and emulsions with water for use as diesel engine fuels. *Energy Conversion and Management* 1997, Volume 38, Issues 15-17, p. 1785-1795.
- [20] O. Armas, R. Ballesteros, F.J. Martos, J.R. Agudelo. Characterization of light duty diesel engine pollutant emissions using water-emulsified fuel, *Fuel* 2005, Volume 84, Issues 7-8, p. 1011-1018.
- [21] M. Nadeem et. al. Diesel engine performance and emission evaluation using emulsified fuels stabilized by conventional and gemini surfactants. *Fuel* 2006, Volume 85, Issues 14-15, p. 2111-2119.
- [22] A. Lif, K. Holmberg. Water-in-diesel emulsions and related systems. *Advances in Colloid and Interface Science* 2006, Volumes 123-126, 16, p. 231-239.
- [23] K. Kannan and M. Udayakumar. NO_x and HC emission control using water emulsified diesel in single cylinder diesel engine, *ARPJ Journal of engineering and applied sciences* 2009, Volume 4, No 8, p. 59-63.
- [24] R. Ochoterena, A. Lif, M. Nydén, S. Andersson, I. Denbratt. Optical studies of spray development and combustion of water-in-diesel emulsion and microemulsion fuels, *Fuel* 2010, Volume 89, Issue 1, p. 122-132.
- [25] A. Maiboom, X. Tauzia. NO_x and PM emissions reduction on an automotive HSDI diesel engine with water-in-diesel emulsion and EGR: An experimental study, *Fuel* 2011, Volume 90, Issue 11, p. 3179-3192.
- [26] W. Zhang, Z. Chen, Y. Shen, G. Shu, G. Chen, B. Xu. Influence of water emulsified diesel & oxygen-enriched air on diesel engine NO-smoke emissions and combustion characteristics. *Energy* 2013, p. 369-377.
- [27] R. Prakash, R.K. Singh, S. Murugan. Experimental investigation on a diesel engine fueled with bio-oil derived from waste wood-biodiesel emulsions. *Energy* 2013, p. 610-618.

- [28] B. Fanz, P. Roth. Injection of a H_2O_2 /water solution into the combustion chamber of a direct injection diesel engine and its effect on soot removal. *Proceedings of the Combustion Institute* 2000, Volume 28, Issue 1, p. 1219-1225.
- [29] Born, C. and Peters, N.. Reduction of soot emission at a DI diesel engine by additional injection of hydrogen peroxide during combustion. SAE Technical Paper 982676, 1998.
- [30] Nagaprasad K. S., D. Madhu. Effect of injecting hydrogen peroxide into diesel engine. *International Journal of Engineering Sciences & Emerging Technologies* 2012, Volume 2, Issue 1, pp. 24-8.
- [31] D. S.-K. Ting, G. T. Reader. Hydrogen peroxide for improving premixed methane-air combustion. *Energy* 2005, Volume 30, Issues 2-4, p. 313-322.
- [32] G. B. Chen, . H. Li, T. S. Cheng, H. W. Hsu, Y. C. Chao. Effects of hydrogen peroxide on combustion enhancement of premixed methane/air flames. *International Journal of Hydrogen Energy* 2011, Volume 36, Issue 23, p. 15414-15426.
- [33] Mulenga, M., Ting, D., Reader, G., and Zheng, M.. The potential for reducing CO and NOx emissions from an HCCI engine using H_2O_2 addition. SAE Technical Paper 2003-01-3204.
- [34] J. B. Heywood. *Internal combustion engine fundamentals*. McGraw-Hill, 1988.
- [35] M. F. J. Brunt, K. C. Platts. Calculation of heat release in direct injection diesel engines. SAE Technical Paper 1999-01-0187.

LIST OF FIGURES

Figure 1 Schematic diagram of (a) mixing system, and (b) experimental set-up

Figure 2 In-cylinder pressures at power of (a) 1.55 kW, (b) 3.02 kW, (c) 4.67 kW, and (d) 6.20 kW

Figure 3 (a) Start of combustion, and (b) ignition delay at different engine powers

Figure 4 (a) Combustion center, and (b) combustion duration at different engine powers

Figure 5 Heat release rate at engine power of (a) 1.55 kW, (b) 3.02 kW, (c) 4.67 kW, and (d) 6.20 kW

Figure 6 Heat release at engine power of (a) 1.55 kW, (b) 3.02 kW, (c) 4.67 kW, and (d) 6.20 kW

Figure 7 In-cylinder temperatures at power of (a) 1.55 kW, (b) 3.02 kW, (c) 4.67 kW, and (d) 6.20 kW

Figure 8 Exhaust gas temperatures and brake thermal efficiency at different engine powers

Figure 9 Exhaust gas emissions at different engine powers

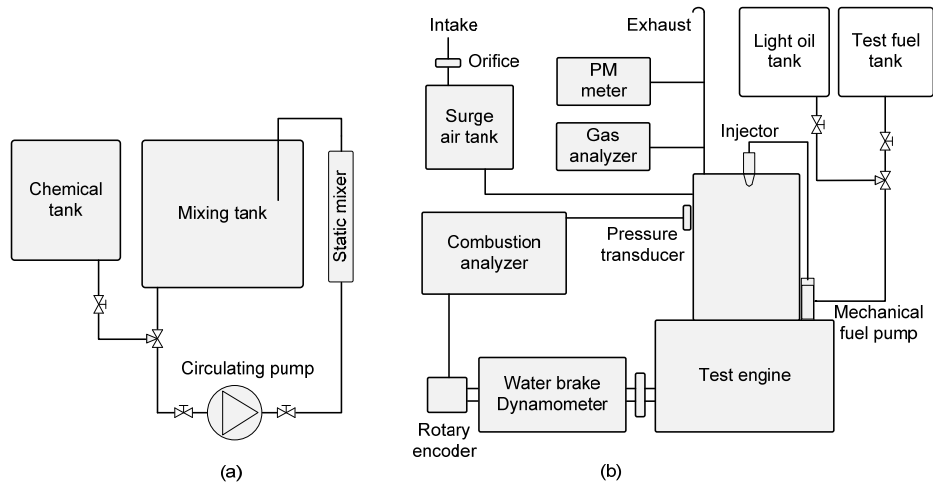


Figure 1 Schematic diagram of (a) mixing system, and (b) experimental set-up

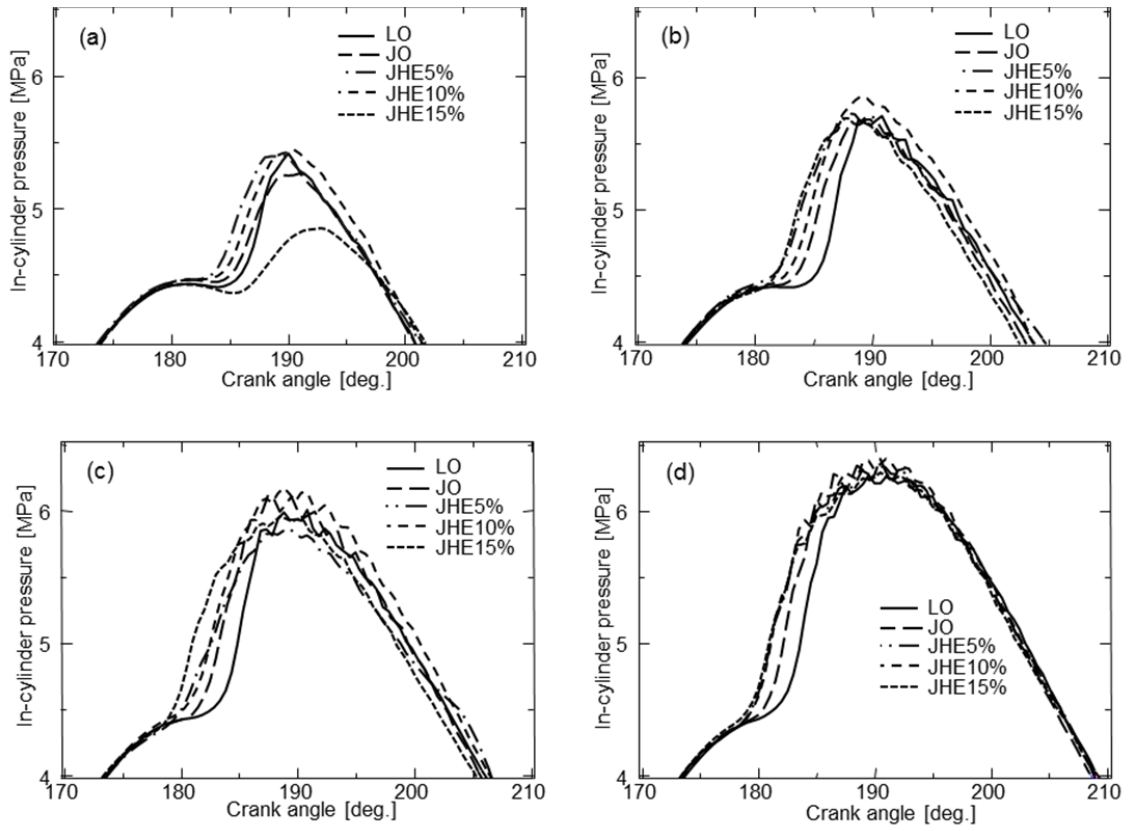


Figure 2 In-cylinder pressures at power of (a) 1.55 kW, (b) 3.02 kW, (c) 4.67 kW, and (d) 6.20 kW

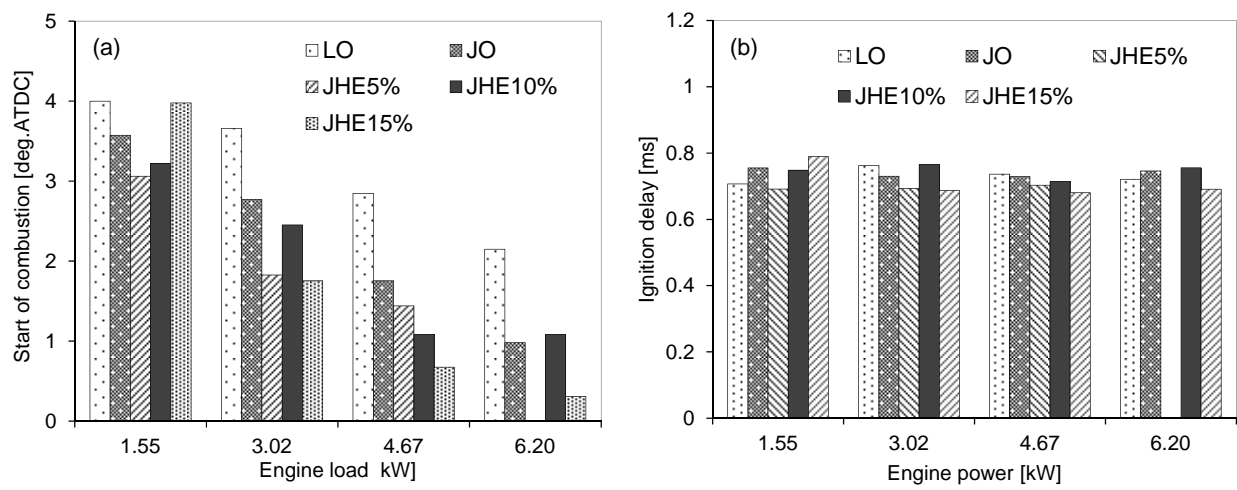


Figure 3 (a) Start of combustion, and (b) ignition delay at different engine powers

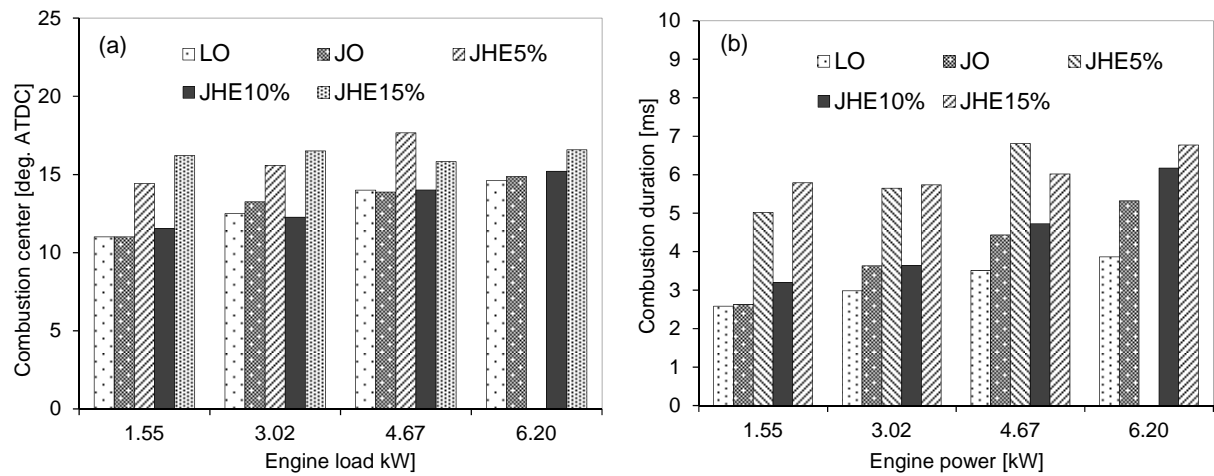


Figure 4 (a) Combustion center, and (b) combustion duration at different engine powers

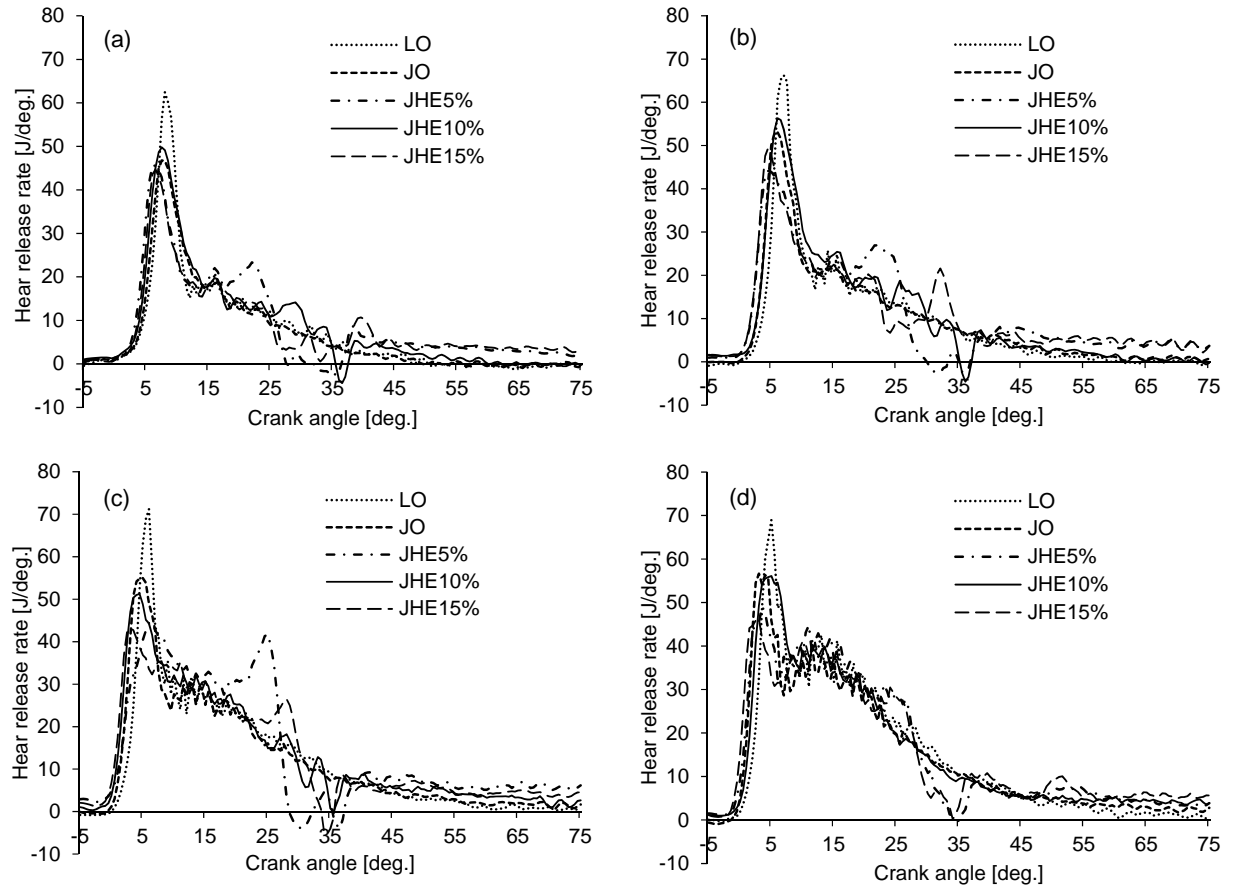


Figure 5 Heat release rate at engine power of (a) 1.55 kW, (b) 3.02 kW, (c) 4.67 kW, and (d) 6.20 kW

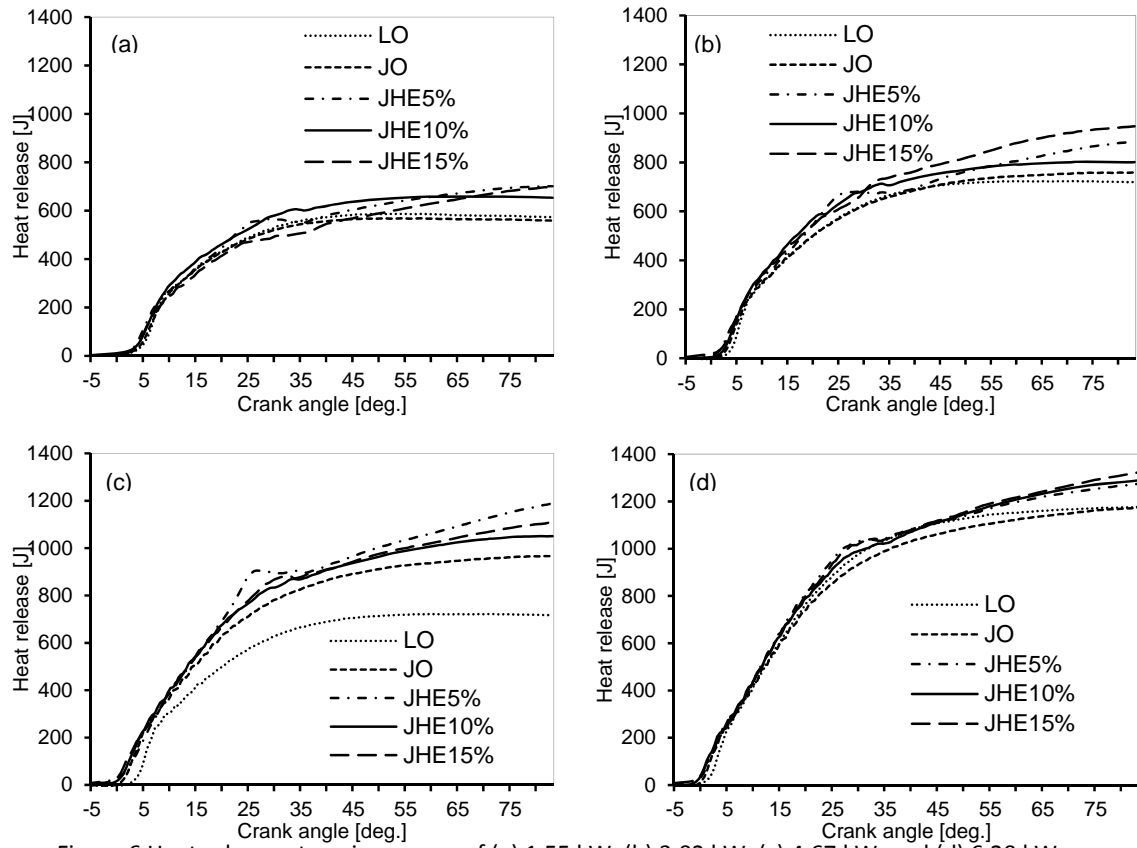


Figure 6 Heat release at engine power of (a) 1.55 kW, (b) 3.02 kW, (c) 4.67 kW, and (d) 6.20 kW

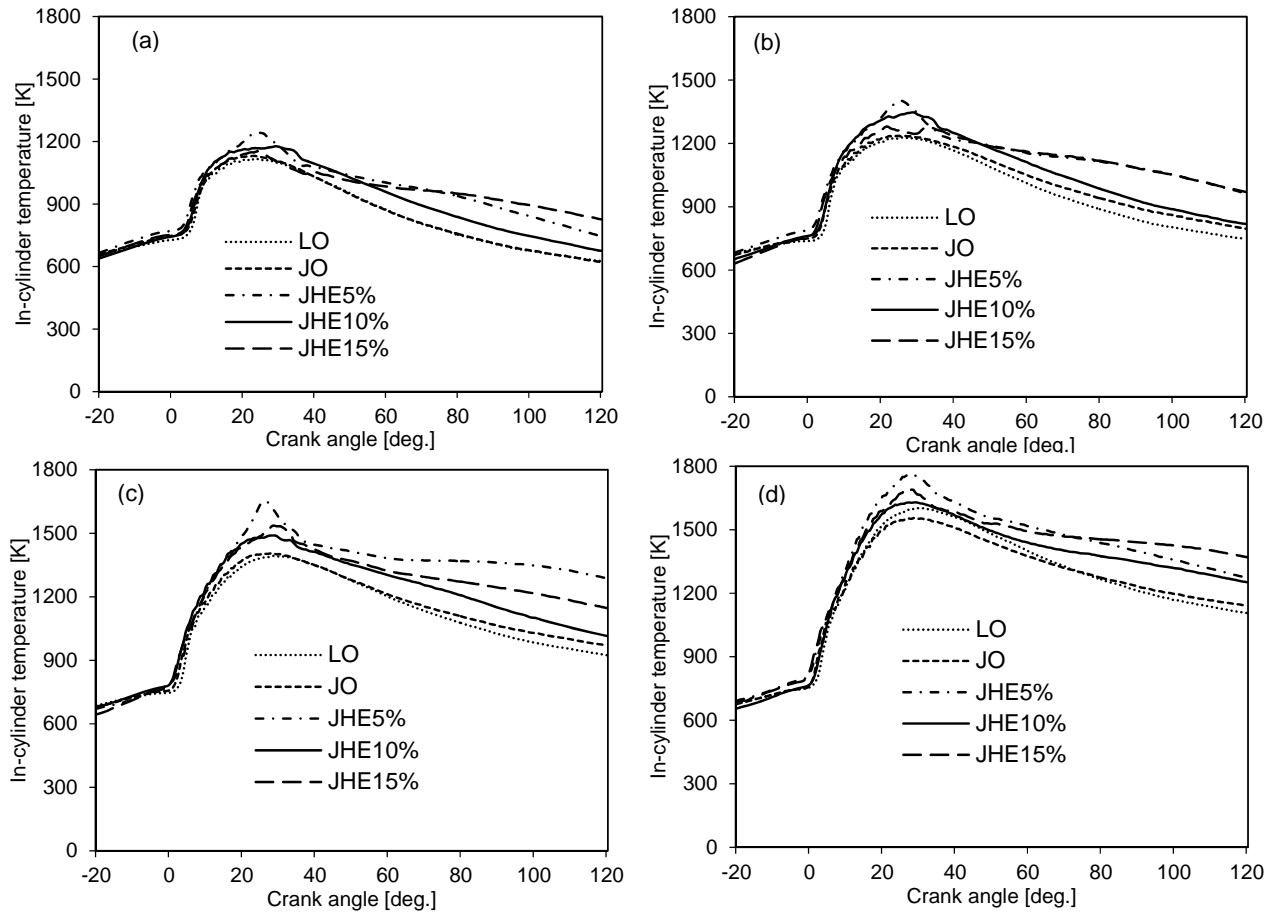


Figure 7 In-cylinder temperatures at power of (a) 1.55 kW, (b) 3.02 kW, (c) 4.67 kW, and (d) 6.20 kW

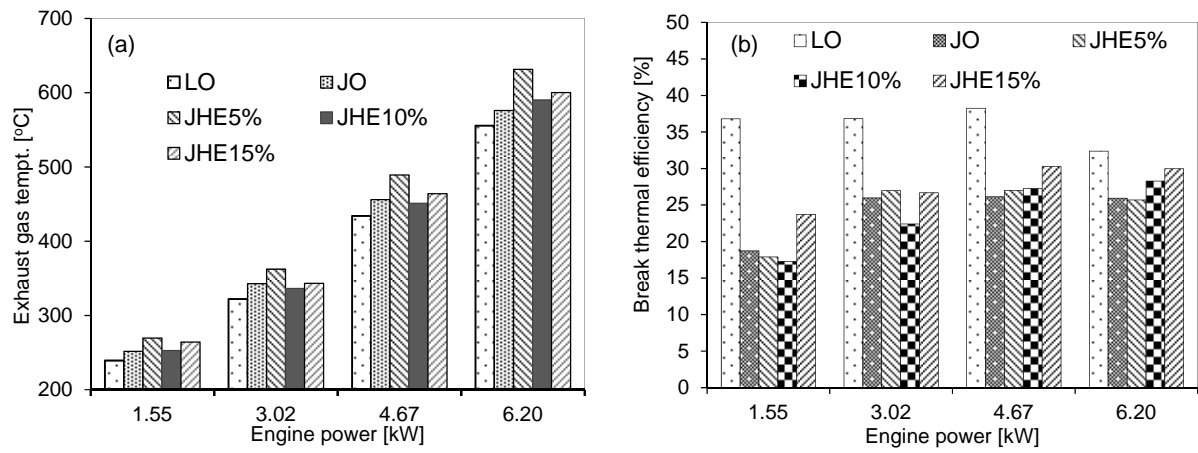


Figure 8 (a) Exhaust gas temperatures, and (b) break thermal efficiency at different engine powers

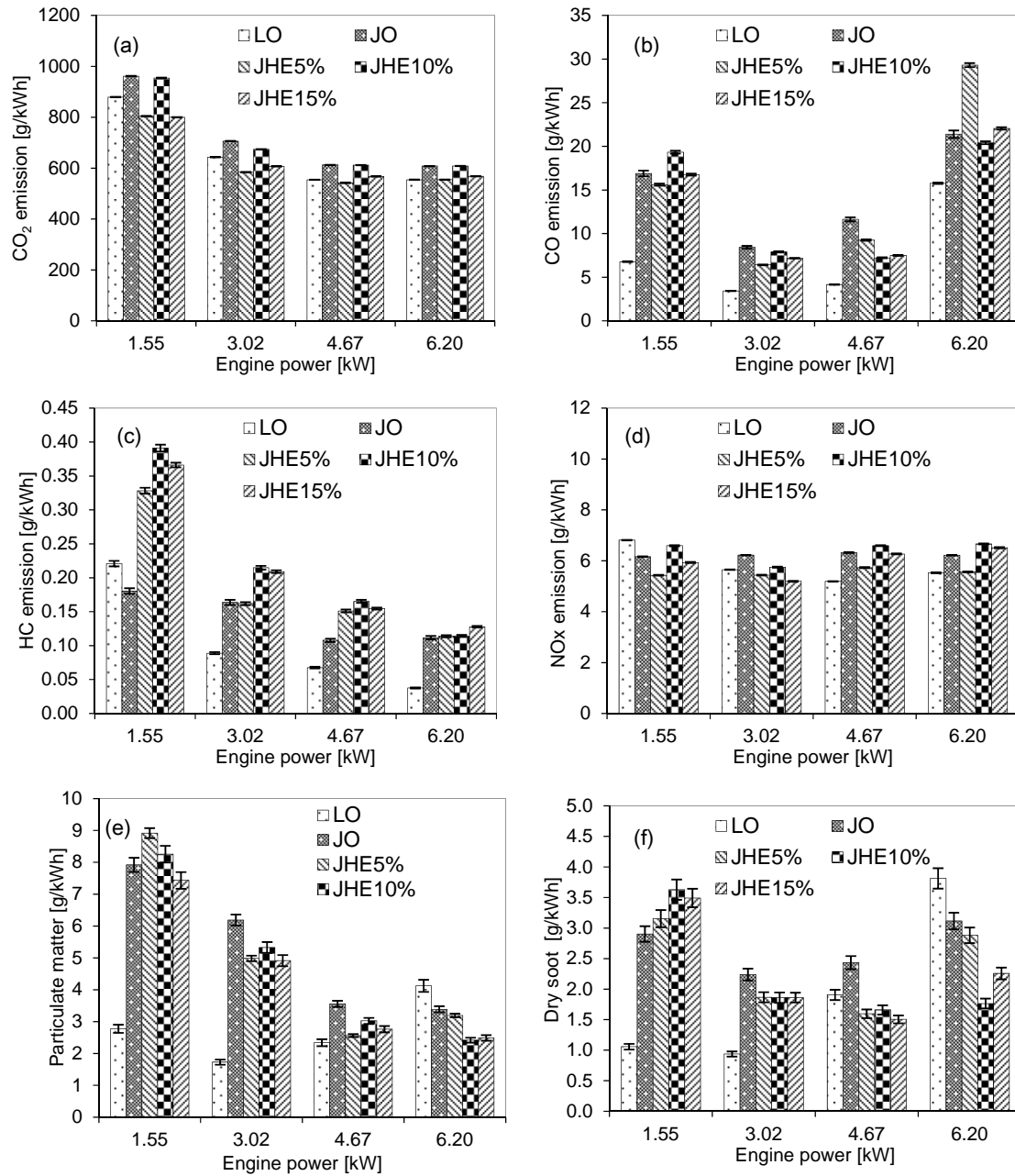


Figure 9 Exhaust gas emissions at different engine powers

LIST OF TABLES

Table 1 Properties of test fuels

Table 2 Specification of test engine

						Tabl
Parameter	Light oil	JO	JHE5%	JHE10%	JHE15%	e 1
Density (kg/m ³) at 30 deg. C	821.45	895.4	897.1	898.2	900.1	Prop
Heating value (kJ/kg)	42490	39774	-	-	-	ertie
Viscosity (cSt) at 30 deg. C	19.1	44.7	30	27.1	23.2	s of
Cetane number	45-55	40-45	-	-	-	
Flash point (°C)	50	240	-	-	-	test

fuels

Table 2 Specification of test engine

	Specification
Model	YANMAR NFD 13-ME
Engine type	Horizontal, 1-cylinder, 4-stroke engine
Combustion type	Direct injection
Bore × Stroke	92 × 96 mm
Displacement	0.638 liter
Compression ratio	17.7
Rated output	8.1 kW @ 2400 rpm
Injection nozzle	4-hole nozzle
Nozzle opening pressure	19MPa