Effect of Diesel–Palm Fatty Acid Distillate Ethyl Ester–Hydrous Ethanol Blend on the Performance, Emissions, and Long-Term Endurance Test on an Unmodified DI Diesel Engine

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ABSTRACT: In this research, the diesel–palm fatty acid distillate ethyl ester–hydrous ethanol, palm fatty acid distillate ethyl ester, and diesel were studied to investigate the gas emissions and performances of the direct injection diesel engine at different engine loads and engine speeds. At all engine speeds and loads, nitrogen oxide emissions from all fuel blends (D50PE40E10, D40PE50E10, and D30PE60E10) were significantly lower than the baseline diesel. At all engine speeds and engine loads, the fuel blends released less carbon dioxide than the baseline diesel, with the exception of the D30PE60E10 blend. Furthermore, D30PE60E10 diesel was used to test wear for 500 h long-term endurance of diesel engine components. The results indicated that biodiesel in fuel blends may reduce engine component wear by forming a thin coating on the metal surface of the engine component. However, after 100 h of continuous operation with D30PE60E10 blend, the engine cannot be restarted because only a part of the fuel pump had many pores on the surface of the plunger, barrel, delivery valve, and valve holder. However, these components may have to be considered to prevent corrosion when this fuel blend was employed.

1. INTRODUCTION

In several countries, petroleum fuels were implemented to advance the country in essential sectors such as industrial agriculture and transportation. Because the coronavirus disease 2019 spread quickly and extensively, many people at the community level were affected. However, rising petroleum consumption is expected in the future as increasing travel and transportation activities contribute to higher levels of air pollution. Due to the growing environmental concerns, biofuels including biodiesel and diesohol have emerged as an alternative liquid fuel for diesel engines as a type of liquid fuel. Because, the benefits of biodiesel include reduced environmental impact of NOx, CO, CO2, particulate matter (PM), and hydrocarbon emissions (HC). Furthermore, increasing the percentage of ethanol in diesel, also known as diesohol, has been demonstrated to lower NOx, CO, and PM. Oxygen in ethanol improves combustion efficiency, while also increasing the amount of oxygen in exhaust gases. However, ethanol’s solubility in the fuel blends is limited in several aspects. To solve these issues, emulsifiers have been suggested to avoid the phase separation of fuel blends. For diesel–biodiesel–ethanol fuel blends, the biodiesel was used as an emulsifier for fuel blends of diesel–ethanol. Their similar structure can improve the solubility and property of ethanol–diesel blends. In this study, palm fatty acid distillate (PFAD), a byproduct of crude palm oil physical refining, was used...
investigated as a renewable energy source for biofuel production. In an acid-catalyzed esterification, the pure PFAD from cooking oil factory will be converted to palm fatty acid distillate ethyl ester (PFADDE). It was added into fuel blend of diesel—ethanol to test in a diesel engine without major modifications. However, several researchers have mentioned that the main drawbacks of diesohol (diesel—ethanol) are the properties of low dynamic viscosity, low calorific value, and low cetane number, which are the causes of auto-ignition and ignition delay. As mentioned above, ethanol has a low solubility in diesel fuel because the possibility of phase separation of diesohol may occur quickly when a high ethanol concentration is employed. In order to increase fuel blend phase stability, they recommended using biodiesel as an emulsifier. Subbaiah et al. reported that the phase separation problem is caused by the chemical structures of ethanol and diesel. The fatty acid methyl ester or biodiesel was employed as an emulsifier to increase a solubility of ethanol in diesohol. Because biodiesel is a fuel with both lipophilic and hydrophilic components that are structurally comparable to ethanol and diesel. The use of biodiesel as an emulsifier in diesohol to improve phase stability is similar to that described by Chotwichien et al., Cheenkachorn et al., and Al-Hassan et al. Furthermore, the utilization of ethanol blended into the fuel blends for diesel engine testing was critical for this research. There have been many reports that hydrous ethanol (95 vol %) is not suggested for direct blending in diesel engines. However, the low purity of hydrous ethanol makes it less expensive than anhydrous ethanol (99.9 vol %). High energy consumption and production cost are required when the molecular sieve dehydration is used to remove residual water in hydrous ethanol. There have been instances of adding hydrous ethanol to a fuel blend, and these blends may be applied in diesel engines to improve emissions because of the high oxygenates in its composition. There are reports that over 15 wt % anhydrous ethanol in fuel blend is not recommended in diesel engines because high ethanol contents in fuel blends are the causes of ignition delay, auto-ignition, and low cetane number. According to the ignition delay for increasing ethanol in fuel blends, de Oliveira et al. have described the effects of S, 10, and 15 wt % of ethanol in biodiesel (B7). In their research, they found that adding the ethanol concentration in B7 caused the ignition delay (26.8°C) to increase. As a result, the period of the combustion duration was also reduced. Pidol et al. reported that 20% ethanol in fuel blend had poor auto-ignition characteristics compared to diesel. Srikanth et al. had also reported that the ignition delay occurred when over 15% ethanol content was blended in fuel blends due to a cetane number decrease.

For the performance of using ethanol in fuel blend, Al-Hassan et al. studied the performance of fuel blend of diesel—biodiesel—ethanol in a diesel engine. The results showed that the brake power ($P_b$) decreased when over 5% ethanol was used in a diesel engine. Moreover, the brake-specific fuel consumption (BSFC) of fuel blends at over 5% ethanol is higher than the diesel fuel due to the lower heating value (LHV) decrease. The brake thermal efficiency (BTE) increased in the range of 800 to 1400 rpm of engine speeds, whereas it decreased with 1600 rpm of highest engine speed. However, when the content of ethanol exceeds 10%, the BTE of fuel blends is slightly greater than diesel because the higher latent heat of ethanol vaporization has a quenching effect. Tan et al. studied the fuel blends of diesel—biodiesel—ethanol in the diesel engine using four types of fuel blends of 0, 5, 10, and 15 vol % ethanol compared to diesel. They reported that BSFCs were found to be 0.44, 0.45, 0.46, and 0.49 kg/kW h and BTEs were 41.8, 37.4, 24.7, and 20.0% for 0, 5, 10, and 15 vol % ethanol, respectively. Therefore, increasing ethanol percentage in the fuel blend resulted in a 28.95% increase in BSFC and a 61.39% decrease in BTE for 15% highest content of ethanol compared to that of diesel.

According to an investigation of diesel engine components for long-term endurance testing, Reddy et al. conducted a 250 h endurance testing of fuel injection equipment (FIE) on a diesel engine using 100% of Karanja oil, 100% of biodiesel from Karanja oil, and diesel. They found low wear of FIE for KOME100 when compared to diesel because biodiesel includes ester functional groups, such as $\text{COOCH}_3$, which creates a thin coating on engine components to reduce FIE wear. Kumar et al. studied the long-term testing with diesel and B40 in an unmodified diesel engine at 512 h. They stated that the lubricity of the B40 tested is appropriate for the decreased wear of self-lubricated engine components compared to diesel. The decreasing wear for the plunger and nozzle needle at different locations, while using B40 compared to diesel fuel, was in the range of 23–40% for the diameter of plunger and the wear of the nozzle needle varied in the range of 11.9–25%. Therefore, the observations of optical micrographs indicated that there was some corrosion. B40 can reduce the wear of engine components when compared to diesel. There are a few research studies on long-term endurance testing with hydrous ethanol in fuel blends. However, 95% purity of hydrous ethanol makes it inexpensive compared to high purity anhydrous ethanol, which is an interesting biofuel. Moreover, as mentioned above, very few research has been reported on diesel—biodiesel—hydrous ethanol under the long-term testing at a constant speed of engine. Moreover, the fuel blends have been extensively studied in short-run experiments to determine emissions and performance. However, there have been very few studies on analyzing various parts of diesel engines with long-term endurance testing using the fuel blends. Few studies on the influence of fuel blends on the emissions, performance, and wear of diesel engines have been reported in the literature such as those by Reddy et al. and Kumar et al. who studied the diesel—biodiesel fuel blends in diesel engines, and they concluded that using biofuel and fuel blends might lead to corrosion of delivery valve, fuel injector, barrel, plunger, nozzle needle, and valve holder after long-term engine running, and such complications of components in a diesel engine may not occur with short-range engine operations. Therefore, evaluating the diesel—PFADDE—ethanol fuel blend in a diesel engine is a fascinating method of determining the wear percentage of engine components for long-term endurance testing. In addition, there are few types of research on performances and emissions testing by using diesel—PFADDE—ethanol in DI diesel engine. For our previous paper of Thawornprasert et al., for the purposes of this research, only the stability and fuel characteristics of diesel—PFAD—ethanol, diesel—PFADDE—ethanol, and exhaust gas emission tests at different engine speeds and without engine load were examined. However, the emissions and performances of diesel—PFADDE—ethanol blends under different engine loads were not examined in our previous paper. Therefore, diesel—PFADDE—ethanol blends (D50PE40E10, D40PE50E10, and D30PE60E10 blends), the
gas emissions of NOₓ, CO, CO₂, and O₂, and performances of \( P_b \), BSFC, and BTE in the diesel engine at different speeds and load of engines using a dynamometer were examined. Moreover, D30PE60E10 fuel blend and diesel were compared to test the various parts of the engine for long-term endurance testing. The wear analysis of dimensions, weight losses, visual inspections for vital parts (piston, cylinder liner, piston rings, connecting rod bearing, inlet valve, and exhaust valve), and FIE (plunger, barrel, delivery valve, valve holder, nozzle needle, and fuel injector) was also completed. The results of this study will improve the clarity and reduce a few barriers to the adoption of diesel—biodiesel—ethanol fuel blends in diesel engines.

2. RESULTS AND DISCUSSION

2.1. Engine Performance. 2.1.1. Brake Power (\( P_b \)). Figure 1a shows the difference in brake power achieved with different fuels compared to the baseline of diesel at various engine speeds. \( P_b \) rises with the increasing engine speed until 2000 rpm for all fuels.

However, \( P_b \) of PFADE is rather lower than the diesel baseline at all engine speeds because the LHV of PFADE (40.2 MJ/kg) was close to diesel (43.7 MJ/kg). In addition, higher density and viscosity of PFADE than diesel may result in lower combustion efficiency. 35 For all fuel blends, \( P_b \) decreased from 1100 to 2000 rpm speed of the engine compared to of baseline diesel. Moreover, the \( P_b \) results of all fuel blends were sharply decreased from 1700 to 2000 rpm speed of the engine. Unfortunately, there are no results of \( P_b \) at 2300 rpm maximum engine speed. When more than 15% ethanol was added to the fuel blend, the engine cannot run smoothly due to an ignition problem. Containing the high ethanol amount in fuel blend can cause ignition delay, auto-ignition, and low cetane number. 2,25 A related result was reported by Tripathi et al.,33 they recommended that the cetane number of ethanol in fuel blend should be in the range between 5 and 8. When 10 vol % of ethanol was added to the fuel blend, the cetane number of fuel blends was reduced to 7. It can be noted that adding ethanol into the fuel blend can cause increase of the auto-ignition temperature in the combustion chamber, resulting in a high vaporization effect on the fuel blend. This is the reason for longer ignition delays. 16 It can have an effect on the decreasing engine power, resulting in incomplete combustion in the cylinder. As a result, the engine could not be operated smoothly due to an ignition malfunction, especially when the engine was run at high speed with 10 wt % ethanol-blended fuel. 34 In this experiment, the engine cannot be run smoothly when the engine load was increased to 75% at 2300 rpm. Therefore, there have been no reports of the performance and gas emissions of D50PE40E10, D40PE50E10, and D30PE60E10 blends at 75% engine load at 2300 rpm. Consequently, at the maximum available speed of 2000 rpm, it was observed that 4.07, 87.19, 67.81, and 79.58% of \( P_b \) could be generated from D50PE40E10, D40PE50E10, and D30PE60E10, respectively, which are lower compared to the baseline of diesel.

2.1.2. Brake-Specific Fuel Consumption. The BSFC is the measure of the combustion efficiency of an engine that consumes fuel and generates rotational power. The BSFC shows how effectively the engine converts fuel into work in an hour to generate 1 kW brake power. 29,35 In order to better understand the performance characteristics of biodiesel and diesohol, it is necessary to vary the various engine speeds and engine loads, as shown in Figure 1b. The calorific value is one of the primary factors used to evaluate the BSFC. A decreasing calorific value of fuels has an impact on the fuel consumption required to achieve the power output. For all cases, the BSFC for both PFADE and diesohol is higher than the baseline of diesel; the high BSFCs of these fuel blends were found at 25% engine load and increased when the engine was increased to 50% engine load. The D30PE60E10 blend showed the highest BSFC because the proportion of biodiesel in the fuel blend was very high, and its fuel property had a low calorific value. The BSFC results of all diesohol fuels are greater than diesel results in all conditions because ethanol has a lower calorific value than diesel, and the BSFC of diesohol is also higher than PFADE results in all conditions. Moreover, BSFC results increased when biodiesel contents were increased in the fuel blends because their calorific value had a lower heating value than diesel. Furthermore, BSFC results from PFADE increased when the proportion of biodiesel in the fuel blends was increased. Biodiesel had a lower heating value than diesel. Srikanth et al. 35 observed similar results and reported that BSFC of diesel—biodiesel—ethanol blends increased with high ethanol and biodiesel proportion in fuel blends at all engine loads. The LHV of biodiesel and ethanol was lower than the LHV of diesel. Therefore, more fuel consumption was required when ethanol and biodiesel were used to blend in fuel blends.
At 25% engine load, BSFC results at 1700 rpm maximum engine speed were 6.01, 19.19, 21.85, and 27.83% for PFADE, D50PE40E10, D40PE50E10, and D30PE60E10, respectively, when compared to the baseline of diesel’s BSFC. At 50% engine load, there are no results of fuel blend engine speed of 1700 rpm. Therefore, the maximum available engine speed of 1400 rpm of PFADE and all fuel blends were used to compare the BSFC. The results were 9.32%, 19.19%, 21.39%, and 27.45%. The BSFCs were found from PFADE, D50PE40E10, D40PE50E10, and D30PE60E10 blends compared to baseline of diesel. At 75% maximum engine load, only the effect of PFADE was observed at different loads. The BSFCs were found to be 8.71, 9.80, 11.74, 11.55, and 11.99% at 1100, 1400, 1700, 2000, and 2300 rpm when compared to the baseline of diesel’s BSFC (kg/kW h).

2.1.3. Brake Thermal Efficiency. The BTE is the efficiency with which an engine can convert the heat received from the fuel into work. Figure 1c shows the variation of BTE obtained with different fuels compared to the baseline of diesel at different engine speeds and load conditions. BTEs for all PFADE at engine speeds were greater than diesel at 25% engine load. However, BTEs of PFADE at 50 and 75% decreased less than diesel, which is similar to the results from other fuel blends. Because PFADE contains more oxygen than diesel, it has a superior heat efficiency at low engine load. Because of its higher viscosity, PFADE has a poorer thermal efficiency than diesel when observed at high loads of engine. Setiyo et al.36 described similar results, explaining that biodiesel has a greater content of oxygen than diesel, enhancing combustion efficiency and increasing BTE. Although biodiesel has a greater viscosity than diesel, it has a lower thermal efficiency due to insufficient fuel atomization. The LHV of biodiesel explains its poorer thermal efficiency. At 1700 rpm, BTE of PFADE was 2.54% at 25% engine load, which is higher when compared to the baseline of diesel. However, when the proportion of biodiesel in the dieselohol blends increased, the BTE decreased significantly due to the decreasing LHV of both biodiesel and ethanol. Thus, increasing the proportion of biodiesel and ethanol in the fuel blend decreased the BTE. Their LHV resulted in poorer combustion than diesel due to their LHV.16,29 Similar results are described by Al-Hassan et al.16 and Silitonga et al.,28 they...
found that increasing the latent heat of ethanol vaporization increased heat loss, which resulted in a decrease in the BTE. Furthermore, the lower cetane value of ethanol causes a longer ignition delay, resulting in incomplete combustion and a lower BTE. When the engine was running at 50% engine load, the lowest BTEs of PFADE, D50PE40E10, D40PE50E10, and D30PE60E10 were lower than the diesel BTEs of 4.56, 2.48, 2.22, and 2.91% at 1700, 1400, 1400, and 1400 rpm, respectively. For the engine load of 75%, there were only the BTE results from PFADE; the lowest BTE of PFADE was less than that of diesel by approximately 2.72% at 1700 rpm engine speed.

2.2. Engine Emission. 2.2.1. Exhaust gas Temperature. Figure 2a shows the exhaust gas temperature (EGT) of diesel, PFADE, and diesel–PFADE–ethanol at the conditions of different engine loads and engine speeds. In all cases, EGT was high as the load and speed of engines increased. The EGT from PFADE and all fuel blends were close to diesel at a 25% engine load by engine speeds ranging from 1100 to 1700 rpm. The difference of EGT was less than 5 °C when compared to the baseline of diesel at speeds from 1100 to 1700 rpm. For the low speed of engine, the EGT of PFADE is greater than diesel at the engine speeds of 1100 rpm at 50 and 75% engine load. Because biodiesel has a longer ignition delay and combustion duration than diesel, there is insufficient heat evaporation inside the cylinder at low engine speed. Similar results are obtained by Chuah et al.; they found that the ignition delay and combustion range of biodiesel were both longer than those of diesel. This is due to biodiesel’s higher boiling point, which prevents water from evaporating during primary combustion and causes more heat to be released during late combustion. For the engine speed ranges (1400–2300 rpm), the EGT of PFADE was lower than diesel because the LHV value of PFADE was lower than that of diesel. Similar results were reported by Tan et al.; they reported that the EGT of biodiesel was lower compared to that of diesel because the cetane number and LHV of these fuel blends are lower than those of diesel. Furthermore, ethanol had a lower cetane number, which caused the combustion rate in the cylinder to slow and resulted in a low EGT. At 50 and 75% engine loads, the EGTs of PFADE and dieselohol were mostly lower in the range of EGT from 5 to 18 °C when compared to the baseline of diesel. Almost EGT from PFADE fuel, it clearly 25, 50, and 75% engine loads, NOx emissions of PFADE were found to be greater than the baseline of diesel for all speeds and engine loads except for 1100 rpm at 25% engine load. When the NOx emissions of PFADE at 1400 rpm engine speed were compared to the baseline of diesel, 62.64, 53.09, and 81.52% were emitted at 25, 50, and 75% of engine load, respectively. According to the results of Agarwal28 and Chuah et al., when temperatures were high enough, atmospheric nitrogen oxidation led to NOx emissions. The diesel engine produced greater NOx emissions due to the increased combustion chamber temperature induced by complete burning of the biodiesel high oxygen content. Comparing the 25 and 50% engine loads of diesel at 1100 rpm to the baseline, the NOx emissions of D50PE40E10, D40PE50E10, and D30PE60E10 decreased by 37.29, 50.91, and 15.74% for the 25% engine load and by 31.21, 30.29, and 11.56% for the 50% engine load. Due to its ability to reduce NOx emissions, dieselohol may have a positive effect on the environment. Lower NOx emissions are achieved because the ethanol evaporates quickly in the combustion chamber, lowering the temperature of the exhaust gases. Furthermore, the decreased calorific value of ethanol lowers the combustion temperature. These main benefits of incorporating ethanol into fuel blends are comparable to those found by Nour et al., Farias et al., and Huang et al.

2.2.2. CO and CO2 Emissions. Carbon monoxide (CO) is a colorless, toxic gas formed when carbon in fuel is incompletely burned. Figure 2c shows CO emissions of all fuels at various conditions. When comparing 25, 50, and 75% engine loads of PFADE to the baseline of diesel, the CO emissions of PFADE increased by 21.90% for 25% engine load and 31.52% for 50% engine load and decreased by 63.91% for 75% engine load at 1100 rpm. For a 25% engine load, CO emissions of PFADE increased by 1.26% at 1400 rpm and decreased by 34.94% at that speed for a 50% engine load and 19.26% for a 75% engine load. At 1100 rpm, the CO emissions of D50PE40E10, D40PE50E10, and D30PE60E10 increased by 22.39, 5.99, and 58.03% compared to the baseline of diesel at 25% engine load. At 1400 rpm engine speed, CO emissions of D40PE50E10 and D30PE60E10 increased by 11.95% of 25% engine load and 31.52% for 50% engine load and decreased by 63.91% for 75% engine load at 1100 rpm. For a 25% engine load, CO emissions of PFADE increased by 1.26% at 1400 rpm and decreased by 34.94% at that speed for a 50% engine load and 19.26% for a 75% engine load. At 1100 rpm, the CO emissions of D50PE40E10, D40PE50E10, and D30PE60E10 increased by 22.39, 5.99, and 58.03% compared to the baseline of diesel at 25% engine load. At 1400 rpm engine speed, CO emissions of D40PE50E10 and D30PE60E10 increased by 11.95% of 25% engine load. When comparing 50% engine load of fuel blends to the baseline of diesel, the CO emissions of D50PE40E10, D40PE50E10, and D30PE60E10 increased by 22.39, 5.99, and 58.03% compared to the baseline of diesel at 25% engine load. At 1400 rpm engine speed, CO emissions of D40PE50E10 and D30PE60E10 increased by 11.95% of 25% engine load. When comparing 50% engine load of fuel blends to the baseline of diesel, the CO emissions of D50PE40E10, D40PE50E10, and D30PE60E10 increased by 22.39, 5.99, and 58.03% compared to the baseline of diesel at 25% engine load. At 1400 rpm engine speed, CO emissions of D40PE50E10 and D30PE60E10 increased by 11.95% of 25% engine load. When comparing 50% engine load of fuel blends to the baseline of diesel, the CO emissions of D50PE40E10, D40PE50E10, and D30PE60E10 increased by 22.39, 5.99, and 58.03% compared to the baseline of diesel at 25% engine load. At 1400 rpm engine speed, CO emissions of D40PE50E10 and D30PE60E10 increased by 11.95% of 25% engine load. When comparing 50% engine load of fuel blends to the baseline of diesel, the CO emissions of D50PE40E10, D40PE50E10, and D30PE60E10 increased by 22.39, 5.99, and 58.03% compared to the baseline of diesel at 25% engine load. At 1400 rpm engine speed, CO emissions of D40PE50E10 and D30PE60E10 increased by 11.95% of 25% engine load. When comparing 50% engine load of fuel blends to the baseline of diesel, the CO emissions of D50PE40E10, D40PE50E10, and D30PE60E10 increased by 22.39, 5.99, and 58.03% compared to the baseline of diesel at 25% engine load. At 1400 rpm engine speed, CO emissions of D40PE50E10 and D30PE60E10 increased by 11.95% of 25% engine load. When comparing 50% engine load of fuel blends to the baseline of diesel, the CO emissions of D50PE40E10, D40PE50E10, and D30PE60E10 increased by 22.39, 5.99, and 58.03% compared to the baseline of diesel at 25% engine load. At 1400 rpm engine speed, CO emissions of D40PE50E10 and D30PE60E10 increased by 11.95% of 25% engine load.
be delayed, resulting in increased CO emissions. The direct fuel spray caused incomplete combustion due to the increased pre-mixed combustion rates in the cylinder. Furthermore, the decreased cetane number caused by the addition of alcohol to the fuel blend causes an increase in the ignition delay, which leads to incomplete combustion and an increase in CO emissions.\textsuperscript{23} It could also indicate complete combustion efficiency in an engine in terms of CO\textsubscript{2} emissions.\textsuperscript{21,32} Figure 2d shows the CO\textsubscript{2} emissions from combustions from diesel, PFADE, and diesohol. The CO\textsubscript{2} emissions of PFADE were found to be greater than the baseline of diesel for 25, 50, and 75\% of engine loads, for all speeds and engine loads, respectively. The CO\textsubscript{2} emissions of PFADE at 2300 rpm engine speed were higher than the baseline of diesel by 8.21, 47.86, and 56.67\% at 25, 50, and 75\% of engine load, respectively. Comparing 25 and 50\% engine load of fuel to the baseline of diesel at 1100 rpm, the CO\textsubscript{2} emissions of D50PE40E10 and D40PE50E10 decreased by 3.97 and 18.25\% for 25\% engine load and by 9.02 and 11.39\% for 50\% engine load. In contrast, the D30PE60E10 increased by 3.36\% for a 25\% engine load and 5.79\% for a 50\% engine load. The CO\textsubscript{2} emissions were higher when the proportion of biodiesel blended into the fuel increased. Chuah et al.\textsuperscript{32} investigated the emissions of biodiesel–diesel blends at various percentages in comparison to diesel. They found that biodiesel blends emitted more CO\textsubscript{2} than diesel and that CO\textsubscript{2} levels increased as the content of the biodiesel-to-diesel blend increased. Because the oxygen in the exhaust gases indicates that the fuel was not completely burnt. The low oxygen concentration of the exhaust gas, on the other hand, shows complete combustion because the O\textsubscript{2} was utilized to convert CO to CO\textsubscript{2} during the combustion process.\textsuperscript{32,32} The amount of extra oxygen in the exhaust gas reduced as the engine load increased.\textsuperscript{21} Figure 2e shows the O\textsubscript{2} gas in the exhaust gas, with the O\textsubscript{2} content from the exhaust of PFADE decreasing as engine load increases. For all speeds and engine loads, the O\textsubscript{2} gas of PFADE was found to be lower than the baseline of diesel for 25, 50, and 75\% engine loads. The O\textsubscript{2} gas of PFADE at 2300 rpm engine speed was lower than the baseline of diesel by 2.83, 22.09, and 46.05\% at 25, 50, and 75\% of engine load, respectively. At 25 and 50\% engine loads, the O\textsubscript{2} gas of fuel blends was higher than the baseline of diesel for all speeds except for D30PE60E10 blends. With 25\% engine load, the O\textsubscript{2} gas of D50PE40E10 and D40PE50E10 increased by 2.31 and 3.95\%, respectively, while that of D30PE60E10 decreased by 2.92\% at 1700 rpm engine speed. At 50\% engine load, O\textsubscript{2} gas of D50PE40E10 and D40PE50E10 increased by 7.53 and 4.81\%, respectively, while that of D30PE60E10 decreased by 0.73\% at 1400 rpm engine speed, when compared to the baseline of diesel.

2.3. Wear Analysis. 2.3.1. Dimensional Loss. The effect due to the use of diesel–PFADE–hydrous ethanol blend on various parts of the diesel engine was studied. For the endurance testing, the D30PE60E10 blend was chosen for a longer period of test (500 h) at 2300 rpm maximum speed of engine. Using the highest proportion of biodiesel (60\%) in diesohol is an interesting alternative for wear test of diesel engines. The dimensional loss analysis was performed on vital components (piston, cylinder liner, piston rings, connecting rod bearing, inlet valve, and exhaust valve) before and after the long-term endurance test. The dimensional loss of vital parts was measured at different locations, as shown in Figure 3. The piston’s diameters of both the X- and Y-axes were measured at locations a and b using a micrometer, as shown in Figure 3a.

Figure 3. Dimensional loss measurements of vital parts in diesel engine (a) piston, (b) cylinder liner, (c) piston rings, (d) connecting rod bearing, and (e) inlet and exhaust valves.
The location a of the piston and the dimensional losses of the D30PE60E10 blend were 43.75% in the X-axis and 23.33% in the Y-axis, higher than those of diesel in the X- and Y-axes, respectively. The moving component that assembles with the piston inside is the cylinder liner. In Figure 3b, the cylinder liner’s inner diameters of both the X- and Y-axes were measured at locations a, b, c, d, and e using a dial bore gauge, as shown in Figure 3b. The dimensional loss in the fifth location of diesel of 80.07 and 20.03% was higher wear than the D30PE60E10 blend in the X- and Y-axes, respectively. A critical component of combustion is the piston rings, which are required for sealing the gap between the cylinder liner wall and the piston. The Kubota ZT100 DI diesel engine had four-piston rings: compression ring no. 1, compression ring no. 2, wiper ring, and oil ring. The compression ring no. 1 is the primary compression ring, and the wiper ring is the secondary compression ring. In Figure 3c, the measurements were taken of the piston rings at three different locations of a, s, and h with a vernier caliper. All piston rings of the D30PE60E10 blend have higher dimensional loss than diesel. In terms of dimensional losses of all piston rings, these parts from the D30PE60E10 blend showed higher wear dimensional loss than diesel. When comparing the D30PE60E10 blend to diesel, the maximum loss of compression ring no. 1, compression ring no. 2, wiper ring, and oil ring occurred at s-location by 13.13, 8.15, 12.95, and 4.89%, respectively. The connecting rod bearing diameters of both C and D were measured at locations A and B using a vernier caliper, as shown in Figure 3d. The dimensional loss of the D30PE60E10 blend was higher wear than diesel; it was 28.81% for C-diameter and 9.90% for D-diameter at location A and 73.88% for C-diameter and 70.72% for D-diameter at location B. Finally, the valves were inspected to have dimensional loss at length L and diameter D by using a vernier caliper and a micrometer, respectively, as shown in Figure 3e. The dimensional losses of the inlet valve and exhaust valve after running with the D30PE60E10 blend were higher wear than diesel by 24.00 and 40.00% for L-length and 12.50 and 18.52% for D-diameter, respectively. The engines running on the D30PE60E10 blend have been shown to have minimal wear on vital parts. The surfaces of metal parts are less likely to corrode when using biodiesel-blended fuel. Because the biodiesel created a protective thin film on the metal surface of the engine components, it may assist in minimizing wear on the components. According to the engine’s specification, the wear on all vital parts was within the acceptable wear range.

2.3.2. Weight Loss. Before and after the long-term endurance test, the weight of critical components was examined. As shown in Figure 4, weight loss analysis of vital parts revealed that the wear on vital parts of the engines caused by operating with diesel was higher than that caused by running on the D30PE60E10 blend, except for that on the piston. Because the biodiesels included polar functional groups in their molecules, they create a thin film coating on engine components, reducing friction and wear. After operating with diesel, the compression ring no. 1, compression ring no. 2, and wiper ring had the greatest wear of 0.76, 0.34, and 0.29%, respectively. Furthermore, minor wear was on the exhaust valve, inlet valve, oil ring, cylinder liner, and connecting rod bearing. After a 100 h of continuous running time, the diesel engine was stopped after continuous operation to change the engine oil and oil filter. However, the diesel engine could not be restarted to test for the following run because it was found that there was an issue with the engine while utilizing the D30PE60E10 blend. However, restarting the diesel engine with diesel was not an issue. Therefore, all FIE components (delivery valve, fuel injector, barrel, plunger, nozzle needle, and valve holder) were rechecked to solve this problem for testing of the D30PE60E10 blend. After inspection of all FIE components, it was found that the FIE components were corroded, resulting in the inability to inject fuel into the combustion chamber. Therefore, new FIE components will be changed for testing in the next round every 100 h for the D30PE60E10 blend. The results of weight losses of FIE components for the D30PE60E10 blend after every 100 h running time were higher than those of diesel at 500 h except for the nozzle needle. The delivery valve, barrel, valve holder, plunger, and fuel injector after running with the D30PE60E10 blend showed the highest wear by 3.44, 1.10, 0.60, 0.47, and 0.09%, respectively. Lubricity was a significant fuel characteristic that was responsible for lubricating and reducing system wear. However, the viscosity of the fuel decreased as the concentration of ethanol was increased in the blend. Because of the high concentrations of hydrous ethanol and organic acids in the fuel, the contact surfaces of the fuel injector and injection pump rapidly corroded. Moreover, the presence of oxygen in the oil increases the probability of oil oxidation and corrosion of engine components.44 Related results were reported by Thangavelu et al.45 They mentioned that the oxygenated fuel consisted of water, organic acids, and esters, all of which contributed to the corrosion of fuel system components. The fuel blends had an impact on diesel engine parts composed of mild steel and aluminum, especially the fuel pump.

2.3.3. Visual Inspection Analysis. The wear was determined by visual inspections of vital parts such as the piston, cylinder liner, piston rings, connecting rod bearing, inlet valve, and exhaust valve and also FIE components such as delivery valve, fuel injector, barrel, plunger, nozzle needle, and valve holder at various positions of the D30PE60E10 blend and diesel fuels. Comparison the wear of vital and FIE parts in diesel and the D30PE60E10 blend using a diesel engine is shown in Tables 1 and 2. The carbon deposits on the piston surface of the fuel blend of the D30PE60E10 blend are observed, and their carbons are higher than diesel, as shown in Table 1. It could be the result of heat oxidation and incomplete combustion. The surface of the cylinder liner for the D30PE60E10 blends
converted to light brown, which may be attributed to the heat of combustion influencing the color of the cylinder’s surface. There were no major modifications to the piston rings and connecting rod bearing parts when the D30PE60E10 blend and diesel were used for an extended period of time. For part of the valve, the exhaust valve of the D30PE60E10 blend appears to have smoother surfaces than diesel. The thin film from the biodiesel blend appeared to protect the metal surface better than diesel.30,31 Table 2 shows the FIE components for the D30PE60E10 blend and diesel after a long-term endurance test. The wear analysis of FIE components was carried out using the visual inspection method. The results showed that the FIE components from running with the D30PE60E10 blend show higher wear than diesel. For the D30PE60E10 blend, the plunger and the barrel were so severely damaged that the engine was unable to run after continuous running for 100 h to change the engine oil. Because of the friction between the plunger and the barrel, the surface around the plunger seems scratched around the rims. As seen in Figure 5, corrosion inside and at the bottom of the barrel manifested itself as small round holes on the inside and at the bottom. In the case of the delivery valve and valve holder in Figure 5, the wear on these moving parts was higher than the wear on the same parts while operating on diesel. The parts had many pores on the delivery valve’s surface, as well as pores in the valve holder’s inside and bottom, as shown in Figure 5. After an endurance test, the high volatility of ethanol might have caused some pores on the surfaces of some components. Consequently, the engine cannot restart after it has been stopped, and the cavitation bubble of ethanol collapsed by the cavitation effect occurs inside the fuel pump. Therefore, increased wear and restart issues are possible side effects of fuel pump and injector cavitation.47 Similar results are obtained by Waterland et al.;48 they concluded that increasing ethanol vapor pressure may induce fuel pump vapor lock, which increases fuel pump wear. Moreover, Mickevičius et al.49 found that using low viscosity ethanol fuel might cause wear on the plunger and nozzle needle surfaces. The wear of the plunger and barrel may affect the pump component’s tightness and

Table 1. Visual Inspection of Vital Parts in a Diesel Engine

| Component          | D30PE60E10 | Diesel |
|--------------------|------------|--------|
| Piston             |            |        |
| Cylinder liner     |            |        |
| Piston ring        |            |        |
| Compression ring No.1 |       |        |
| Compression ring No.2 |       |        |
| Wiper ring         |            |        |
| Oil ring           |            |        |
| Connecting rod bearing |      |        |
| Inlet valve        |            |        |
| Exhaust valve      |            |        |

Table 2. Visual Inspection of FIE Components in a Diesel Engine

| Parts             | D30PE60E10 | Diesel |
|-------------------|------------|--------|
| Plunger           |            |        |
| Barrel            |            |        |
| Delivery valve    |            |        |
| Valve holder      |            |        |
| Nozzle needle     |            |        |
| Fuel injector     |            |        |
lubrication. The pump component will no longer function if its tightness is lost. Figure 6 shows the fuel spray characteristics of the fuel injector after running with (a) D30PE60E10 blend and (b) diesel.

Figure 5. Corrosion on (a) plunger, (b) barrel, (c) delivery valve, and (d) valve holder for the D30PE60E10 blend.

Figure 6. Fuel spray characteristics of the fuel injector after running with (a) D30PE60E10 blend and (b) diesel.

The fuel injector and nozzle needle seem to be the least worn components based on visual inspection. However, the fuel injector after running with D30PE60E10 and diesel. Such undesirable spraying characteristics may be caused by the adhesion of unburned substances on the fuel injector parts.

3. MATERIALS AND METHODS

3.1. Materials and Selection of Diesel–PFADE–Ethanol. The details of the fuel blend of diesel–PFADE–hydrous ethanol in terms of phase stability were described in our previous paper. Thawornprasert et al. As mentioned above, the density and viscosity parameters of 17 ternary blends (14 white and 3 black dots in Figure 7) were determined. Both D20PE70E10 and D10PE80E10 blends had viscosities of 4.17 and 4.13 cSt, respectively. As a result, these fuel blends did not meet the specifications for diesel fuel, which required a viscosity of 1.8–4.1 cSt at a temperature of 40 °C. Thus, the remaining 15 ternary blends (12 white and 3 black dots) were considered for testing in diesel engines. However, fuel blends containing more than 15% ethanol should not be employed in a DI diesel engine without modification for the reasons mentioned above. Thus, the remaining three ternary blends of diesel–PFADE–ethanol were D50PE40E10, D40PE50E10, and D30PE60E10, which were examined in terms of emissions of NOx, CO, CO2, and O2 and performances of BTE, BSFC, and FIE (plunger, barrel, delivery valve, valve holder, nozzle needle, and PFADE fuels were produced from the PFAD with the acid-catalyzed esterification process, which was used to blend the diesel–PFADE–ethanol fuel blends. Table 3 summarizes the properties of diesel, PFADE, and hydrous ethanol. The phase behavior of diesel–PFADE–ethanol blends, as shown in Figure 7, was defined by considering the condition of fuel blend. In the three-component phase diagram, the clear liquid single phase included 10–30 wt % of diesel, 30–80 wt % of PFADE, and 10–60 wt % of hydrous ethanol, and there was no phase separation after 90 days. Observing the phase stability of the fuel blends at temperature of 35 °C for 2 days resulted in the formation of clear liquid two phases with a PFADE of less than 30 wt %.

Table 3. Properties of Diesel, PFADE, and Hydrous Ethanol

| property                  | method          | diesel13 | PFADE13 | hydrous ethanol13 |
|---------------------------|-----------------|----------|---------|-------------------|
| higher heating value (MJ/kg) | CHNS/O          | 46.3     | 42.8    | 23.4              |
| LHV (MJ/kg)               | CHNS/O          | 43.7     | 40.2    |                   |
| cloud point (°C)          | ASTM-D 2500     | 14       |         |                   |
| pour point (°C)           | ASTM D 97       | <10°a    | 18      |                   |
| flash point (°C)          | ASTM D 93       | >52°a    | 175     | 18.5              |
| boiling point (°C)        |                 |          | 78.15   |                   |
| density at 30 °C (kg/L)   | ASTM D 1298     | 0.830    | 0.864   | 0.800             |
| viscosity at 40 °C (cSt)  | ASTM D 445      | 3.37     | 5.76    | 1.16              |
| water content (wt %)      | EN ISO 12937    | <0.05    | 0.237   | 5.0 vol %         |
| copper strip corrosion    | ASTM D 130      | <no. 1   | no. 1a  |                   |
| ester (wt %)              | EN 14103        | 86.12    |         |                   |
| triglyceride (wt %)       | EN 14105        | 1.12     |         |                   |
| diglyceride (wt %)        | EN 14105        | 2.29     |         |                   |
| monoglyceride (wt %)      | EN 14105        | 1.00     |         |                   |
| free fatty acid (wt %)    | EN 14105        | 9.47     |         |                   |
| acid value (mg KOH/g)     | ASTM D 664      | 31.54    |         |                   |
| copper strip corrosion    | ASTM D 1130     | no. 1    | no. 1a  |                   |
| cetane number             | ASTM D 613      | >50°a    | 65.4    |                   |

*Refer to Department of Energy Business.*
fuel injector); these components were compared with standard diesel.

3.2. Measurement of Diesel Engine Performance and Emission. The tests were carried out in a four-stroke, single-cylinder, water-cooled compression ignition engine (model: Kubota ZT100 DI). Figure 8 shows the direct injection diesel engine test rig and the eddy current brake dynamometer (model: DW16, Jiangsu Lan Ling Test Equipment Co., Ltd.). A compression ratio of 18:1, maximum power of 7350 W, engine speed of 2400 rpm, and air-standard diesel cycles were
Table 5. Range of Measurements, Accuracy of Devices, and Percentage of Uncertainties

| Measured Variable                  | range of measurement | device                  | accuracy | % uncertainty |
|------------------------------------|----------------------|-------------------------|----------|---------------|
| CO (ppm)                           | 0–10 000 ppm         | gas analyzer            | ±5%      | ±0.01         |
| CO₂ (vol %)                        | 0–25 vol %           | gas analyzer            | ±0.8%    | ±0.1          |
| NOx (ppm)                          | 0–3000 ppm           | gas analyzer            | ±5%      | ±0.01         |
| O₂ (vol %)                         | 0–25 vol %           | gas analyzer            | ±0.8%    | ±0.06         |
| EGT (°C)                           | 0–1000 °C            | temperature sensor      | ±2.6%    | ±0.2          |
| speed (rpm)                        | 0–6000 rpm           | encoder                 | ±10 rpm  | ±0.2          |
| weight (g)                         | 0–300 g              | digital balancing scale | ±0.02 g  | ±0.01         |
| time (s)                           |                      | digital stop watch      | ±0.1 s   | ±0.1          |
| temperature of fuel (°C)           | −250 to 1300 °C      | temperature sensor      | ±2.6 °C  | ±0.2          |
| load (Nm)                          | 0–70 Nm              | strain gauge load cell  | ±0.1%    | ±0.01         |
| LHV (kJ/kg)                        |                      | CHNS/O analyzer         | ±1.3     |               |

Calculated Parameter

- brake power (W) ±0.21
- fuel consumption (kg/h) ±0.31
- BSFC (kg/kW h) ±0.52
- BTE (%) ±1.82

The uncertainty of brake power is \[(\text{uncertainty of speed}) + (\text{uncertainty of load})\], which is equal to \[(0.2) + (0.01)\] = ±0.21. The uncertainty of fuel consumption is \[(\text{uncertainty of weight}) + (\text{uncertainty of time}) + (\text{uncertainty of temperature of fuel})\], which is equal to \[(0.01) + (0.1) + (0.2)\] = ±0.31. The uncertainty of BSFC is \[(\text{uncertainty of fuel consumption}) + (\text{uncertainty of brake power})\], which is equal to \[(0.31) + (0.2)\] = ±0.52. The uncertainty of BTE is \[(\text{uncertainty of BSFC}) + (\text{uncertainty of LHV})\], which is equal to \[(0.52) + (1.3)\] = ±1.82.

In this study, the engine speeds of 1100, 1400, 1700, 2000, and 2500 rpm were tested under the various engine loads of 25, 50, and 75% to determine the emissions and performance. The analyzer used to analyze NOx, CO, and CO₂ emissions and O₂ in the exhaust gas for emission testing (model: Testo 350 XL). Fuel consumption was measured by a digital balancing scale (model: AND EK-300i) to determine the unit of kg/h. After completing the engine test setup, the diesel engine was operated for 30 min at 1100 rpm of constant speed of the engine without load for the running-in period. After completing the run-in period, PFADE and fuel blends were changed to test the engine. After the engine run for 20 min with PFADE and fuel blends, the engine speed was slowly increased to 1100, 1400, 1700, 2000, and 2300 rpm speed at fixed engine loads of 25, 50, and 75%. During the diesel engine operation, the power, torque, speed, oil engine temperature, air temperature, exhaust temperature, fuel temperature, water inlet temperature, water outlet temperature (for determining the engine performance), and NOx, CO, CO₂, and O₂ (for determining the emissions) were recorded by the data logger.

3.3. Uncertainty Analysis. The values of accuracy and error for the data results obtained from the instrumentation of this testing were verified by the uncertainty analysis of the experiments. The percentage uncertainties of instruments are reported in Table 5. The CHNS/O analyzer was used to analyze the LHV. The LHV was used to calculate the uncertainty of the BTE. The percentage uncertainties of braking power, fuel consumption, BSFC, and BTE were found to be 0.21, 0.31, 0.52, and 1.82%, respectively, as noted in the footnote lists in Table 5. These percentages of uncertainties were calculated using the percentage uncertainties from several instrument sensors, including encoder sensors, strain gauge load cells, digital balance scales, digital stopwatches, temperature sensors, and CHNS/O analyzers. Therefore, the overall experimental uncertainty for the entire experiment was determined as ±1.83% using the following equation

The overall experimental uncertainty of performance and emission is

\[
\text{Uncertainty} = \sqrt{(\% \text{ uncertainty of CO})^2 + (\% \text{ uncertainty of CO}_2)^2 + (\% \text{ uncertainty of NO}_x)^2 + (\% \text{ uncertainty of O}_2)^2}
\]
+ (% uncertainty of EGT)² + (% uncertainty of BTE)²]
= square root of [(0.01)² + (0.1)² + (0.01)² + (0.06)²
+ (0.2)² + (1.82)²]
= ±1.83%

3.4. Long-Term Endurance Test. The effect of diesel—PFADE—ethanol blends on the different engine parts of the diesel engine was investigated for long-term endurance test. The fuel blend of D30PE60E10 was chosen out of three fuel blends of D50PE40E10, D40PE50E10, and D30PE60E10. 60% highest biodiesel content in dieselohol is an interesting blend to evaluate wear during long-term endurance tests. Furthermore, biodiesel has a greater density and viscosity than diesel, which may cause wear on engine components when evaluated under long-term endurance conditions. Therefore, the D30PE60E10 blend was extremely tested during the diesel engine durability testing. To prepare for the endurance testing, new parts of the diesel engine were fully dismantled and physically inspected to determine the dimensions of the different vital moving parts. Analysis of dimension, weight, and visual inspection of the vital parts (piston, cylinder liner, piston rings, connecting rod bearing, inlet valve, and exhaust valve) and FIE components (plunger, barrel, delivery valve, nozzle needle, valve holder, and fuel injector) in the diesel engine was carried out. After observation, the various parts were reassembled into the engine for a longer period test (500 h) at 2300 rpm maximum speed of the engine with D30PE60E10 fuel blend and diesel. The wear analysis evaluated the reliability of using D30PE60E10 blend and diesel after disassembling the various parts for a longer period test. During the running-in period, the diesel engine was run for 30 min at a constant speed of 1100 rpm. After the run-in period was completed, the diesel engine was increased to a constant speed of 2300 rpm for a longer period of 500 h. The diesel engine was stopped after every 100 h of continuous operation to replace the engine oil and oil filter, as recommended by the manufacturer’s instructions. After a 500 h run, the vital parts and FIE components were carefully dismantled to clear the surface deposits. For the analysis equipment, the engine parts were measured by a micrometer in the range of 0–25 mm (Mitutoyo, code number 103–137, Kanagawa, Japan), micrometer in the range of 75–100 mm (Mitutoyo, code number 103–140, Kanagawa, Japan), dial bore gauge (Starrett, United States), and vernier caliper (BEC, China) for the dimensional analysis. The precision weighing balances (OHAUS precision balances, model: PA4102, New Jersey, United States) were used to measure engine parts for weight analysis. The engine parts were photographed by a mobile phone 12-megapixel camera (iPhone 6s) with a macro lens 10X (Ztylus) for the visual inspection analysis.

4. CONCLUSIONS

The diesel—PFADE—hydrous ethanol blend was tested for performance, emissions, and long-term endurance on unmodified DI diesel engines. $P_h$, BSFC, and BTE of the fuel blends were studied in this research to determine the efficiency and emissions in diesel engines at various speeds and loads using a dynamometer. Furthermore, the wear of engine parts was examined after a long-term test with the maximum PFADE percentage in fuel blend. There are four main categories conclusions such as (1) engine performance, (2) emission analysis, (3) erosion analysis of engine components, and (4) recommendation condition for this study, which are described as follows:

1. For all fuel blends, the break power ($P_b$) of all fuel blends was significantly lower than that of diesel in the range of 1700 and 2000 rpm engine speed; however, $P_b$ started to increase after the engine speed reached 2000 rpm. For BSFC results, both PFADE and diesohol were greater than baseline diesel at 25 and 50% engine load. In all cases, the BSFCs of all fuel blends were greater than PFADE and diesel fuel. Ethanol has a lower heating value than PFADE and diesel. The highest BSFC was found in D30PE60E10 blend, which is the highest proportion of biodiesel mixing in the diesohol. The BTE of PFADE was higher than the baseline diesel at all engine speeds. At 50 and 75% engine load, PFADE’s EGT was lower than that of a baseline diesel. Thermal efficiency of PFADE is lower than that of diesel at high engine loads.

2. In all cases, the NOx emissions of diesohol were lower than that of diesel at 25 and 50% engine loads. The CO emissions from PFADE and all fuel blends were generally higher than baseline diesel. The delay ignition, which was due to containing a high amount of biodiesel and ethanol in the blends, can cause excessive CO emissions. Lower CO2 emissions occurred in the D50PE40E10 and D40PE50E10 blends at 25 and 50% engine load than in the baseline diesel. However, at 25 and 50% engine load, CO2 emissions increased at all engine speeds in the D30PE60E10 blend, which is the condition of the highest content of biodiesel contained in the fuel blend. For O2 gas emissions, all fuel blends emitted more than baseline diesel at all speeds except for the D30PE60E10 blend at 25 and 50% engine load.

3. The wear of engine parts comparison between D30PE60E10 blend and diesel was analyzed for 500 h long-term endurance test on diesel engine. The dimensional loss of D30PE60E10 blend contributed to a little lower than that of diesel on the essential components, except for the cylinder liner. However, it was found that running with D30PE60E10 caused less weight loss than with diesel for all vital parts except for the piston. Another finding was that the engine cannot be restarted after 100 h of continuous operation with D30PE60E10 blend because many pores were present at some parts of the fuel pump such as the surface of the plunger, barrel, delivery valve, and valve holder because of the effect of the cavitation effect from ethanol, which is one of the fuel blend’s components inside the fuel pump.

4. The PFAD, a byproduct of crude palm oil physical refining, was converted to PFADE by a single step of the esterification reaction. Because the PFAD is solid wax at 30 °C, it is not suitable for use as raw material for blending in fuel blends. The PFADE was added into fuel blend of hydrous ethanol—diesel blends for easy storage and longer stability of fuel blends. However, the PFADE and ethanol are good alternative fuels to reduce the use of diesel. Moreover, these fuels can be produced in Thailand, reducing the cost of import fuel from abroad. The diesel—PFADE—ethanol blend can be easily applied to the agricultural diesel engine, which is very beneficial.
for farmers and rural communities. Furthermore, this fuel blend can be used to generate electricity through the diesel engine generator. In this test, the D40PE50E10 blend was recommended as the optimal blended fuel for considering both engine performance and the impact on the environment. It can be regarded as an environmentally friendly fuel because using this fuel blend in the engine can reduce NOx, CO, and CO2 emissions. According to the results of this study, the D40PE50E10 fuel blend provided great performance under the condition of 50% engine load and 1400 rpm. It demonstrated a great potential to reduce NOx emissions by much more than 30% when compared to the baseline of diesel. Thus, D40PE50E10 blend can be applied as an environmentally friendly fuel in diesel engines. However, it is possible that the fuel pump, which is an important engine component, would need to be examined in terms of the types of materials that will be used to avoid corrosion from the cavitation bubble of ethanol when this fuel blend was adopted.

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