INTRODUCTION

A fast growing number of population, industries and automobiles have consumed massive stock from the limited remaining stock of the fossil fuel. Hence, it causes a rapid increase in atmospheric pollution, which contributes to the global warming because of the emission of greenhouse gases (GHG) from the engine. IPCC, 2014: Summary for policymakers reported that total anthropogenic CO₂ emission between 1750 and 2010 had tripled increased from 420 ± 35 Giga ton CO₂ to 1300 ± 110 Giga ton CO₂.¹ This condition is still increasing continuously due to the economic and population growth.² Therefore, many governments and researchers focus on reducing the GHG emission. Ministry of the Environment, Government of Japan, reported that the total emission was 1.322 million tonnes of CO₂eq/year in 2016. The total emission in 2016 decreased if it is compared to the total emission in 2015, 2013, and 2005 respectively from 1.2%, 7.3%, and 5.2%.³

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RESEARCH ARTICLE

Performance, rate of heat release, and combustion stability of dual-fuel mode in a small diesel engine

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Abstract
Biodiesel fuel and biogas fuel as promising alternative energy sources for dual-fuel-mode diesel engine attract more researchers. Therefore, this study highlights the dual-fuel mode (DFM) operation of diesel engine using biodiesel fuel from used cooking oil and simulated biogas fuel with different methane contents (M40, M60, M80, and M100) on the combustion, rate of heat release, combustion stability, and performance. The observation of diesel engine was conducted by varying the torques from 3.5 Nm, 10.5 Nm, 17.6 Nm, and 24.6 Nm and by varying the engine speeds from 1800 rpm, 2000 rpm, 2200 rpm, 2400 rpm, and 2600 rpm, respectively. It was found that in DFM at a low torque the thermal efficiency decreased, but biogas replacement, biogas energy, and sfc values increased. The relation of the results of biodiesel replacement, biogas energy ratio, and brake thermal efficiency with methane content ratio in DFM show that the methane content ratio has the maximum effect in DFM. In addition, the carbon dioxide content in the biogas can enrich the brake thermal efficiency. The combustion stability of all conditions (1.7%-4.89%) is still acceptable.

KEYWORDS
biodiesel, biogas, brake thermal efficiency, diesel engine, dual fuel, rate of heat release

1 | INTRODUCTION

A fast growing number of population, industries and automobiles have consumed massive stock from the limited remaining stock of the fossil fuel. Hence, it causes a rapid increase in atmospheric pollution, which contributes to the global warming because of the emission of greenhouse gases (GHG) from the engine. IPCC, 2014: Summary for policymakers reported that total anthropogenic CO₂ emission between 1750 and 2010 had tripled increased from 420 ± 35 Giga ton CO₂ to 1300 ± 110 Giga ton CO₂.¹ This condition is still increasing continuously due to the economic and population growth.² Therefore, many governments and researchers focus on reducing the GHG emission. Ministry of the Environment, Government of Japan, reported that the total emission was 1.322 million tonnes of CO₂eq/year in 2016. The total emission in 2016 decreased if it is compared to the total emission in 2015, 2013, and 2005 respectively from 1.2%, 7.3%, and 5.2%.³

In industries and automobiles trend around these recent years, the comprehensive applicability of diesel engine as power source in transportation, industries, and other needs have
rapidly increased over past century. It may be due to the high thermal efficiency, excellent fuel economy, and lower emission of unburned hydrocarbons (UHCs), carbon monoxide (CO), and carbon dioxide (CO₂) compared to the gasoline engine.⁴,⁶

But, the pollutant emission exhaust of smoke, particulate matter (PM), and nitrogen oxide (NOx) of the diesel engine is higher than the gasoline engine.⁷,⁸ which means pollutants from diesel engine potentially impact human health such as; visibility, vocational carcinogen in human lungs, influencing worst to the compromised lung function patients, and have another negative impacts.⁹,¹⁰ Therefore, biofuels, such as biodiesel fuel and biogas fuel, are the promising alternative energy sources for diesel engine due to the resource availability, nonhazardous environment, economy, and engine efficiency.

Biodiesel fuel can be produced from vegetable oil or animal fat by transesterification reaction, dilution, pyrolysis, and micro-émulsification.¹¹,¹² In addition, used cooking oil is also considered to be used in biodiesel synthesis, which can decrease the competition with food demand.¹¹ Therefore, biodiesel is a promising source energy that has a future of sustainability. Biodiesel has many advantages comparing to petroleum diesel: (a) It does not need engine modification or can be mixed with petroleum diesel.¹⁴,¹⁵ (b) The combustion efficiency and ignition property of biodiesel are better than petroleum diesel due to its high cetane number.¹⁶ (c) Carbon monoxide emission, smoke, and particulate matter of biodiesel are less than those of petroleum diesel fuel.¹⁷,¹⁸ (d) Biodiesel is nontoxic, has less sulfur and aromatic content, and has high lubricity,¹⁹ which can decrease the wear of engine.²⁰ However, biodiesel also has some disadvantages: (a) Biodiesel needs more amount of fuel compared to petroleum diesel in same power drive condition.²¹ (b) Impurity of biodiesel can cause corrosion in the diesel engine.²²,²³

Biogas fuel is another renewable resource energy that attracts many researchers to develop and apply in diesel engine. Generally, biogas is produced by anaerobic fermentation from the biological decomposition of organic matter such as biodegradable organic waste in landfill and agriculture. Table 1 shows the compositions of biogas from many resources.²⁴ The purification of biogas is important to remove the contents of hydrogen sulfide (H₂S) and water vapor that possibly leads to corrosion.²⁵ Application of biogas in diesel engine can reduce the pollutant emission.²⁶ However, biogas fuel cannot be applied directly into the diesel engine (CI engine) as it cannot work as it starts ignition automatically. It must be converted into spark-ignition engine or into dual-fuel mode using diesel biogas or biodiesel biogas, with diesel or biodiesel as pilot fuel.²⁷,²⁸

There are many researchers focused on the application of dual-fuel system in these recent years. Cacua et al.²⁹ conducted an investigation to evaluate the effect of air enriched with oxygen on dual-fuel engine using biogas as primary fuel. Feroskhan et al and Sarkar et al.³⁰,³¹ reported the effect of charge preheating on the performance of dual-fuel CI engine. Yousefi et al.³² reported that improving diesel injection timing decreases the unburned methane, which means higher thermal efficiency. Verna et al.³³ conducted an experiment with various compression ratios (CR), 16.5 17.5, 18.5, and 19.5 with using exhaust gas recirculation (EGR) and found that higher CR improved the energy efficiency and EGR had positive effect to the engine. Huang et al.³⁴ compared the result of experiment and simulation using CFD software and confirmed that both result can be applied to the simulated practical diesel and natural gas in dual-fuel engine mode. Also, Huang et al.³⁵ reported the effect of multiple injection on combustion in natural gas-diesel dual-fuel mode engine.

Moreover, an interesting phenomenon in brake thermal efficiency in dual-fuel mode diesel engine has been reported by Ambarita³⁶ and Nathan et al.²⁷ Increasing the biogas energy ratio in diesel gradually does not increase the brake thermal efficiency gradually. As a result, it will have a maximum condition of operation and fuel composition. This trend is an interesting phenomenon that should be observed.

This research is focused on the study of diesel engine run in dual-fuel mode without substantial modification. The ROHR analysis will be used to analyze the combustion performance of the engine. The interesting phenomena of the brake thermal efficiency will also be discussed in this paper. Besides that, the analysis of coefficient of variation mean effective pressure (COV imep) and combustion peak pressure in recommended engine performances is discussed in this paper.

**Table 1** Composition of Biodiesel from many sources

| Source      | CH₄ (%) | CO₂ (%) | O₂ (%) | N₂ (%) | H₂S (ppm) | Benzene (mg/m³) | Toluene (mg/m³) |
|-------------|---------|---------|--------|--------|-----------|-----------------|-----------------|
| Landfill    | 37.6-67.9 | 24.9-40.1 | <1-0.5 | 10-25 | 15.1-427.5 | <1-35.6         | 10-287          |
| Treatment   | 57.8-62.6 | 33.9-38.6 | 0-0.5  | 3.4-8.1 | 24.1-8000 | 0.1-5           | 0.1-5           |
| Agriculture | 45-75    | 25-55   | 0.01-2 | 0.01-5 | 10-30 000 | <300            | <300            |
2 | MATERIALS AND METHODS

2.1 | Simulated biogas and biodiesel properties

Methane (CH\textsubscript{4}) gas and carbon dioxide (CO\textsubscript{2}) gas are the main compositions of biogas fuel (Table 1). Therefore, the simulated biogas contains only CH\textsubscript{4} and CO\textsubscript{2} gases, which are used in this research. The CH\textsubscript{4} and CO\textsubscript{2} gases are produced by HOKKAIDO AIR WATER, INC., Japan. The composition of each gas is explained in Table 2. The simulated biogases of the following composition were used in the experiment: 40% CH\textsubscript{4} and 60% CO\textsubscript{2} (labeled M40), 60% CH\textsubscript{4} and 40% CO\textsubscript{2} (labeled M60), 80% CH\textsubscript{4} and 20% CO\textsubscript{2} (labeled M80), and 100% CH\textsubscript{4} (labeled M100) with flow rate 5 l/m. The static mixer T6-12-2PT was used to produce an artificial biogas from mixing CH\textsubscript{4} and CO\textsubscript{2}.

The pilot fuel was a biodiesel fuel from used cooking oil converted by using NaOH catalyst in transesterification reaction. The biodiesel was produced by Epoch Service Co., Ltd., Shiraoi, Japan. Table 3 shows the composition of biodiesel fuel. The biodiesel fuel acts as the source ignition on combustion chamber. The 100% pure biodiesel was used in whole experiments without being mixed with diesel fuel. In this experiment, we used the following fuel setup:

1. Biodiesel.
2. Biodiesel + M40 (Methane:CO\textsubscript{2} = 40:60).
3. Biodiesel + M60 (Methane:CO\textsubscript{2} = 60:40).
4. Biodiesel + M80 (Methane:CO\textsubscript{2} = 80:20).
5. Biodiesel + M100 (Methane:CO\textsubscript{2} = 100:0).

| TABLE 2 | Specification of methane and carbon dioxide |
|-----------------|-------------------|
| **Content** | **Chemical compound** |
| Molecular weight (g/mol) | 16.043 | 44.01 |
| Component concentration (%) | 99.99 | >99.95 |
| Density (273.15K, 0.1013 MPa) (g/L) | 0.7168 | 1.977 |
| Specific gravity (Air = 1, 21.1°C, 1 atm) | 0.554 | - |
| Vapor pressure (MPa abs) (20°C) | - | 5.733 |
| Solubility (m\textsuperscript{3}/m\textsuperscript{3}) (20°C) | 0.033 | 0.878 |
| Log Pow | 1.09 | 0.83 |
| Melting point (°C) | -182.5 | -56.6 |
| Boiling point (°C) | -161.5 | -78.5 |
| Flash point (°C) | -187.8 | - |
| Spontaneous ignition temperature (°C) | 600 | - |
exhaust gas. All the data of the exhaust gas temperature, the crankshaft angle, and the combustion pressure that recorded by the data logger were used to analyze the performance and combustion of the diesel engine.

In this study, the experimental procedures are explained as follows. The diesel engine is run in 5 fuel conditions as explained in section 3.1. In every fuel condition, torques are varied from 3.5 Nm, 10.5 Nm, 17.6 Nm, and 24.6 Nm, and engine speeds are varied from 1800 rpm, 2000 rpm, 2200 rpm, 2400 rpm, and 2600 rpm. These variations were conducted to measure the combination of engine performance from the low rpm to high rpm and from the low torque to high torque. The measurement is carried out for 5 minutes in stable condition of diesel engine in every torque and speed. Except of these conditions, the engine could not be measured in the stable condition during the measurement, which means the proper combination of torque and rpm is only by these variations. A total of 300 experiments have been performed, every experiment is repeated for three times, and the results are averaged.

### 2.3 Analysis procedure

In order to evaluate the engine performance, several equations are used here. The performance of diesel engine was analyzed using the thermal efficiency, the specific fuel consumption, and the diesel fuel replacement ratio. These equations are defined as below.

#### TABLE 3 Biodiesel composition

| Items            | Standard value | Result   | Method          |
|------------------|----------------|----------|-----------------|
| Kinetic viscosity | 3.50 mm²/s     | 4.196 mm²/s | JIS K 2283  |
| Triglyceride     | Max 0.20 mass % | 0.00 mass% | EN 14105       |
| Glycerin         | Max 0.02 mass % | 0.02 mass% | EN 14105       |
| Methanol         | Max 0.20 mass % | 0.02 mass% | EN 14110       |
| Water            | Max 500 ppm    | 565 ppm  | JIS K 2275     |

#### TABLE 4 Engine specification

| Engine model                     | 4 Stroke Air cooled, Single cylinder, Direct-injection |
|----------------------------------|------------------------------------------------------|
| Product number                   | DY-30                                                |
| Compression ratio                | 21                                                   |
| Bore × Stroke                    | 76 × 66 mm                                           |
| Stroke volume                    | 0.000299 m³                                          |
| Maximum power                    | 4.84 kW/1750 rpm                                    |
| Maximum torque                   | 31.57 Nm/1160 rpm                                   |
| Fuel injector opening pressure   | 19123 kPa                                            |
| No. of nozzle hole (Diameter)    | 4 (0.22 mm)                                          |
| Intake air                       | Naturally aspirated                                  |

FIGURE 1 Experimental setup
The thermal efficiency $\eta$ [%] that indicates how much heat input to the diesel engine will convert output power is shown as:

$$\eta = \frac{W_e}{W_f} \cdot 100$$

where $W_e$ [kW] is the net horsepower, $W_f$ [kW] is the fuel horsepower.

Net horsepower: $W_e$ [kJ/h]

$$W_e = 3.60 \left( 2\pi \frac{0.4833N_e}{60} \right)$$

Torque: $T$ [Nm]

$$T = wq_c l$$

where $N_e$ [rpm] is the engine rotation, $w$ [kgf] the weighing torque, $l$ [m] (= 0.3581 m) the shaft length of the dynamometer, and $q_c$ [N/kgf] (= 9.80 N/kgf) the power conversion factor.

Fuel horsepower: $W_f$ [kJ/h]

$$W_f = W_{\text{diesel}} + W_{\text{biogas}}$$

Fuel horsepower of diesel fuel: $W_{\text{diesel}}$ [kJ/h]

$$W_{\text{diesel}} = 3.60 \left( H_{\text{diesel}} G_{\text{diesel}} \right)$$

Fuel horsepower of biogas: $W_{\text{biogas}}$ [kJ/h]

$$W_{\text{biogas}} = 3.60 \left( H_{\text{biogas}} G_{\text{biogas}} \right)$$

Mass flow rate of diesel fuel: $G_{\text{diesel}}$ [g/s]

$$G_{\text{diesel}} = \frac{\rho_{\text{diesel}} b}{t}$$

Mass flow rate of CH₄: $G_{\text{CH₄}}$ [g/s]

$$G_{\text{biogas}} = \frac{v_{\text{CH₄}} \rho_{\text{CH₄}}}{60} \cdot P$$

where $H_{\text{diesel}}$ and $H_{\text{biogas}}$ [kJ/kg] are the lower calorific values of diesel fuel and CH₄, $\rho_{\text{diesel}}$ and $\rho_{\text{CH₄}}$ [g/cm³] the density of diesel fuel and CH₄, $b$ [cc] (= 50 cc) the fuel tank volume, $t$ [s] the fuel consuming time, $v_{\text{CH₄}}$ [L/min] the volume flow rate of CH₄, $P$ [%] the content of CH₄ fraction.

The specific fuel consumption (SFC) [g/kWh] is a comparison of fuel consumption to the useful energy. Here, it can be viewed as how many liters of fuel are needed to produce 1 kWh of electrical energy. For single-fuel operation mode, it can be calculated as below.

$$SFC_{\text{single}} = 3600 \cdot \frac{G_{\text{diesel}}}{W_e}$$

While for dual-fuel mode is defined as:

$$SFC_{\text{dual}} = 3600 \cdot \frac{G_{\text{diesel}} + G_{\text{biogas}}}{W_e}$$

The diesel fuel replacement ratio [%] shows how much the percentage of diesel fuel replaced by the biogas. It is defined as following formula.

$$r = \frac{G_{\text{diesel}} - G_{\text{dual}}}{G_{\text{diesel}}} \cdot 100$$

where $G_{\text{diesel}}$ [g/s] is the mass flow rate of diesel fuel, $G_{\text{dual}}$ [g/s] the mass flow rate of diesel fuel in dual-fuel mode.

Biogas energy ratio (ber) is calculated to analysis the value of energy input from biogas to the total energy in dual-fuel mode.

$$\text{ber} = \frac{W_{\text{biogas}}}{W_f}$$

Rate of heat release (ROHR) in Equation is commonly used to analyze the combustion performance of the engine, where $\lambda$ is the ratio of specific heat ($\lambda = 1.35$ in this research), $P$ is combustion pressure, $V$ is cylinder volume, and $\theta$ is crankshaft angle.

$$\frac{dQ}{d\theta} = \frac{\lambda}{\lambda - 1} P \frac{dV}{d\theta} + \frac{1}{\lambda - 1} \frac{dP}{d\theta}$$

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$$\text{COV}_\text{imep} = \frac{\sigma_{\text{imep}}}{\bar{x}} \times 100$$

Besides ROHR, coefficient of variation of indicated mean effective pressure (COV_imep) analysis is also important in combustion analysis. $\sigma_{\text{imep}}$ is the standard deviation, and $\bar{x}$ is average imep value. By using the above equations, the performance of the diesel engine was analyzed.

## RESULTS AND DISCUSSIONS

### 3.1 Performance of dual fuel (biodiesel-simulated biogas)

The results of experiments are presented and discussed in detail here. Brake thermal efficiency, biodiesel replacement,
biogas energy ratio, specific fuel consumption, and exhaust gas temperature are discussed in every subsection.

3.1.1 | Brake thermal efficiency

Brake thermal efficiency (BTE) is one of the indicators to evaluate the performance of diesel engine. Figure 2 shows the BTE in variation applied torque with different engine speeds. Generally, increasing the torque increases the value of BTE. However, the BTE in dual-fuel mode is lower than diesel mode at low torque, 3.5 Nm, and higher than diesel mode at the highest torque condition 24.6 Nm.

Due to the effect of engine speed, the highest BTE in 2600 rpm is 24.95% in operation M100 with torque 24.6 Nm, the highest BTE in 2400 rpm is 26.28% in operation M80 with torque 24.6 Nm, the highest BTE in 2200 rpm is 26.27% in operation M100 with torque 24.6 Nm, the highest BTE in 2000 rpm is 26.72% in operation M100 with torque 24.6 Nm, and the highest BTE in 1800 rpm is 27.79% in operation M80 with torque 24.6 Nm. Due to the effect of engine torque, the highest BTE in torque 3.4 Nm is 10.73% in operation biodiesel with speed 2200 rpm, the highest BTE in torque 10.5 Nm is 21.79% in operation M40 with speed 1800 rpm, the highest BTE in torque 17.6 Nm is 26.75% in operation M40 with speed 1800 rpm, the highest BTE in torque 24.6 Nm is 27.79% in operation M80 with speed 1800 rpm. This experiment result has a same trend with Bora et al.37 that report the result of rice bran biodiesel and biogas in dual-fuel diesel engine mode. These experiment results also show that brake thermal efficiency increased by increasing the load.38

The effect of fuel composition in this research: In the low torque of 3.5 Nm with all methane content ratio adjustment reduced the BTE, higher methane adjustment more reduced the BTE. It happened because the low temperature of the engine and the present of methane reduced the oxygen to help the combustion. In the medium torque 10.5 and 17.6 Nm, BTE increased in certain percentage of methane gas. This occurred due to the content ratio of carbon dioxide in the combustion, carbon dioxide dissociated into carbon monoxide and oxygen as explained by Sarkar et al.39 Increasing oxygen in the mixture will improve engine performance. High torque of 24.6 Nm shown in all methane content ratio adjustment improves the BTE. However, the best achieved is in the lower rpm, this condition occurred because dissociated carbon dioxide needs high temperature and the low speed engine gives
more time to burn the fuel air mixture and produce better efficiency. Overall, low applied torque reduces the brake thermal efficiency. This condition occurred due to poor combustion of biogas in low temperature as a proof the highest brake thermal efficiency achieved at 27.79% in operation M80, 1800 rpm with torque 24.6 Nm. The BTE is exactly influenced by torque, engine speed, and biogas composition.

3.1.2 | Biodiesel replacement and biogas energy ratio

Biodiesel replacement (BR) is calculated using Equation . BR calculation is important to know how much percentage of biodiesel fuel is replaced by the biogas. Figure 3 shows graphs of relationship between torque and BR results. Due to the effect of engine speed, the highest BR in 2600 rpm is 42.9% in operation M100 with torque 3.5 Nm, the highest BR in 2400 rpm is 49.1% in operation M100 with torque 3.5 Nm, the highest BR in 2200 rpm is 51.9% in operation M100 with torque 3.5 Nm, and the highest BR in 1800 rpm is 60.2% in operation M100 with torque 3.5 Nm. Due to the effect of engine torque, the highest BR in torque 3.4 Nm is 60.2% in operation M100 with speed 1800 rpm, the highest BR in torque 10.5 Nm is 54.8% in operation M100 with speed 1800 rpm, the highest BR in torque 17.6 Nm is 49.2% in operation M100 with speed 1800 rpm, and the highest BR in torque 24.6 Nm is 40.6% in operation M100 with speed 1800 rpm.

The above explanation shows that the low torque applied and engine speed show higher biodiesel fuel replacement. This BR result also has a correlation to the BTR in section before. The highest BR value is 60.2% in operation M100, 1800 rpm, and torque 3.5 Nm.

The characteristic of biogas energy ratio has a similar trend with biodiesel replacement. Figure 4 shows the biogas energy ratio in variation applied torque with different engine speeds. The low torque applied and engine speed show higher biogas energy ratio value. The highest biogas energy ratio value is 71.4% in operation M100, 1800 rpm, and torque 3.5 Nm. And this biogas energy ratio result is in the range of the result obtained by Sarkar et al.40

3.1.3 | Specific fuel consumption

Specific fuel consumption (SFC) is used as an indicator to evaluate the engine ability to transform the amount of consumed fuel into mechanical energy produced. The low SFC value means the engine consumes less fuel to produce mechanical energy; hence, it is highly efficient. Equation is used to calculate the SFC. Figure 5 shows specific fuel consumption in variation applied torque at different engine speeds. Generally, by increasing engine torque decreased the

![Figure 3](image_url)
SFC. The SFC is influenced by engine speed, engine torque, and fuel composition. Figure 5 shows that the low torque applied and engine speed show higher SFC value. The SFC in this experiment varies from 303.93 g/kWh to 1010.64 g/kWh. The lowest SFC is in engine condition 1800 rpm, fuel composition M100, and torque 24.6 Nm. The highest SFC is in engine condition 1800 rpm, fuel composition M100, and torque 3.5 Nm. However, this result is confirmed in the range as reported in reference 27 that the range of SFC is 0.42 kg/kWh to 2.57 kg/kWh.

3.1.4 | Exhaust gas temperature

Exhaust gas temperature was measured in this experiment. Figure 6 shows the result of exhaust gas temperature in variation applied torque. In general, increasing engine torque increased the exhaust gas temperature. The exhaust gas temperature in this experiment varies from 61.64°C to 312°C. The lowest exhaust gas temperature is in engine condition 1800 rpm, biodiesel, and torque 3.5 Nm. The highest is in engine condition 2600 rpm, biodiesel, and torque 24.6 Nm. However, experimental result of Bora et al. has a range of exhaust gas temperature under 400°C, which means these exhaust gas temperature results are also reasonable.

Figure 6 indicates that high CO₂ content in biogas has higher exhaust temperature than other DFM condition. Biodiesel mode has higher exhaust gas temperature than DFM M100 condition. Especially in the high speed like 2400 rpm and 2600 rpm, the methane gas needs more time for the burning process in the combustion. The condition did not make all the methane gas burned in this condition that caused decreasing exhaust gas temperature. This condition has to be controlled to prevent the knocking that may happen in the experiment, which can affect the life of the engine. The clear explanation of the combustion will be discussed in the next section 4.2.

3.1.5 | Analysis of methane composition of fuel setup

Sections 4.1.1 and 4.1.2 show the result of brake thermal efficiency and biogas energy ratio to the torque of the diesel engine. The results show a fact that higher methane composition in dual-fuel mode does not directly increase the thermal brake efficiency of the diesel engine. However, increasing the methane content decreased the brake thermal efficiency. It can be observed by comparing the value of the brake thermal efficiency result to biogas energy fuel.
Two researchers found the same conditions of this problem. Nathan et al. reported the relation of thermal brake efficiency and biogas energy ratio in different bmep from 2.5 bar to 4 bar. Ambarita reported the relation of thermal brake efficiency and biogas energy ratio in different flow rate of different composition of injected fuel. In this manuscript, the methane composition of fuel setup was conducted. Figure 7 shows the relation of brake thermal efficiency with biogas energy ratio in every fuel setup in different speeds and loads.

In all conditions of torques and speeds, M40 has the highest thermal brake efficiency with the lowest biogas energy ratio value. Contrarily, M100 has the lowest thermal brake efficiency with the highest biogas energy ratio value. An interesting condition was found between M60 and M80. Increasing composition from M60 to M80 for 2000 rpm, 2200 rpm, and 2600 rpm made the efficiency decreased in torque 3.5 Nm and 10.5 Nm, but it increased in load 17.6 Nm and 24.6 Nm. However, increasing composition from M60 to M80 for 2400 rpm made the efficiency decreased in torque 3.5 Nm 10.5 Nm and 17.6 Nm, but it increased in load 24.6 Nm. The trend of Figure 7 shows that increasing methane composition is more effective in the high speed and increasing the methane composition will definitely increase the efficiency in low speed. It may be due to the carbon monoxide that unaffected the brake thermal efficiency as reported by. The relation of methane composition due to the composition and ROHR will be discussed in the next section.

### 3.2 Combustion pressure and rate of heat release of dual fuel (biodiesel-simulated biogas)

Diesel engine works in four steps, and it can be observed in the combustion process graph. Commonly in diesel engine, four stages in combustion process are as follows: (1) ignition delay period (start of fuel injection), (2) precombustion (all the fuel had been injected, and the pressure increases rapidly), (3) main combustion (the fuel is burned and produces power), and (4) late combustion (less combustion takes place, and the pressure is going down). Since the combustion process of dual-fuel mode is more complex than the diesel mode, O.M.I Nwa for divided the combustion process into 5 stages in the dual-fuel mode. The difference is in the ignition delay process because it needs
time for the biogas fuel to be burned. All results of combustion pressure and ROHR in these experiments are shown in figures from Figures 8-12. The dotted line is ROHR, and the line is combustion pressure.

Figure 8 shows the combustion pressure and ROHR of the engine in biodiesel fuel and dual-fuel mode in 1800 rpm with different torques. Higher methane content ratio decreases the combustion peak pressure in the low torque 3.5 Nm (Figure 8A) and medium torque 10.5 Nm (Figure 8B). In the medium torque 17.6 Nm, Figure 8C shows that the combustion pressure between dual-fuel mode and biodiesel mode in certain methane content has higher combustion pressure. In high torque 24.6 Nm, Figure 8D shows that all the combustion pressure of dual-fuel mode becomes higher than that of the biodiesel mode.

The ROHR in 1800 rpm (low speed) in Figure 8 shows the combustion pressure and ROHR of the engine in biodiesel fuel and dual-fuel mode in 1800 rpm with different torques. Generally, the combustion pressure shows similar patterns as in 1800 rpm in Figure 8. However, the difference of ignition time delay was found in this condition. The shorter ignition delay was occurred compared to 1800 rpm, and higher load produced shorter ignition delay. In the medium to high loads, the ROHR of biodiesel became lower than that of the dual-fuel mode. As seen from diagram after the premixed combustion occurred from medium and high loads, ROHR curve clearly shows higher mixing controlled combustion phase. Furthermore, more fuel injected during higher load extended the combustion process. The ROHR pattern during low load, medium load, and high load are reported in this literature.

Figure 9 shows the combustion pressure and ROHR of the engine in biodiesel fuel and dual-fuel mode in 2000 rpm with different torques. Generally, the combustion pressure shows similar patterns as in 1800 rpm in Figure 8. However, the difference of ignition time delay was found in this condition. The shorter ignition delay was occurred compared to 1800 rpm, and higher load produced shorter ignition delay. In the medium to high loads, the ROHR of biodiesel became lower than that of the dual-fuel mode. As seen from diagram after the premixed combustion occurred from medium and high loads, ROHR curve clearly shows higher mixing controlled combustion phase. Furthermore, more fuel injected during higher load extended the combustion process. The ROHR pattern during low load, medium load, and high load are reported in this literature.

Figures 10 and 11 show the combustion pressure and ROHR of the engine in biodiesel fuel and dual-fuel mode, respectively, in 2200 rpm and 2400 rpm with different torques in both conditions, the curves between maximum pressure for each load clearly shows that dual fuel in low load produces lower combustion peak pressure and higher methane content reduces the combustion peak pressure. However, in the medium to high load the combustion pressure becomes
higher when dual fuel is applied. The ROHR curve shows that rapid combustion and longer mix controlled combustion occurred in higher load. The increase of speed and amount of fuel injected during high load increase the pressure and temperature, which makes the ignition delay become shorter; in the high load, the ignition starts 3 crankshaft angle degrees earlier. It also reported in the literature.\textsuperscript{43}

In the high engine speed 2600 rpm shown in Figure 12, the combustion pressure overall becomes lower compared to all the other engine speed before. High speed makes less time in the combustion process and affects the combustion pressure and ROHR of the engine. In the low torque, 3.5 Nm longer ignition delay can be clearly found shifting to 4 crankshaft angle degrees after the top dead center. During higher load applied the ignition delay become shorter, but has longer in mix controlled combustion similar to the previous engine speed. It made a critical time for fuel to be burned especially in dual-fuel mode that indicates some of the fuel was not burned. Increasing engine speed decreased the ROHR value. This condition shows that not all fuel was burned in the high speed condition. The unburned fuel may be released together with exhaust gas.

The combustion pressure, ROHR, and the ignition delay are influenced by the presence and the content of the biogas, torque, and engine speed. Overall, increasing the engine speed increased the ignition delay. In addition, the presence of higher methane content ratio in the low torque dropped the combustion peak pressure. Ignition delay can be influenced by some factors such as: (a) Physical delay (fuel is atomized, vaporized, mixed with the air) depends on type of fuel, injection pressure, etc.\textsuperscript{28} (b) Chemical delay (preflame reaction takes place, depending on the temperature of the combustion chamber).\textsuperscript{38} However, biogas self-ignition point is higher than biodiesel, and higher combustion temperature helps the ignition of the biogas. Mustafi et al\textsuperscript{44} report that perhaps this condition caused by biogas has higher specific heat compared to diesel.

### 3.3 Combustion stability

The combustion stability was analyzed to compare the stability of the combustion of each fuel using coefficient of variation of the indicated mean effective pressure (\(\text{COV}_{\text{imep}}\)) at 2400 rpm. Figure 13 shows the relation between \(\text{COV}_{\text{imep}}\) and torque. Total of variation of the averaged 30 engine cycles values were used to build \(\text{COV}_{\text{imep}}\) in this figure. Figure 13 shows that all the \(\text{COV}_{\text{imep}}\) results are acceptable due to being lower than 10%.\textsuperscript{45,46} Generally, five percent
FIGURE 8  Pressure in cylinder and ROHR vs crankshaft angle position of the engine with speed 1800 rpm at (A) 3.5 Nm, (B) 10.5 Nm, (C) 17.6 Nm, and (D) 24.6 Nm
FIGURE 9  Pressure in cylinder and ROHR vs crankshaft angle position of the engine with speed 2000 rpm at (A) 3.5 Nm, (B) 10.5 Nm, (C) 17.6 Nm, and (D) 24.6 Nm
FIGURE 10  Pressure in cylinder and ROHR vs crankshaft angle position of the engine with speed 2200 rpm at (A) 3.5 Nm, (B) 10.5 Nm, (C) 17.6 Nm, and (D) 24.6 Nm
FIGURE 11  Pressure in cylinder and ROHR vs crankshaft angle position of the engine with speed 2400 rpm at (A) 3.5 Nm, (B) 10.5 Nm, (C) 17.6 Nm, and (D) 24.6 Nm
FIGURE 12  Pressure in cylinder and ROHR vs crankshaft angle position of the engine with speed 2600 rpm at (A) 3.5 Nm, (B) 10.5 Nm, (C) 17.6 Nm, and (D) 24.6 Nm
value of COV$_{\text{imep}}$ is considered as the value of the cutoff for the combustion stability. At low torque 3.4 Nm, increasing methane composition in gas fuel increased the value of COV$_{\text{imep}}$. However, increasing the torque decreased the COV$_{\text{imep}}$ because higher load mean increases the amount of fuel injected and helps rich fuel mixture in the combustion chamber. Therefore, the most stable operation value is the biodiesel. In the dual-fuel operation, there is a pattern that certain methane gas content with the present of the carbon dioxide produces more stable combustion.

Figure 14 shows the relation between COV$_{\text{imep}}$ and biogas fuel ratio in 2400 rpm as an indicator stability of combustion. Increasing biogas fuel ratio increases the COV$_{\text{imep}}$ percentage value. The combustions were more stable in lower biogas energy ratios.

4 | CONCLUSIONS

The experiment was conducted in diesel engine without substantial modification run in dual-fuel mode and biodiesel mode with variants of five fuels, five engine speeds, and four torque variations. The performance, rate of heat release, and the stability of combustion in both modes have been discussed in section 4. The results show:

1. In the low torque, dual-fuel mode decreased the thermal efficiency. Oppositely, the thermal efficiency increased in high torque. Furthermore, engine speed and methane content ratio uncertainty affected thermal efficiency. The highest thermal efficiency 27.79% was achieved when the engine speed was at 1800 rpm, 24.6 Nm, M80. In addition, the highest thermal efficiency was achieved at 24.6 Nm, M100, when engine speed was at 2600 rpm.
2. There is a condition that carbon dioxide content in the biogas can enrich the brake thermal efficiency of the engine. The highest brake thermal efficiency 27.79% was achieved at dual-fuel mode M80.
3. Biodiesel replacement and biogas energy share become higher when low torque and low engine speed were applied in DFM with higher methane content ratio. The maximum 60.2% and 71.4%, respectively, for biodiesel replacement and biogas energy share at 1800 rpm, 3.5 Nm, M100.
4. The relation between biodiesel replacement, biogas energy ratio, and brake thermal efficiency due to methane content show that there is a maximum effect of methane content ratio in dual-fuel mode. From exhaust temperature shown in the high torque, and high engine speed, the
exhaust temperature significantly decreased because of an increased unburned rich fuel mixture.

5. Engine speed, loads and methane content ratio uncertainty affected the specific fuel consumption. The highest specific fuel consumption 1010.64 g/kWh at 1800 rpm, 3.5 Nm, M100. The lowest specific fuel consumption 303.93 g/kWh at 1800 rpm, 24.6 Nm, M100.

6. The combustion pressure and the rate of heat release showed that methane content ratio affected engine speeds and loads conditions. Overall, low torque produces lower combustion pressure, and lower rate of heat release by using dual-fuel mode particularly when more methane content ratio was adjusted. In addition, longer heat release with shorter ignition delayed in high torque operation.

7. The instability of the engine escalating when COV imper values increased significantly from 1.7% to 4.89% when the methane content ratio rises in the low torque condition.

CONFLICTS OF INTEREST

The authors declare no conflict of interest.

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