International Journal of Automotive and Mechanical Engineering
ISSN: 2229-8649 (Print); ISSN: 2180-1606 (Online);
Volume 14, Issue 1 pp. 4030-4049 March 2017

DOI: https://doi.org/10.15282/ijame.14.1.2017.16.0326

Performance and emission characteristics of micro gas turbine engine fuelled with bioethanol-diesel-biodiesel blends

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ABSTRACT

Biodiesel is receiving increasing attention as an alternative fuel due to the ever-growing demand for energy. However, the inferior physiochemical properties of biodiesel render it incompatible for gas turbine application, which needs to meet the standard requirement of gas turbine fuel accordance to ASTM D2880. In this quest, the biodiesel-diesel–bioethanol blends might be a good option. In this paper, the research work was carried out to study experimentally the performance and exhaust emission characteristics of a 25kW micro gas turbine engine (Capstone Model C30) fuelled with biodiesel-diesel-bioethanol blends. The assessment on the improved fuel properties of biodiesel by blending with bio-ethanol had shown more superior atomisation characteristics performance compared to unmodified biodiesel. Moreover, the performance test in the micro gas turbine was limited up to 20% blend of biofuel, which showed improved thermal efficiency during the test. Subsequently, the emission test carried out in this work also showed significant enhancement in emissions, except nitrogen oxides (NO_x) which contributed to the higher formation in comparison with the distillate diesel. Finally, B80E20 (80:20 of biodiesel-bioethanol) was proposed to be selected as an ideal blended fuel ratio to be applied in micro gas turbine engine due to its adaptability to replace diesel fuel, while showed better performance and emission properties as compared to the pure petroleum diesel.

Keywords**:** Gas turbine; biodiesel; bioethanol;, atomisation; Sauter mean diameter; engine performance; emissions

INTRODUCTION

Today, energy-intensive activities are the highest contributors to the increase in carbon dioxide (CO_2) emissions and fossil fuel combustion efficiency accounts for 90% of the total CO_2 emissions [1-3]. Power generation remains the most important sector related to fossil fuel consumption. Therefore, the power sector choice of fossil fuel is of the utmost importance in reducing CO_2 emissions [4]. The global average annual growth rate of 2.4 ppm in atmospheric CO_2 concentrations in 2012 was rather high [5]. Renewable energy (RE) resources have become an increasingly significant part of power generation in the efforts to reduce fossil fuel consumption and pollutant emissions [6]. Among these, the two main types of liquid biofuels are bioethanol and biodiesel; both drawing considerable attention in the recent years [7]. Biodiesel is methyl ester of triglyceride prepared from edible or non-edible vegetable oils (virgin or used) or animal fats by the conversion of the triglycerides to esters via transesterification. The reproducibility, nontoxicity, and sulphur-free property of biodiesel have generated a lot of research interest. The primary focus has been on the use of biodiesel in diesel engines [1-3, 8-11]. The compatible physiochemical properties of biodiesel to diesel fuel have allowed up to 20% blending of biodiesel with petroleum diesel for the application in diesel engines without any modification [12-17].

Recently, there has been a lot of interest to consider biodiesel fuel for gas turbine application. Micro gas turbines (MGTs) are becoming more popular and experiencing greater demand due to their advantages of being small, modular, reliable. It is more flexible in terms of fuel, have compact size and light weight, are low in maintenance costs and emissions levels and high in efficiency, and lower electricity costs [18, 19]. Micro gas turbines are poised to take over from petrol and diesel reciprocating engines in a number of key applications due to their superior performance. Micro gas turbines are originally designed for the use of crude oil derivatives. However, it has been shown that biodiesel has shortcomings that need to be improved before it can be considered for gas turbine application. A study by Ivaniš, Radović [20] reported that biodiesel has higher density and viscosity than conventional diesel which results in poor atomisation of the fuels and may clog fuel nozzles. The properties of a liquid fuel that affect atomisation are viscosity, density, and surface tension. Atomisation refers to the process of breaking up bulk liquids into droplets. Atomisation plays a major role in the combustion efficiency and emission in gas turbines engines. Adequate atomisation enhances mixing and completes the combustion efficiency in a direct injection gas turbine. Higher density causes the fluid to resist acceleration and tends to result in a larger average droplet size. An increase in fuel density will also increase the surface tension of the fuel. Surface tension tends to stabilise the fluid and prevents its breakup into smaller droplets; hence, adversely affects the atomisation of the liquid. Viscosity causes the fluid to resist agitation, prevents its breakup and leads to a larger average droplet size. The use of a fuel with higher viscosity delays atomisation by suppressing the factors required to make the fuel spray to break up. Extensive research by Gao, Deng [21] on the spray characteristics of biodiesel found that the Sauter mean diameter (SMD) of biodiesel-blended fuels is larger than that of fossil diesel because of the higher viscosity and surface tension of biodiesel. Evaporation and atomisation of the biodiesel are relatively more difficult because of the higher surface tension and viscosity of the biodiesel.

Several techniques are available in order to modify the physical properties of the biofuel [22]. The properties of the fuels require modification according to the demands of the equipment. Several studies tried to use preheating in order to reduce the viscosity of the biodiesel [9, 23-29]. Some researchers showed that distillation process can be applied to improve the physical properties that affect atomisation characteristics of biodiesel for gas turbine application such as viscosity, density, and surface tension [30]. However, these processes are energy intensive and time consuming; hence, finding a non-laborious process (such as blending with bioethanol) is highly recommended. The outcome of the work by Yilmaz and Sanchez [31] implied that biodiesel–bioethanol blends are more effective than biodiesel–methanol blends in improving engine performance and emission. The addition of less viscous fuels with smaller surface tension into biodiesel is potentially capable of recovering the deteriorated spray characteristics compared to diesel. Similarly, Yoon, Park [32] showed that adding up to 20% (by volume) of bioethanol into biodiesel

(BE20) enhances the spray characteristics of fuel blends. Studies showed that blending lower alcohols such as methanol and bioethanol with biodiesel only in lower proportions improves the engine performance. This is because of their lower heating value and higher latent heat of vaporisation [33, 34].

Accordingly, limited numbers of studies have focused on the application of bioethanol–biodiesel blends for the gas turbine engine. Similarly, there is a scarcity of published works on data examining combustion efficiency and emission of biodiesel blended with bioethanol, in particular, bioethanol produced from waste glycerol (a by-product of biodiesel transesterification process). The notion that there is an enhancement of the combustion efficiency of biodiesel fuel when added with oxygenated fuels such as bioethanol needs a further study. Such investigation is important to understand which kind of blends is more effective in reducing both NO_x and particulate emissions of a gas turbine engine. Hence, in evaluating the potential of bioethanol blended with biodiesel as an alternative fuel for gas turbines, it is a priority to ensure that the properties of the fuels are compatible with the gas turbine fuel properties so as not to violate the warranty of the equipment. Thus, the aim of the study is to investigate the performance and emissions of a gas turbine engine operating on bioethanol-Diesel-biodiesel blends, using biodiesel produced from palm oil and bioethanol produced from waste glycerol.

METHODS AND MATERIALS

Blend Fuel Preparation and Property Test Evaluation

The biodiesel obtained in this study was produced by utilising palm oil as the feedstock (via the alkali catalyst mediated trans-esterification process), which was obtained from the Sime Darby Biodiesel plant (Klang, Malaysia). Hence, the bioethanol used was produced from waste glycerol, as described elsewhere by Saifuddin and Refal [35]. Prior to its use in the experiments, the bioethanol used was dehydrated, which had undergone the prior water removal step as performed by following the method of Tomanee [36] without any modification. Distillate diesel as the baseline fuel was obtained locally (Petronas Gas Station, Malaysia). There were seven types of fuel used in the study for preliminary atomisation which included pure bioethanol (E100), pure biodiesel (B100), B20E80 (biodiesel 20% blend with 80% bioethanol), B40E60 (biodiesel 40% blend with 60% bioethanol), B60E40 (biodiesel 60% blend with 40% bioethanol), B80E20 (biodiesel 80% blend with 20% bioethanol), and distillate diesel (DD). However, there were also three types of fuel used for the performance and emission tests for micro gas turbine, namely, (i) 90% Diesel: 9.0% Biodiesel: 1.0% Bioethanol (DBE10%), (ii) 85% Diesel: 12.75% Biodiesel: 2.25 % Bioethanol (DBE15%) and (iii) 80% Diesel: 16% Biodiesel: 4% Bioethanol (DBE20%). In this paper, all performance and emissions tests were performed at five engine loads, namely 5, 10, 15, 20, and 25KW. All the fuel preparations were subjected to the property test evaluation in accordance with ASTM D2880 Standard Specification fuel oil requirements for gas turbine application, which were performed by third-party laboratories, namely, the TNB Research Laboratory and ITS Testing Services (M) SDN BHD.

Numerical Evaluation of Preliminary Atomisation Characteristics

The atomisation characteristics of blended biodiesel with various ratios of bioethanol were measured by determining the SMD parameter. SMD is designated as D_{32} and a very common parameter in fluid dynamics used for expressing the fineness of a spray in terms of the surface area, viscosity, and density of the spray droplets. The atomisation characteristic analysis was done numerically using Equation (1) reported by Lefebvre and McDonell [37].

$$\frac{D_{32}}{d_0} = 0.48\left(\frac{\sigma}{\rho_A U_R^2 d_0}\right)^{0.4}\left(1+\frac{1}{ALR}\right)^{0.4} + 0.15\left(\frac{\mu_L^2}{\sigma\rho_L d_0}\right)^{0.5}\left(1+\frac{1}{ALR}\right) \tag{1}$$

where D_{32} = Sauter Mean Diameter; d_0 = Liquid discharge opening diameter (m); σ = Liquid surface tension (N/m); ρ_A = Density of air (Kg/m^3); U_R = Relative co − flowing velocity of the two streams (m/s); ρ_L = Density of liquid (kg/m^3); μ_L = Liquid viscosity (kg/ms); and ALR = Air to liquid mass flow ratio.

Droplet evaporation time is another crucial element in atomisation and depends on the droplet size of the fuel. The fuel droplet size after atomisation depends on fuel delivery geometry and the properties of the fuel such as density, viscosity, and surface tension, while the evaporation rate depends on the specific heat, temperature at the evaporation zone and chemical structure of the-fuel molecules. Bolszo [38] analysed the evaporation time of diesel fuel in micro gas turbines using effective evaporation constant and droplet evaporation lifetime. Thus, a similar approach was used to evaluate the evaporation time of bioethanol and biodiesel. The details of Equations (2) to (4) are given for effective evaporation constant (λ_{eff}) calculated based on Bolszo [38].

$$t_e = \frac{D_{32}^2}{\lambda_{eff}} \tag{2}$$

where t_e = Effective evaporation time (s); D_{32} = Sauter mean diameter (m) ; λ_{eff} = Effective evaporation constant (m^2/s).

Since

$$B = \left(\frac{1}{L}\right)\left[Cp(T_\infty - T_\iota + (Q\ \frac{Y_{0\infty}}{i}\right] \tag{3}$$

where L = Latent heat of vaporisation per unit mass of fuel (kJ/kg); T_∞ = Temperature of compressed air after recuperating (^{o}C); T_ι = Surface temperature of fuel droplet (^{o}C); $Y_{0\infty}$ = Mass fraction of oxidant in surrounding; Q = Heat of reaction ($^\circ$C); i = Stoichiometric mixture ratio; Cp = Specific heat of liquid fuel droplet (kJ/kg.K),

and

$$\lambda_{eff} = \frac{8\kappa}{\rho_\iota Cp}\ln(1+B) \tag{4}$$

Where κ = Thermal conductivity of gas around the droplet (W/m.K); ρ_ι = Fuel density (Kg/m^3); Cp = Specific heat of gas around the deplete (kJ/kg.K); B = Heat transfer number.

Performance and Emission Tests for Micro Gas Turbine

The performance test of biofuel blend with distillate diesel was conducted in a liquid fuelled micro turbine (Capstone Model C30) at UNITEN Gas Turbine laboratory. A schematic plant-layout of the micro gas turbine engine and accessories are shown in Figure 1. The micro turbine generator performance test for this study was measured by thermal efficiency and specific fuel consumption. The performance test was conducted at ambient temperatures of 23.9 °C to 24.2 °C, achieved by using the cooling equipment. This is because the power output is affected by ambient temperature; hence, conducting the test at the lowest ambient temperatures is recommended to achieve the highest power output. The micro turbine was calibrated in accordance with the manufacturer correction curves during commissioning prior to the testing. The micro turbine was started and run for 30 min using diesel to warm up the equipment before switching to the designated test fuel when the micro turbine has reached the steady-state condition. This steady-state condition was achieved when the temperatures of the exhaust gases, mass flow rate, and electric power output reached a stable reading. Then, the power output was varied from idle to 25kW with intervals of 5kW, whilst ensuring that steady state conditions were reached before measurements were recorded.

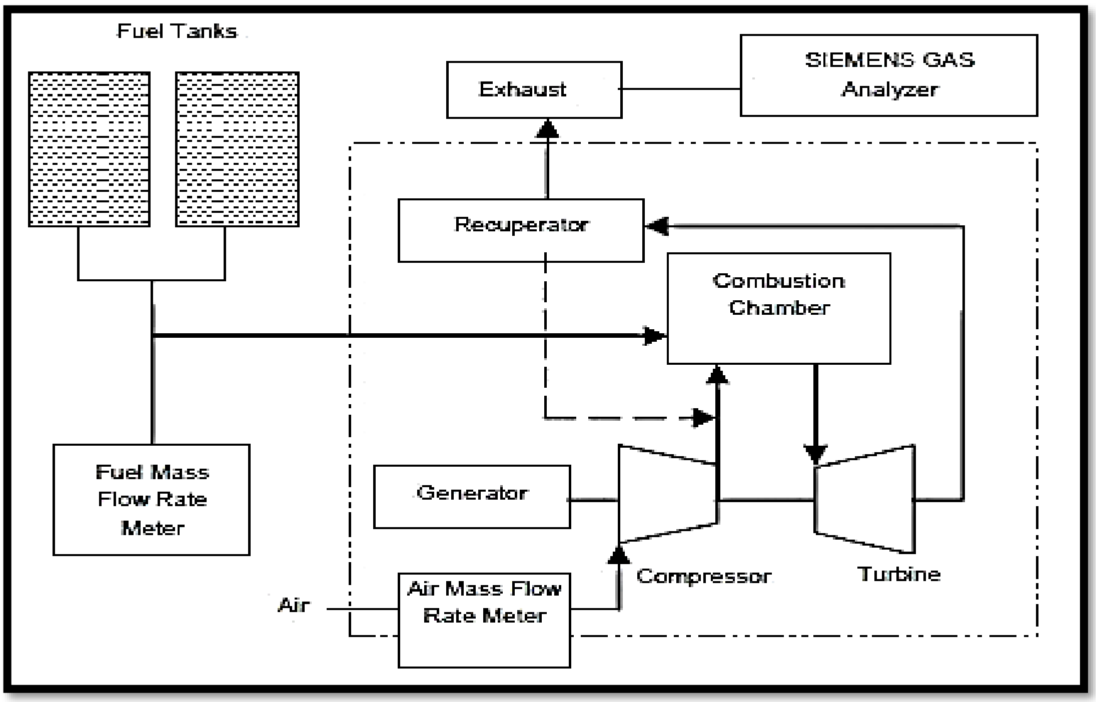


Figure 1. A schematic plant-layout of the micro gas turbine engine.

Meanwhile, the test fuels were also evaluated accordingly with a continuous emission monitoring system (CEMS) to evaluate the emissions of the micro turbine. CEMS was used at a sample port on the exhaust stack of the micro turbine for emission monitoring. The data for emission exhaust gases such as carbon CO, CO_2, O_2, and NO_X were recorded at 5 s intervals. Prior to use, the instrument analyser was calibrated periodically with an available sample with known quantities of gas. The entire performance test was conducted in accordance with calculations of Standard [39]. The following equations were used for the calculation of thermal efficiency and brake-specific fuel consumption respectively. Thermal efficiency was derived from the formula shown in Equations (5) to (8).

$$\eta = \frac{\mathrm{W_{net}}}{\mathrm{HI}} \tag{5}$$

where W_{net}= Generator power obtained from the data acquisition system (KW); HI = Heat input (KW). Meanwhile, heat input was derived from Equation (6).

$$\mathrm{HI} = \mathcal{Q}_{\iota}\rho_{\iota}\left(\mathrm{LHV_p}\right) + \mathrm{SH_p} \tag{6}$$

where $\mathcal{Q}_{\iota}$ = Volumetric flow (Litre/sec); ρ_{ι} = Density of the liquid fuel at operating temperature (kg/m^3); LHV_p =Lower heating value at constant pressure (kJ/kg); SH_p =Sensible heat at constant pressure (kJ/sec).

From Equation (7), the lower heating value can be obtained. The higher heating value can be calculated from equation (8):

$$\mathrm{LHV_p} = \mathrm{HHV}_{\mathcal{V}} - 91.20(\mathrm{H}) \tag{7}$$

where $HHV_{\mathcal{V}}$ = Higher heating value at constant volume in accordance with ASTM D4809 (kJ/kg); H = Percentage of hydrogen by weight contained in the liquid fuel and determined in accordance with ASTM D1018.

$$\mathrm{SH_p} = \mathcal{Q}_{\iota}\rho_{\iota} - (\mathrm{h_\iota} - \mathrm{h_{\iota,77}}) \tag{8}$$

where h_ι = Specific enthalpy of the liquid fuel at operating temperature of 27.38°C (kJ/kg); $h_{\iota,77}$ = Specific enthalpy of the liquid fuel at a standard operating temperature of 25 °C (kJ/kg).

The brake-specific fuel consumption (BSFC) from the micro turbine system was derived from Equation (9).

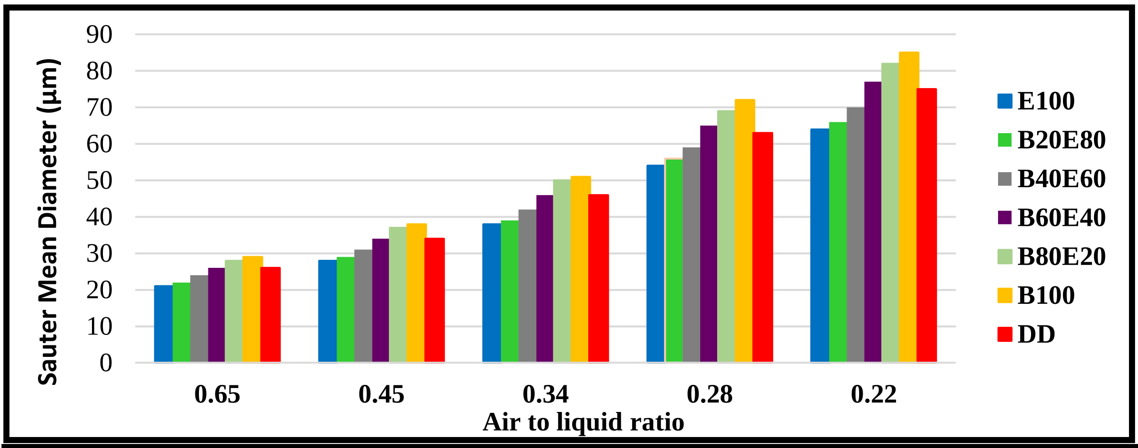
$$\mathrm{BSFC} = \frac{\text{Average Mass Flow Rate of Fuel}}{\text{Average Load}} \tag{9}$$

RESULTS AND DISCUSSION

Sauter Mean Diameter Analysis of Bioethanol Blends with Biodiesel

The major goal of atomisation is to increase the surface to volume ratio to enhance liquid evaporation and combustion efficiency. The biggest requisite for atomisation is that a relative velocity between the liquid to be atomised and surrounding air is high. One way to obtain this is by inserting moving liquid on a high-velocity airstream. Among the many methods of atomisation, the micro turbine deploys air blast atomising for its fuel combustion system as the air used to atomise the liquid promotes a good blend and better atomisation. SMD is the most widely used parameter to define the droplet size in a spray. Due to cost constraints, an SMD formula generated from Lefebvre correlation was adopted to numerically evaluate the SMD of bioethanol and its blends with biodiesel fuel, as a similar correlation was previously used to evaluate SMD of diesel fuel in Capstone C30 micro gas turbine with air blast atomiser, which has been experimentally validated using the phase doppler anemometer (PDA), air to liquid ratio (ALR), and relative co-flowing velocity exiting (UR) value adopted from a report by Bolszo [38].

Based on Figure 2, using Lefebvre equation, the patterns of all fuels were identical where SMD decreased gradually as the atomisation air to liquid mass flow ratio ALR spanned from 0.2 to 0.65. In a prompt atomisation process, air velocity, ALR, and fuel properties such as surface tension and density play primary roles, while viscosity takes a reduced role [40]. In general, test fuels have smaller droplet size at a high air velocity injection. At the low values of ALR, the kinetic energy of the atomising air was insufficient to overcome the viscous and surface tension forces which opposed the disintegration of the liquid. As ALR increased, it was evident that not only the droplets were quite smaller but also the difference between the largest and smallest droplets decreased significantly. In fact, for the highest values of ALR shown in Figure 2, there was a difference between the droplet sizes of the entire fuels. However, as can be anticipated, the highest atomising air velocities result in the finest atomisation. The biodiesel had a higher SMD value for all cases, though at the highest velocities produced an average SMD larger than the other fuels. Yoon, Park [32] in their investigation revealed that the measured results of biodiesel-bioethanol blended fuel showed SMD decreased with the increase of the relative velocity between the injected fuel and ambient gas. In conclusion to their work, the atomisation performance of test fuels was remarkably affected by the difference of relative velocity. Therefore, BE10 and BE20 have a smaller SMD distribution compared with ultra-low sulphur diesel (ULSD). The SMDs calculated using Lefebvre equation for all sample fuels prepared is illustrated in Figure 2. The results exhibited in Table 1 show that bioethanol has the lowest surface tension, density, and viscosity compared to biodiesel and distillate diesel, while the most significant reduction was found for viscosity.



E100	Pure Bioethanol	B80E20	80% Biodiesel/20% Bioethanol
B20E80	20% Biodiesel/80% Bioethanol	B100	Pure Biodiesel
B40E60	40% Biodiesel/60%Bioethanol	DD	Pure Distillate Diesel.
B60E40	60% Biodiesel/40%Bioethanol		

Figure 2. Effect of air to liquid ratio on SMD using different blends of fuel (bioethanol and distillate diesel)

Pure biodiesel fuel (B100) has the largest SMD, followed by B80E20, B60E40, DD, B40E60, B20E80, and pure bioethanol. SMD of biodiesel blends was much larger when compared to diesel because of the higher value of viscosity and surface tension of biodiesel [41]. When bioethanol was added, the blended fuels gave smaller droplets and when the blending ratio of bioethanol increased, the diameter became even smaller. This is because the addition of bioethanol led to a more active breakup process. This indicates that the addition of bioethanol reduces the droplet size and enhances the mixture formation. The higher viscosity and larger surface tension of biodiesel inhibited its atomisation process, and as bioethanol increased, the viscosity and surface tension of the blended fuel decreased (Table 1). Referring to Table 1, the higher ratio of bioethanol in a fuel will correspond to the lower density, viscosity, and surface tension. This will favour the break of droplets due to the reduced resistance to shear stress and thus, better atomisation performance was achieved.

Table 1 Physical property of test fuel used for atomisation assessment

Fuel Samples	Density (kg/m^3)	Viscosity (mm^2/s)	Surface Tension (mN/m)
E100	811.0	1.19	22.30
B80E20	860.0	4.26	31.32
B60E40	854.0	3.37	28.88
B40E60	845.0	2.18	25.93
B20E80	837.0	1.36	24.11
DD	845.0	3.95	23.00
B100	874.0	4.60	34.00

It is interesting to note that distillate diesel (DD) and blended 60BE40 fuel had a similar tendency of droplet size over the entire range of measurement. This was consistent with the study by Guan, Tang [42] who characterised the diameter size of biodiesel (B100) and diesel by the particle/droplet image analysis (PDIA) technique. They examined that the diameters for the fuels were 25 µm and 20 µm. However, when di-n-butyl ether (DBE) was blended into biodiesel with a volume fraction of 15% and 30%, the diameters decreased to 21 µm and 16 µm, respectively. Likewise, Yoon et al. [18] concluded that by adding up to 20% (by volume) of bioethanol into biodiesel (BE20), the droplet size of BE10 and BE20 was smaller than of diesel. The smaller the size of fuel drops in the air-fuel mixture, the faster the air-fuel mixture evaporates in the cylinder and the higher the combustion velocity of the air-fuel mixture [43]. This affirms that the improvement in physical properties by blending with different biofuels which are less viscous can reduce SMD of different percentage ratios of biodiesel compared with pure biodiesel and lead to better combustion efficiencies.

Droplet Evaporation Time (DET) of Bioethanol and Biodiesel

DET is another crucial element that influences the combustion efficiency and its dependence on the droplet size of the fuel. Decreasing the size of the fuel drops decreases evaporation time of the air-fuel mixture and therefore, will increase the combustion velocity of the air-fuel mixture [44]. The evaporation time of droplets was analysed using effective evaporation time constant and the droplet lifetime was determined using the D^2 law. According to the experimental data by Bolszo [38], when using ideal premixing and pre-vaporization at 30kW load, the Capstone C30 required 11 milliseconds (ms) for the droplets with SMD 50µm of diesel to be fully vaporized corresponding to the value (0.28) of ALR. Hence, in order to determine the evaporation time of neat bioethanol and biodiesel, experimental data from Bolszo's work were adapted for this analysis to estimate the required evaporation time for the droplets (to be fully vaporized) in the micro turbine. Table 2 shows DET of bioethanol, biodiesel, and distillate diesel, respectively. It was observed that at lower ALR, (same operation condition as Bolszo's work), the biodiesel and diesel required 21 and 18 ms, respectively to be fully vaporized, whereas bioethanol required 23 ms for a 54 µm droplets, which require a much longer time to evaporate.

Chong and Hochgreb [45] reported that fuel droplets with higher SMD contain high momentum and thus have extended evaporation time. However, the results of this research showed that decreasing the size of the bioethanol drops increased the evaporation time of the air-fuel mixture. This is in contrast for the liquid fuel like biodiesel and diesel; although their droplet size was bigger, the evaporation time was lower compared with bioethanol. This phenomenon is due to the very high evaporation enthalpy of bioethanol, causing the droplets evaporation rate to be limited. Thus, bioethanol behaved much more like a high boiling point fuel compared to biodiesel and diesel. As can be seen from Table 3, the latent heat of vaporization of bioethanol (840 kJ/kg) was the highest, which was three times higher than that of biodiesel and diesel (230 and 250 kJ/kg, respectively). Consequently, more heat energy is required to evaporate a sufficient amount of fuel to make a combustible fuel to air mixture. This issue is more severe for bioethanol as compared to biodiesel and diesel due to its higher vaporization energy.

Table 2. DET of bioethanol, biodiesel and distillate diesel.

ALR	Bioethanol (E100)		Distillate Diesel (D100)		Biodiesel (B100)	
	SMD (µm)	DET (ms)	SMD (µm)	DET (ms)	SMD (µm)	DET (ms)
0.65	21	4	26	3	29	4
0.45	28	6	34	5	38	6
0.34	38	11	46	9	51	11
0.28	54	23	63	18	72	21
0.22	64	33	75	25	85	29

Table 3. Physical properties of fuels that affect the rate of evaporation time.

Fuels	Latent Heat of Evaporation (kJ/kg)	Specific Heat of Fuel Droplet (kJ/kg.K)
E100	840	2.55
DD	250	1.85
B100	230	1.90

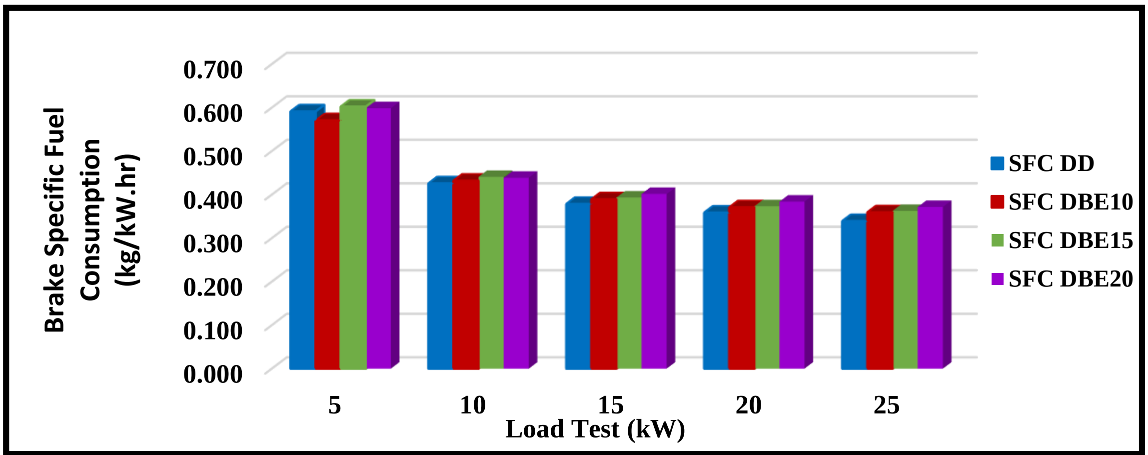
The experiment results obtained in this study were supported by the work of Benjumea, Agudelo [46], who stated that the lower heat of vaporisation of palm methyl ester of approximately 200~220 kJ/kg had faster evaporation compared to diesel which had 375 kJ/kg latent heat of vaporisation at the same operating condition. Similarly, Zhang, Xu [47] reported that the evaporation of methanol and bioethanol fuels was much slower than that of gasoline because of their lower vapour pressures and higher latent heats of vaporization. Bagul AD [48] stated that the vaporization of bioethanol blends requires more heat input than needed to vaporize the same mass of gasoline. The lower vapour pressures and higher latent heats of vaporization are still the challenges for the evaporation of alcohol fuels because inadequate vaporization of the fuel can lead to an increase in hydrocarbon emissions. Iranmanesh [49] reported that the higher heat of evaporation of the bioethanol or diethyl ether in the fuel blends tends to produce slow vaporization and poorer fuel-air mixing which subsequently produces incomplete combustion efficiency of the mixture.

Specific heat capacity is another critical factor that influences the evaporation rate. Bioethanol has a higher specific heat than biodiesel and diesel, which means that it requires higher energy to raise the temperature of the liquid fuel. The results from this work is in a good agreement with [50, 51] where they reported that the specific heat of the bioethanol fuel is higher than that of pure gasoline and this leads to the decrease in the drop of the charge temperature. Hence, more addition of bioethanol decreases combustion temperature (caused by its increased specific heat) in contrast with biodiesel and diesel. It can be concluded by using direct proportional assumption (as shown in Table 3) that the increase in bioethanol percentage ratio in blended fuel could lead to prolonged ignition delay period and reduced combustion flame temperature. This is explained as mentioned earlier by the higher latent heat of vaporization and specific heat value of bioethanol, which are considered as the most influential factors [52]. Based on the results, it is recommended that the micro gas turbine operation is limited to up to 20% blend of bioethanol so as to achieve a balance of good droplet size and combustion efficiency characteristics.

Micro Gas Turbine Engine Performance Characteristics Evaluation

The analyses of performance and emissions characteristic of micro gas turbine fuelled with biofuel (Biodiesel-Bioethanol) blended with distillate diesel and using load ranging from 5kW to 25kW were carried out. The comparative parameter to determine the efficiency of conversion of fuel into work to power the micro gas turbine was measured by brake specific fuel consumption (BSFC). Fuel consumption is a measure of the volumetric fuel consumption for any particular fuel and depends on a number of parameters, namely the calorific value. In general, the brake thermal efficiency is simply the inverse of the product of fuel consumption and the lower calorific value of the fuel. Figure 3 indicates the variations of the BSFC for different diesel-biodiesel-bioethanol blended fuels ratio under various engine loads. The brake specific fuel consumption trends for diesel and the blends are similar in nature. The results showed that increasing bioethanol proportion in the fuel blend increased the BSFC. This behaviour is attributed to the heating value per unit mass of the bioethanol (31825 kJ/kg), which was noticeably lower than that of the diesel and biodiesel fuels (45088, 40023 kJ/kg, respectively). Yilmaz [53] studied the performance and emission of biodiesel-diesel-ethanol blends (B45E10D45, B40E20D40) in a diesel engine at different load conditions. They found that the use of ethanol in the biodiesel-diesel blend showed higher fuel consumption than that of diesel fuel. From Figure 3, it can be seen that BSFC for the blend fuel DBE10 was the most comparable ratio to neat diesel (DD) for all loads tested. This is because of the high heating value of the blend in comparison to the BSFC of DBE15 and DBE20. These results agree with those found by other authors [54, 55].

The brake specific fuel consumption is greater at smaller loads, but it decreases at medium and higher loads. For the same loads, the bioethanol blends exhibited higher consumption due to lower heating values (meaning less energy content than pure diesel fuels). Different properties of the test fuels significantly affected the brake thermal efficiency of the engine. The higher thermal efficiency in turn helped to achieve better combustion efficiency and lower emissions correspondingly. The variation of brake thermal efficiency for pure diesel and its blends of up to 20% of biodiesel-bioethanol for the low and high loads are shown in Figure 4. The brake thermal efficiency for the 20% ratio (80:16:4 of diesel-biodiesel-bioethanol) fuel was found to be 21.92% (at test load of 20kW); which was very close to the value of pure diesel (22.87%) at a similar load. It should be noted that an increase in the thermal efficiency was observed in spite of the fact that the calorific values of the blends were lesser than the values for pure diesel. The increase in the thermal efficiency can be attributed to the addition of the oxygenated additive, which decreased the viscosity of the mixture, improved the atomisation and fuel vaporization and thereby enhanced the combustion efficiency of the fuel to a greater extent. Besides that, the thermal efficiency of blends was also improved due to faster burning of bioethanol in the blend (An increase in the rate of heat release due to rapid combustion of bioethanol by flame propagation). These results agreed with those found by other authors [56, 57].



Fuel blending description	Diesel (D)	Biodiesel (B)	Bioethanol (E)	%Vol of Biofuel in blends [(B) + (E) / (Total Vol)]
DBE20	80	16	4	20%
DBE15	85	12.75	2.25	15%
DBE10	90	9	1	10%
DD	100	0	0	-

Figure 3. Brake specific fuel consumption at various load using different fuels blends

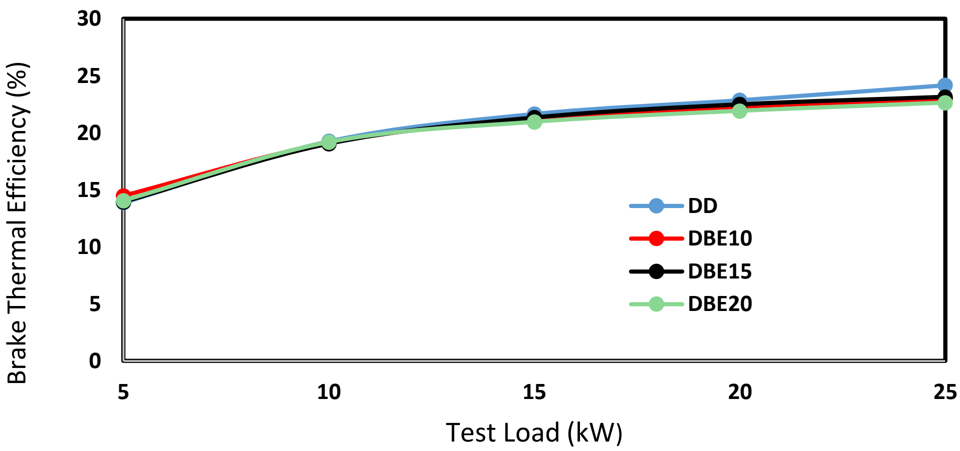


Figure 4. Variation of brake thermal efficiency of the micro gas turbine at different loads using various fuel blends.

At the maximum test load of 25kW, it was observed that the thermal efficiency of the blend was marginally lower than diesel. Thermal efficiency was 22.99%, 23.16%, and 22.64% for DBE10 (90:9:1 of diesel-biodiesel-bioethanol), DBE15 (85:12.75: 2.25 of diesel-biodiesel-bioethanol), and DBE20 (80:16:4 of diesel-biodiesel-bioethanol), respectively. The slight variations in the thermal efficiency of the DBE blends were mainly due to the lower calorific value of bioethanol when compared to diesel and biodiesel. The results of this work concurred with Krishna, Bandewar [58] who found the brake thermal efficiency of the blend was 26.73% as compared to 23.21% of Karanja oil and 27.01% of pure diesel. Similar results were also reported by Anand R [59], who showed that at 100% load condition, the maximum brake thermal efficiency of biodiesel-bioethanol blends (B90E10) was higher than that of B80E20 and lower than that of diesel fuel. Thus, it can be concluded that most works have reported that the thermal efficiency of the fuel blends is marginally lower than diesel at the maximum engine power output.

Exhaust Emission Comparison of Micro Gas Turbine Operating on Biofuel Blends with Distillate Diesel

Renewable bio-fuels also emit pollutants that are equally detrimental to the environment and specifically dependant to the combustion dynamic of a particular engine. CO_2 is one of the main combustion products which are very important in determining the completeness of a combustion reaction of the fuel. The variation of CO_2 with various loads for diesel fuel and diesel-biodiesel-bioethanol blends is illustrated in Figure 5.

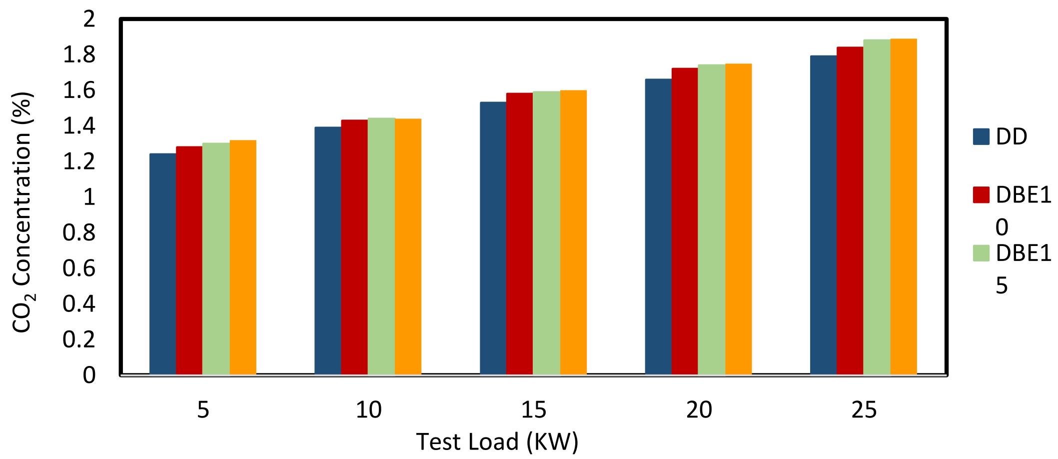


Figure 5. Variation of CO_2 emissions from micro gas turbine at different test loads for different fuel blends

It was found that as the load increased, CO_2 emissions increased as more and more fuel was burned at high load while complete combustion efficiency was achieved at the high loads due to high temperature. This trend was similar for all fuel blends. The CO_2 emissions of DBE10, DBE15, and DBE20 (at the highest test load of 25 kW) were higher (1.84%, 1.88%, and 1.89%, respectively) than those of diesel fuel and increased with the increase of bioethanol percentage. This was due to the complete combustion efficiency caused by the presence of highly oxygenated bioethanol supply, thus the emission of CO_2 increased with the increase in bioethanol percentage of blends. These results were in agreement with Cheenkachorn and Fungtammasan [60] and Subbaiah, Gopal [61] who showed that at the high engine speed and load, the CO_2 emissions increased as more and more fuel burned more excess air. They also observed that the CO_2 concentrations emissions from all biodiesel-bioethanol-diesel blend fuels were higher than that of diesel fuel. Deshpande SS [62] stated in their research that for the blended fuel of DBE10, the CO_2 emissions were higher at all loads and maximum increase was 66.37% as compared to the pure diesel. However, they reported that for the blend DBE20, the CO_2 emissions were lower at all loads in comparison with the fuel DBE10. Their result was in contrast with this study, which showed that DBE20 emitted more CO_2 emissions for all test loads compared to DBE10, and DBE15.

Another important emission gas is carbon monoxide (CO). During the complete combustion efficiency, the conversion of CO into CO_2 takes place whereas if the combustion efficiency is incomplete due to shortage of air or the low gas temperature, more CO will be formed. Formation of CO indicates loss of power, resulting in oxygen deficiency in the combustion chamber [63]. The variation of CO with loads for different fuels is shown in Figure 6.

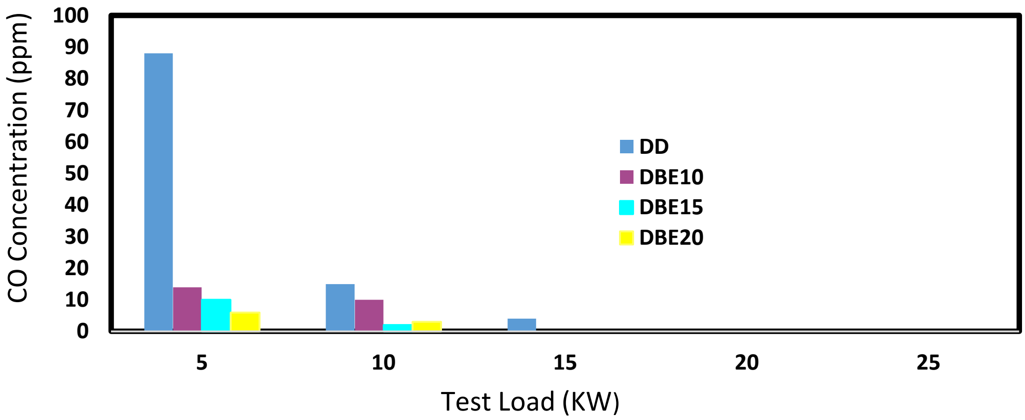


Figure 6 Variation of CO emissions from micro gas turbine at different test loads for different blended fuels.

Emissions of CO from engine mainly depend on the physical and chemical properties of the fuel. The CO emissions of the diesel-biodiesel-bioethanol blend fuels were much different from that of conventional diesel at the low loads as shown in Figure 6. However, the CO emissions slightly increased at the low loads and decreased significantly at the higher loads with all the fuel modes. The report by Hulwan and Joshi [54] indicated that CO emissions were drastically increased at the low loads using the high percentage of bioethanol in diesel–bioethanol blends. They reported that the drastic increase in the CO percentage at the low load for the blend was due to the decrease in the cylinder gas temperature and delayed combustion efficiency process, even though enough oxygen was available for the combustion efficiency process. The reduction in CO emissions was noticed for blends at the high load due to the high temperature and enrichment of oxygen owing to the bioethanol addition, in which an increase in the proportion of oxygen will promote further oxidation of CO during the engine exhaust process [59-64].

The experimental results of Barabas, Todoruţ [65] showed that at the high engine loads, the lowest CO emission (0.234 % vol.) was for the biodiesel 10%-diesel 85%-ethanol 5% (B10D85E5) mixture. This compared to the one seen in the diesel fuel case (0.575% vol.) represented a 59% reduction. In the previous works by several researchers, it was suggested that the higher oxygen content of the blended fuels could improve the combustion efficiency process while the lower viscosity and density of the blended fuels could lead to better air-fuel mixing, which can sufficiently burn all fuel, resulting in the lower CO emissions [52, 66, 67].

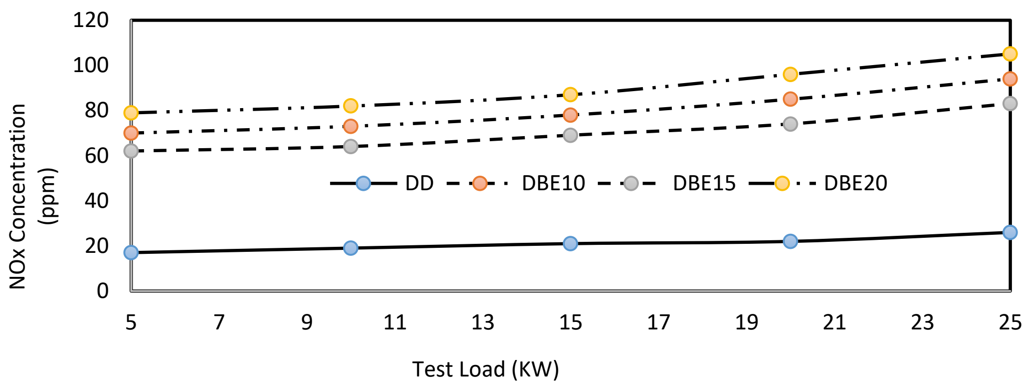


Figure 7. Variation of oxides nitrogen emissions from the micro gas turbine at different test loads for different blended fuels.

The most troublesome emissions from engines are NO_x. It is produced during the combustion efficiency process when nitrogen and oxygen are present at elevated temperatures. The oxides of nitrogen in the exhaust emissions contain nitric oxide (NO) and nitrogen dioxide (NO_2). The formation of NO_x is highly dependent on the combustion temperatures, oxygen concentration, and residence time for the reaction to take place [68]. The variation of nitrogen oxides with loads for diesel fuel and diesel-biodiesel-bioethanol blends is presented in Figure 7. It can be observed that NO_x emitted by all fuels blends are higher than the ones for the corresponding pure diesel fuel case. The NO_x emissions of diesel-biodiesel-bioethanol blends were lower at the low loads and higher at medium and high loads than those of diesel fuel. This was due to the higher combustion temperature as well as the oxygen content of the bioethanol at medium and high loads versus the diesel fuel. The NO_x emissions of DBE10, DBE15, and DBE20 were 94%, 83%, and 105% higher compared to those of the diesel at full load (25 kW) of the engine. Generally the higher oxygen content results in the higher combustion temperature which leads to the higher NO_x emission. This result was comparable with Mofijur, Rasul [69] who also found less influence of oxygenated components of the fuel blends in the NO_x formation at smaller loads. Nevertheless, at the medium and high engine load conditions, the NO_x emission increased by 10–26% compared to diesel fuel. Most of the studies on this fuel concluded that the higher oxygen content and low viscosity in bioethanol fuel can lead to better mixing, improve combustion efficiency and rise the combustion chamber temperature, which contributes to the higher formation of NO_x emissions [54, 70]. Thus, with the increase of bioethanol in the blended fuel, NO_2 emission increased correspondingly at the high load engine. Opposite results were also observed by several considerable studies that developed new methods for the reduction of NO_x emission from the diesel engine by using selective catalytic reduction technology. An experiment research by Xiaoyan, Yunbo [70] used three types of catalyst for NO_X emission reduction. They observed approximately a 5.5% increase in NO_x emission from the diesel–biodiesel–ethanol blend without any catalyst assembly. However, when they used the Ag/Al_2O_3 catalyst, NO_x was reduced by 73%. Again, when the exhaust was passed through the Ag/Al_2O_3+Cu/TiO_2 catalyst and Ag/Al_2O_3+Cu/ TiO_2+Pt-supported catalysts, the reduction was 71% and 61% respectively at the high load engine. The same trend was obtained in the research published by Baskar and Kumar [71], who studied the effect of oxygen concentration in the intake air and diesel-water emulsion as fuel for combustion, performance and emission characteristics for a direct injection diesel engine. A reduction in NO_x emission was observed in their work due to the reduction in combustion chamber temperature as the water concentration increased.

Another gas found in the emissions obtained in the combustion exhaust during the experiment was oxygen. Oxygen (O_2) was not perceived as a pollutant in this aspect. The necessity to examine the concentration of O_2 in the exhaust was important to establish the benefits of bioethanol as being a carbon neutral and oxygenated fuel. The graph in Figure 8 presents the O_2 emission in the exhaust gas, which shows that the concentration was the lowest in distillate diesel (DD) and high with respect to the increasing bioethanol volumetric ratio in the blending. The O_2 concentration in the exhaust stream was pretty much stable and ranged from 17.95 to 19.25% from the high to low load input. Generally, the O_2 emissions increased with the higher amount of bioethanol in diesel-biodiesel-bioethanol blended fuel as compared with the pure diesel. The results indicated that the O_2 levels did not have any drastic change. In the lean and stoichiometric conditions where the amount of air is enough to sustain complete combustion efficiency, the levels of O_2 are abundant, and the presence of additional O_2 atoms in the bioethanol is directly noticed [59]. From Figure 8, it can be noticed that the oxygen concentration emissions were reduced with load for all the fuels modes. This was due to the fact that as the load increased, more O_2 was used for complete combustion efficiency. The O_2 emissions were reduced by 18.49%, 18.35%, and 18.36%, respectively with the blends DBE10, DBE15, and DBE20 compared with the diesel fuel at the full load of engine (25kW).

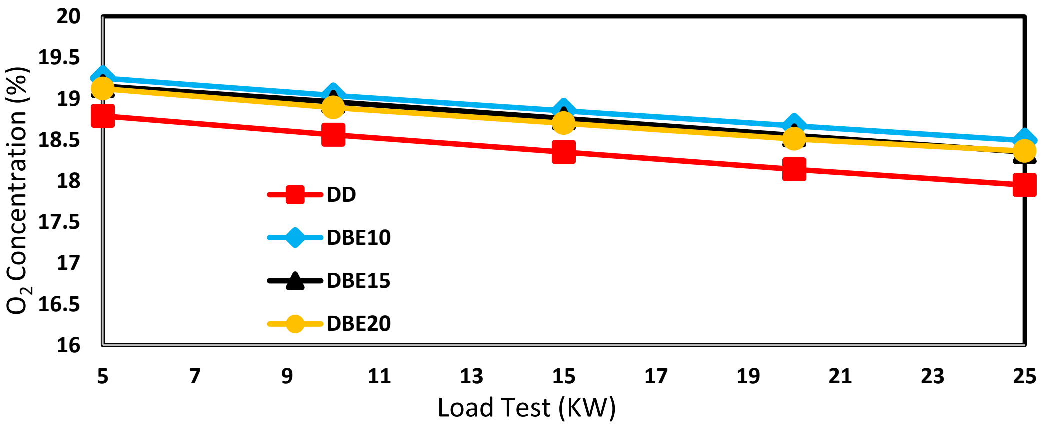


Figure 8. Variation of oxygen emissions from the micro gas turbine at different test loads for different blended fuels.

CONCLUSIONS

Renewable energy utilisation for power generation is still not wide spread and more data is needed on the performance of biofuel for gas turbine engines. The current study provided new information regarding the use of biofuels in micro turbines. The physicochemical characteristics of all the biofuels used must lie within the specifications for their use in micro gas turbines. Recent reports on the use of biodiesel in micro gas turbines have described problems associated with viscosity and density. In addition, the current study on the use of bioethanol in micro gas turbines has also demonstrated the drawbacks related to the increase in evaporation droplet time, which was explained in the part of the high latent heat of evaporation of ethanol. Hence, the blend ratio B80E20 (80%

biodiesel-20% bioethanol) indicated by this research work was the most ideal blended fuel ratio to be applied in the micro gas turbine engine. This blend can replace diesel fuel, and therefore a 100% biofuel can be used in the existing gas turbine engines without the need of engine modifications. The results of this study are important to establish the limits of biofuel properties essential for the utilization in gas turbine application. Subsequently, the emissions test reported in this work also showed significant enhancement in emissions. This study has therefore shown that better returns that can be gained with the integration of the production of biodiesel and bioethanol by turning the waste glycerol, a by-product of biodiesel production, into fuel like bioethanol. Finally, it is worth mentioning that this study is among the few that has provided valuable data on the usage of biofuel for power generation using gas turbine. The usage of biodiesel and bioethanol (produced from waste glycerol) for power generation in micro gas turbine engine will also help to defray the cost of biodiesel production. Further work is necessary to evaluate the preliminary atomization in terms of the spray length and angle using thermal imaging for various blend fuels to mimic the actual condition that occurs in gas turbine prior to gas turbine application.

ACKNOWLEDGMENTS

The authors would like to thank the Malaysian Ministry of Education for valuable financial support through the Exploratory Research Grant Scheme No. 12012013. We also acknowledge the Sime Darby Malaysia, for supplying the (crude glycerol, biodiesel) and Universiti Tenaga Nasional for the research facilities.

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