

Article

Effects of Pig Manure and Corn Straw Generated Biogas and Methane Enriched Biogas on Performance and Emission Characteristics of Dual Fuel Diesel Engines

Salman Abdu Ahmed ^{1,*}, Song Zhou ^{1,*}, Yuanqing Zhu ¹, Asfaw Solomon Tsegay ²,
Yoming Feng ¹, Naseem Ahmad ¹ and Adil Malik ¹

- ¹ College of Power and Energy, Harbin Engineering University, 145 Nantong St, Harbin 150001, China; zhuyuanqing@hrbeu.edu.cn (Y.Z.); fengyongming@hrbeu.edu.cn (Y.F.); naseem_saddiqui@hrbeu.edu.cn (N.A.); adilmalik@hrbeu.edu.cn (A.M.)
- ² State Key Laboratory of Urban Water Resource and Environment, Harbin Institute of Technology (SKLUWRE, HIT), Harbin 150090, China; solomontse@hit.edu.cn
- * Correspondence: salman2014@hrbeu.edu.cn (S.A.A.); songzhou@hrbeu.edu.cn (S.Z.)

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Abstract: In recent years, due to stringent emission regulations vehicle manufacturers have been compelled to cut down noxious pollutants released from diesel engines. Different alternative solutions have been recommended to achieve this challenging task. One of these alternative solutions is the utilization of biogas in addition to the use of liquid diesel. In this regard, the current study investigates the combustion characteristics and exhaust emissions of a turbocharged, direct injection, diesel engine operating at constant speed (1800 rpm) and under dual fuel mode with diesel as the pilot fuel and biogas (generated from pig manure and corn straw) and methane enriched biogas. Simulations were carried out at four various engine loads corresponding to brake mean effective pressure (BMEP) of 0.425, 0.85, 1.275, and 1.7 MPa using GT-Power package. The BTE values of biogas-diesel were higher as compared to diesel fuel. The CO₂ ratio of biogas did not impact BTE considerably. The highest BTE value of 38.22% was recorded for BG45. However, the Brake specific fuel consumption (BSFC) values for the biogas-diesel fuels were higher than that of diesel fuel operations. With respect to emissions, compared to diesel fuel operation, the hydrocarbon (HC) and CO₂ of the biogas-diesel were higher, but NO_x and CO pollutants were much lower. The utilization of biogas with diesel by all accounts is attractive to cut down discharges and improve performance of the engine. The engine performance did not deteriorate with up to 45% CO₂ proportion in biogas.

Keywords: biogas; biogas-diesel; diesel engines; emission; performance

1. Introduction

Due to their greater thermal efficiency, diesel engines play a key role in the world's transport and industrial facilities [1], in particular in heavy-duty systems such as lorries, buses, building and farm facilities, locomotives, vessels, etc. [2]. Nonetheless, these engines emit greenhouse gases (GHG) pollutants such as methane (CH₄), carbon dioxide (CO₂), and carbon monoxide (CO) and have high pollutants of nitrogen oxides and smoke which results in a severe health hazards [3]. The fossil fuels used by internal combustion (IC) engines add to the global environmental deterioration due to their GHG pollutants. Besides, due to the restrictions of reserves, supply, and higher demand for petroleum fuels induced by industrialization, prices of the fossil fuels are continually rising. Moreover, laws of diesel engine emissions have also been tightened [4]. Renewed interest in renewable alternative fuels

has been fostered due to the increased energy use, increasing prices of oil, diminishing oil reserves, and environmental pollution issues.

Many studies have been carried out to enhance combustion features so that engine effectiveness is maximized by reducing fuel consumption and dangerous emissions, in order to comply with harsh emission rules and regulations, increasing energy demands, and decreasing non-renewable fuel. One of the most attractive techniques for improving fuel efficiency and for reducing environmental emissions is the use of biofuels [5]. Biofuels such as biogas, alcohol, and biodiesels [6] have been given significant consideration due their renewable trend and their ability to reduce net carbon dioxide emission [7]. Ge et al. [8] used palm oil biodiesel blends to improve performance and investigated emission characteristics and their formation. Jamrozik et al. [9] found that compressed gas burned with diesel fuel in a dual fuel engine greatly reduces CO₂ and CO pollutants. Using 500 mg/L carbon nanotube mixed with diesel fuel Markov et al. [10] reduced smoke emissions by 57% at full load condition. In another study Cong Ge et al. [11] using 20% of canola oil biodiesel fuel mixed with 80% of diesel fuel reduced emissions of CO, HC, and particulate matter (PM) at various engine loads and constant engine speed of 1500 rpm. Biofuel production contributes to improving local farming, reducing reliance on oil imports, and reducing emissions. In comparison to fossil fuels, biofuel production is spread more equally and can be produced at a minimum investment cost. For diesel engines, biodiesel, bio-alcohols, liquid biomass, and hydro-treated vegetable oils (HVO) are used as the partial substitute for conventional diesel fuel. Alcohols (for example, ethanol, methanol, and butanol) [12], vegetable oils, animal fats, biodiesel, biogas, and LPG are getting a great deal of interest among alternative fuels [13–17].

The broad ignition boundaries of gaseous fuels and their ability to create homogeneous mixtures made them attractive. Furthermore, the hydrogen to carbon proportion of gaseous fuels is large. Thus, when used in IC engines very small emissions are feasible [18]. Diesel engines can function at greater compression, enabling them to use alternative fuels such as biogas with low energy content [19]. Biogas (BG) is one such a fuel, which is a blend of gasses produced in the absence of air during biological disintegration of organic matter [20]. It is a carbon neutral gaseous fuel, bringing about no new expansion of ozone harming substances to nature. The quality of biogas in terms of composition varies depending on biomass, precursors, additives, and the conversion process. CH₄ is the primary portion of biogas, and the extent shifts from feedstock to feedstock. It is composed mostly of CH₄ (50–70%) and CO₂ (25–50%), with low fractions of H₂ (1–5%), N₂ (0.3–3%), and hydrogen sulfide (H₂S) traces [21]. Studies have indicated that biogas generated from organic waste represents a useful alternative to fossil fuels and can be utilized by CI engines as they are better able to mix with air and smooth burning nature [22]. Besides, the use of biogas offers alternative energy sources and protects the environment against damaging greenhouse gasses [23]. Biogas as a car fuel has major environmental benefit of significantly reducing greenhouse gas (GHG) emissions in the transport sector. Biogas powered cars may decrease CO₂ emissions by 75–200% compared with fossil fuels [24]. When fluid residues are used as organic fertilizers (subsequent to mineral fertilizers), it can prevent emissions of CH₄ by landfilling or storing manure and further save GHGs [25]. In the past few years there has been a great deal of research in the use of biogas in conventional spark ignition (SI) and compression ignition (CI) engines. Its high self-ignition temperature hinders it to be used directly in CI engines, however, it can be used in a CI engines in a dual fuel strategy [18,26].

Biogas combustion with diesel fuel can provide low emission combustion. Its low Cetane number being a gaseous fuel makes it suitable for CI engines in a dual fuel approach. The high anti-knock properties of biogas compared with regular fuel makes it an applicable fuel for dual-fuel engines. However, the presence of CO₂ reduces thermal efficiency due to prolonged ignition delay and decreased flame temperature [27]. CO₂ could also impact the combustion process (burning velocity), which might result in incomplete combustion prompting HC discharges and reduced engine efficiency [28]. Moreover, the CO₂ present has some negative influence towards some parameters such volumetric energy density, fuel conversion efficiency, and the combustion enthalpy [29]. CO₂

behaves as a non-reactive gas and impacts the burning velocity of the in-cylinder charge, thereby resulting in inadequate burning, that may cause the increase of brake specific fuel consumption (BSFC) [30]. Nathan et al. [31] reported that the CO₂ in the biogas smothered the high rate of release which is common in homogeneous charge compression ignition (HCCI) engines. However, they found improved efficiency and low levels of NO_x and soot. In another study, Makareviciene et al. [32] assessed the impact of CO₂ in biogas used in a diesel engine. Their findings revealed that lower emission levels were observed when the engine was run with the EGR system. Lounici et al. [33] reported that the high proportion of CO₂ present in biogas significantly reduces NO_x and PM. Owczuk et al. [34] found that biogas containing up to 50% CO₂ can be applicable as a tractor fuel.

A lot of studies have been carried out on the usage of biogas in a diesel engine in a dual fuel approach using biogas-diesel (BG-D), biodiesel-biogas (BD-BG), and diesel-biogas-hydrogen (D-BG-H). They have embraced different engines and fuel alteration to enhance the thermal efficiency and cut down the level of pollutants emanated from CI engines. A review of research works on the usage of biogas in CI engines are outlined and given in Table 1.

Table 1. Reviews of research work on the usage of biogas.

Researcher Reference	Engine Type/No of C	Fuels	Pressure	HRR	BTE	NO _x	HC	CO
E. Porpatham et al. [18]	4.4 kW at 1500 rpm/01	Biogas	↑	↑	↑	↑	↑	↑
Park et al. [27]	117.6kW SI gas Engine/06	N ₂ + Biogas	X	X	↑	↓	↑	X
Park et al. [27]	117.6Kw SI gas Engine/06	H + Biogas	X	X	↓	↑	↓	X
Ramesha et al. [35]	10hp/dual fuel/(01)	BD-BG	↑	↑	X	↓	↑	↑
Yoon and Lee [36]	46kW at 4000rpm/04	BG-D and BD-BG	↓	↑	X	↓	↑	↑
Makareviciene et al. [32]	66kW/dual fuel/04	BG-D	X	X	↓	↓	↑	↑
Bora et al. [37]	3.5kW/1500rpm/01	BG-D	X	X	↓	↓	X	↓
Mustafi et al. [19]	32.6Nm at 1800 rpm/01	NG + D and BG-D	↑	X	↑	↓	X	↑
Ambrita [38]	4.41kW at 2600rpm/01	BG-D	X	X	↑	x	↑	↑
Nathan et al. [31]	3.7kW at 1500rpm/01	BG-D	↓	↓	↓	↓	↑	↑
Ibrahim et al. [39]	18.24kW at 3600rpm/02	BG-D	X	X	↑	↓	↓	X
Yilmaz [40]	48kW/04	BG-D	↑	↑	X	NE	↓	X
Bora et al. [41]	3.5kW at 1500rpm/01	BD-BG	X	X	↑	↓	↑	↑
Zhang et al. [42]	10.3kW/01	BG-H	↑	X	X	↓	X	X
Karen et al. [43]	20kW at 3000rpm/02	BG-D	↑	X	↑	X	X	X
Pattanaik et al. [44]	3.78kW at 1500rpm/04	BD-BG	X	X	↓	↓	↑	↑
Barik et al. [29]	4.4kW at 1500rpm/01	BG-D	↑	↑	↓	↓	↑	↑
Bora et al. [45]	3.5kW at 1500rpm/01	BD-BG	↑	↑	↑	↑	↓	↓
Kalsi et al. [1]	7.4kW at 1500rpm/01	BD-BG	X	↓	↑	↓	↑	↑
Barik et al. [46]	4.4kW at 1500rpm/01	BD-BG	↓	X	↑	↑	↓	↓
Verma et al. [47]	4.4kW at 1500rpm/01	D-BG-H	↑	↑	X	↑	↓	↓

↑ = Increased ↓ = Decreased, NE = not affected, HRR = heat release rate, BTE = brake thermal efficiency, X = study not conducted, BD = biodiesel, BG = biogas, H = hydrogen, NG = natural gas.

As indicated by the research works above, biogas can be utilized as a fuel in CI dual-fuel engines without any engine modification. Moreover, these studies have demonstrated that biogas-diesel fuels improve the efficiency and reduce emissions. In spite of the fact that there are various investigations with biogas, they were focused on one, two, or four cylinder engines; there are few studies in the literature for the assessment of biogas in multi-cylinder turbocharged engines which use biogas generated from pig manure (PM) and corn straw (CS). As of now, published works in this field indicate limited outcomes on the impacts of biogas generated from PM and CS on engine performance and exhaust emissions of multi-cylinder engines. Thus, it is crucial to recognize such impacts so as to help in future engine design. This work showed improved engine performance and lower emissions using low grade biogas.

In perspective of the above setting, the target of the present work was to investigate the usage of biogas generated from PM and CS as well as enriched biogas to operate a six cylinder CI diesel engine under dual fuel approach with biogas (BG) as the primary fuel and diesel as the pilot fuel at various engine load conditions (0.425, 0.85, 1.275, and 1.7 MPa). The performance of the biogas-diesel fuels was compared to the reference diesel fuel.

2. Materials and Methods

The biogas used in this research was generated by co-digestion of pig manure and corn straw (197 g/L*D of PM and 19 g/L*D of CS of which 51% of VS (volatile solid) and 66% of TS (total solid)). A daily continuous load of 150 mL/day of mixed substrate raw material was conveyed to two continuous stirred tank reactor (CSTR) reactors (4.5 m³ working volume) and co-digested in anaerobic environment to produce biogas as shown in Figure 1a. The biogas flow rate from each reactor was measured by a digital gas flow meter which is connected to a computer. Agilent 7890A gas chromatograph (GC-7890A, Agilent Technology, Santa Clara, CL, USA) (Figure 1b) fitted with a thermal conductivity detector (TCD) and helium as the carrier gas was used to measure the CH₄-CO₂ content in the biogas. The CO₂ percentage in biogas was intermittently measured using a Fyrite gas analyzer (Bacharach Inc.) as per the method indicated by the maker.

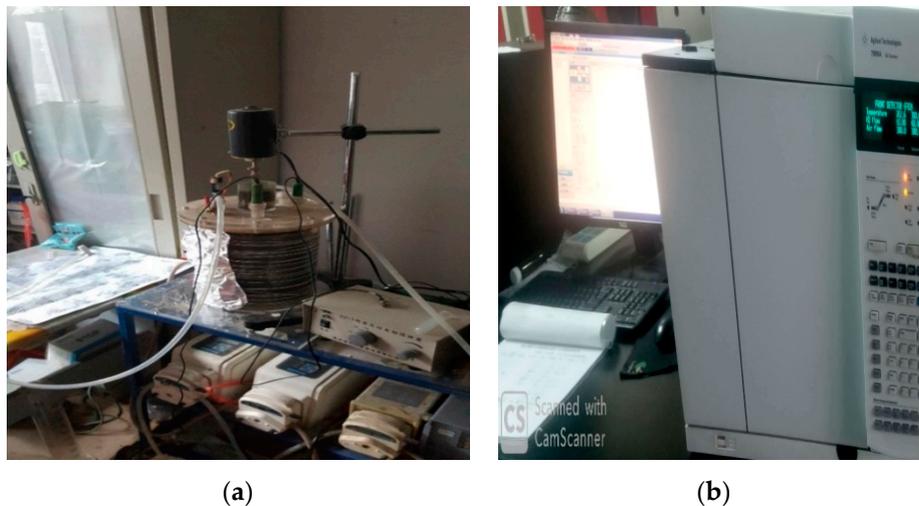


Figure 1. (a) Anaerobic continuous stirred tank reactor (CSTR) reactor; (b) gas chromatography machine.

Figure 2a depicts the daily biogas output with respect to the digestion time. The average everyday biogas generation during the 110 day digestion time was observed to be 2000 mL/day per kg of PM (pig manure) and CS (corn straw). Figure 2b depicts the percentage of CH₄ and CO₂ over 110 days of the digestion period. We were able to produce a 0.06 m³ biogas from 1 kg of pig manure biomass.

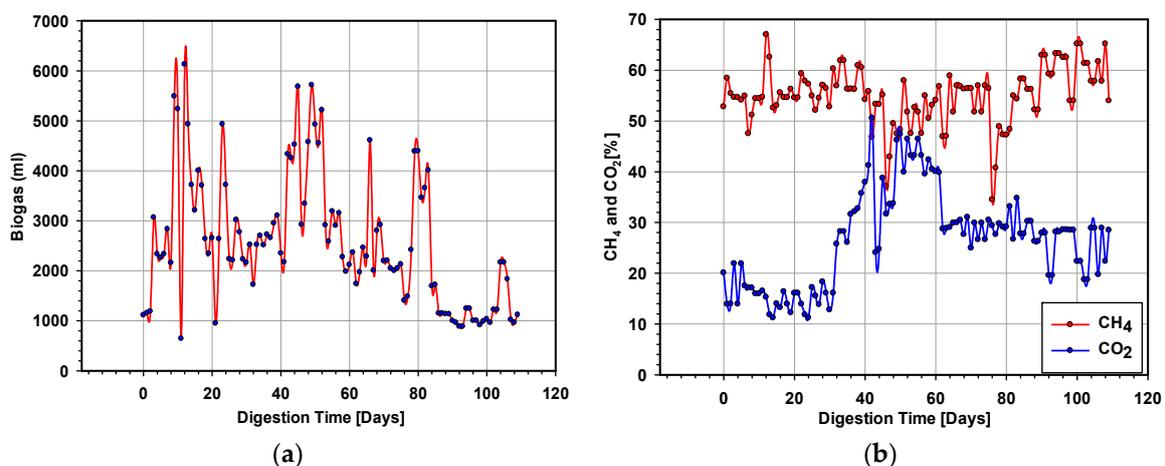


Figure 2. (a) Daily biogas yields with respect to the digestion time. (b) The contents (%) of CH₄ and CO₂.

2.1. Overview of the Model

The model deployed for this research is turbocharged, has six cylinders, and a direct injection engine. The model is depicted in Figure 3 and engine parameters are given in Table 2. The model consisted of an intercooler unit, turbocharger unit, a biogas cylinder, throttle valve that controls the mass flow rate of the air–biogas mixture and a high-pressure fuel injection system (injector with eight holes and 0.25 mm diameter). The engine control unit (ECU) regulated the injection system, the main engine variables (engine speed, crank angle, and mass flow of fuel and air). Performance and exhaust emission were recorded at a fixed engine speed of 1800 rpm and four loads of BMEP 0.425, 0.85, 1.275, and 1.7 MPa.

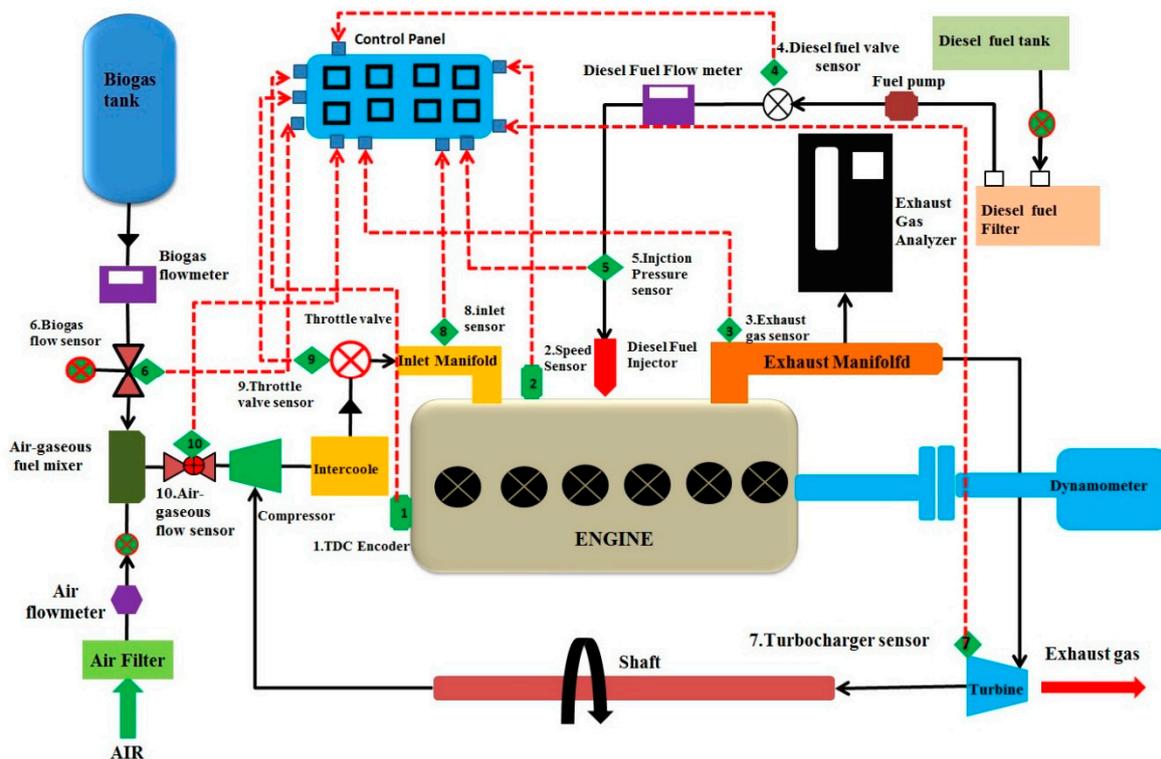


Figure 3. Engine setup.

Table 2. Engine specifications.

Engine Parameters	Values
Rated brake power	298 [kW]
Torque	1580 [N-m]
Bore × Stroke × CRL	119 × 175 × 300 [mm]
CR	13
Total displacement	11.7[L]
Turbocharger	1 unit
Cylinder arrangement	6 in-line
FIN	8 holes
Pilot Fuel injection timing	15 °CA
Injection pressure	2000bar

The present work was carried out with 1D simulation software GT-Power 7.4 version which is commonly used for the modeling and assessment of engines and is based on thermodynamic analyses. Geometrical input information (discharge coefficient and valve lift count) for the valves, pipes, and cylinders were used as input data for modeling the engine. The engine working parameters, such as

surrounding conditions, injection timing, and engine speed, were also used as input information. The compressor and the turbine chart defined the turbocharger's output. The boost pressures were kept up at 1.1, 1.75, 1.9, and 1.9bar corresponding to 0.425, 0.85, 1.275, and 1.7 MPa of BMEP.

2.2. Fuels

Diesel, biogas generated from pig manure and corn straw, and enriched biogas were used as fuels. The biogas was generated from pig manure and corn straw as explained above. After the data was analyzed, three different compositions of biogas with methane percentages of 45%, 50%, and 60% were selected for this study. The properties of the biogas with different compositions were determined and used in the engine numerical model. The biogas was then enriched so that the percentage of methane increased to 75% and 85% (by vol.). The fuels used are designated as follows, BG45 refers to 45% methane gas and 55% CO₂ by volume, BG50 refers to 50% methane gas and 50% CO₂ by volume, BG60 refers to 60% methane gas and 40% CO₂ by volume, BG75 refers to 75% methane gas and 25% CO₂ by volume, BG85 refers to 85% methane gas and 15% CO₂ by volume, and D100 refers to neat diesel fuel. The properties of diesel and biogas are summarized in Table 3. The mass of the injected pilot diesel fuel is depicted in Figure 4.

Table 3. Fuel properties.

Fuel Properties	Diesel	BG45	BG50	BG60	BG75	BG85
Lower heating value MJ/kg	42.5	11.46 *	13.33 *	17.65 *	26.09 *	33.66 *
Density [kg/m ³]	840–880	1.329 *	1.270 *	1.151 *	0.973 *	0.855 *
Flame speed cm/s	-	-	-	25 [31]	-	-
Stoichiometric A/F (mass)	14.5	7.76 *	8.62*	10.32 *	12.93 *	14.6 *
Octane number	-	-	-	130 [21]	-	-
Auto ignition temperature	210	-	-	650 [18]	-	-
Cetane number	45–55	-	-	-	-	-

* Calculated values.

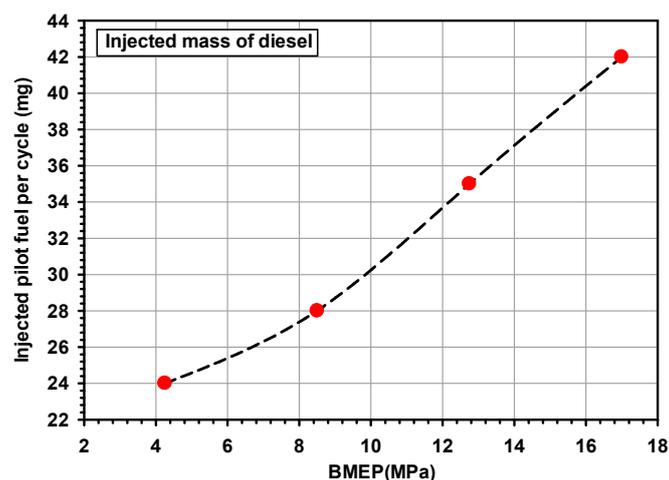


Figure 4. Injected mass of diesel fuel.

Simulations were conducted in order to analyze the impacts of the different compositions of biogas on engine performance and emission. The engine was run at a constant speed of 1800 rpm at various brake mean effective pressures (BMEP) of 0.425, 0.85, 1.275, and 1.7MPa, respectively. Initially, simulations of diesel (single fuel) were performed at four different loads to generate reference data, then further simulations for biogas fuels (15% up to 55% CO₂ by vol.) were performed using diesel as pilot fuel in dual-fuel mode.

3. Numerical Analysis

The numerical analysis was performed using a 1D cycle simulation. A two-zone model was used to analyze the combustion process. Energy equations in each zone were solved separately for each time step and are given by Equations (1) and (2).

$$\frac{d(m_u e_u)}{dt} = -p \frac{dV_u}{dt} - \dot{Q}_u - \left(\frac{dm_f}{dt} h_f + \frac{dm_a}{dt} h_a \right) + \frac{dm_{f,i}}{dt} h_{f,i} \quad (1)$$

For unburned zone (1)

$$\frac{d(m_b e_b)}{dt} = -p \frac{dV_b}{dt} - \dot{Q}_b - \left(\frac{dm_f}{dt} h_f + \frac{dm_a}{dt} h_a \right) \quad (2)$$

For burned zone (2)

Where the four terms on the right hand-side of Equation (1) show heat transfer, combustion pressure work, and addition of enthalpy from the injected fuel, respectively [3].

The numerical model for the heat release rate at each crank angle is given by Equation (3) [3].

$$\frac{dQ_n}{d\theta} = \frac{dQ_{comb}}{d\theta} - \frac{dQ_{ht}}{d\theta} = \left(\frac{\gamma}{\gamma-1} \right) p \frac{dV}{d\theta} + \left(\frac{1}{\gamma-1} \right) V \frac{dP}{d\theta} \quad (3)$$

where dQ_{ht}/dt is given by Equation (4) as follows:

$$\frac{dQ_{ht}}{dt} = h_g A (T_g - T_w) \quad (4)$$

For calculating the in-cylinder gas to the wall heat transfer coefficient, the Woschni Heat Transfer Model is used and is indicated by Equation (5) [3]:

$$h_g = 130 * [B^{-0.2} * p^{0.8} * u(t)^{0.8} * T_g^{-0.53}] \quad (5)$$

The formation of nitric oxides was determined by the extended mechanism of Zeldovich and is given by Equations (6)–(8) [3,48]:



where k^+ and k^- are forward and reverse reactions.

4. Discussion

4.1. Effect of Biogas-Diesel Fuels on Engine Performance and Exhaust Emissions

Engine performance and emission parameters such as brake thermal efficiency (BTE) change in BSFC, Exhaust gas Temperature (EGT), HC, CO, CO₂, and NO_x at four various engine loads and at a constant engine speed of 1800 rpm were assessed for the biogas-diesel fuels and were compared to D100.

4.2. BTE

Figure 5 shows the BTE as a function of engine load for D100 and the biogas-diesel. It was found that BG45, BG50, and BG60 gave better BTE in comparison to that of diesel fuel operation. The BTE values of BG45, BG50, and BG60 were 4.2%, 4.1%, and 3.2%, respectively, higher than that of diesel fuel operation at engine load of 1.7 MPa. However, BG75 and BG85 had 6.5% and 16.3% lower BTE values than D100, respectively. BTE values of all fuels were within the range between 17.4% and 38.2%. This phenomenon can be elucidated by the presence of CO₂ in the biogas. CO₂ functions as an idle gas and in this manner lessens the flame propagation speed. Contrarily, the CO₂ can be separated into O₂ and CO as the burning diesel temperature is sufficiently high to trigger separation [30]. CO is a rapid burning gas so it can be viewed that the combustion rate of the blend may be hastened by the availability CO [33]. Moreover, the availability of an extra amount of oxygen from the separation of CO₂ raises the O₂ level in the blend and enhances the combustion. This may clarify the higher BTE for BG45 compared with the other biogas-diesel fuels. The CO₂ content of biogas does not impact BTE significantly [26]. The BTE increased for all fuels due to increased cylinder temperatures at relatively higher loads. [29]. The increase in cylinder pressure and heat release rate at higher loads improved the BTE.

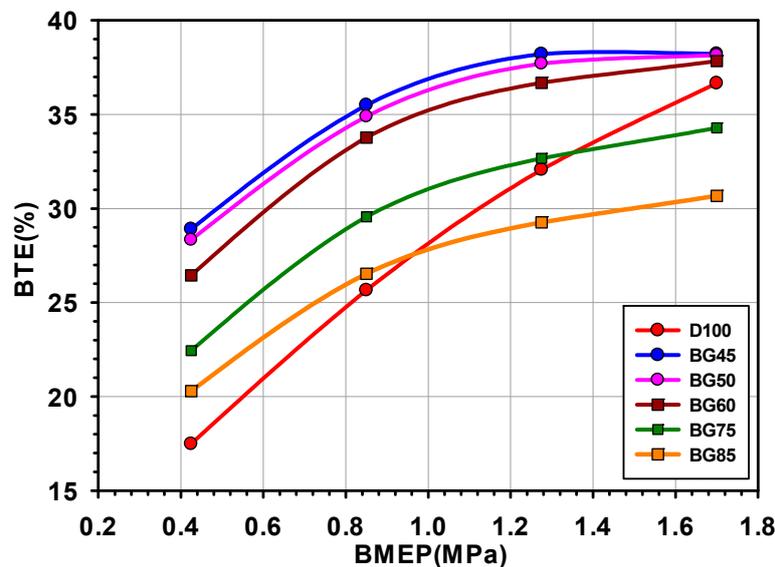


Figure 5. Variations of brake thermal efficiency (BTE) for D100 and biogas-diesel fuel.

The results of this study are in agreement with the finding of Feroskhan and Ismail [26]. They found that at higher loads, the amount of excess air is smaller and therefore the BTE enhances due to the creation of more combustible methane-air mixtures. In another study Sorathia and Yadav [49] also reported almost no decline in BTE for biogas-diesel fuel operated CI dual fuel engine. Nathan et al. [31] reported improved BTE with 50% raw biogas energy share and preheated inducted charge. They concluded that the existence of CO₂ in the biogas delayed combustion leading to a higher BTE for biogas-diesel fuel operation. Feroskhan et al. [50] obtained 5% increase in BTE by preheating the biogas-air mixture in the intake. Similar results can be found in [38,51]. However dissimilar findings were obtained by Bora et al. [37]. They claimed that the decrease in BTE for biogas-diesel fuels was due to lower calorific value, low combustion temperature, and low flame propagation speed of biogas.

4.3. BSFC

Figure 6 shows BSFC as a function of engine load for diesel and biogas-diesel fuels. As indicated in the Figure 6, the BSFC for the biogas-diesel fuels was slightly lower than that of diesel fuel operation at 0.425 MPa engine load. It was found that BG60, had the lowest BSFC among the fuels at lower

engine conditions. Generally, as the ratio of carbon dioxide in biogas increased, the BSFC also increased. At higher loads the BSFC was found to be higher for dual fuel operation, under all engine running conditions. At 1.7 MPa of engine load, the BSFC for diesel fuel operation was found to be 228.4 g/kW-h. The BSFC for BG45, BG50, BG60, BG75, and BG85 was found to be higher by about 59.9%, 48.2%, 29.8%, 19.7%, and 20.7%, respectively, than that of D100. The increase in the BSFC for the biogas-diesel fuel operation is credited to the low calorific value of biogas which causes higher BSFC, hence, more fuel is required to deliver comparable power as diesel. According to Bari [30] as the level of CO₂ increases in the blend, CO₂ remains unseparated. The unseparated CO₂ behaves as a non-reactive gas. The addition of such non-reactive gas influences the burning speed of the gas-air blend; this brings about inadequate combustion that might cause the increase in BSFC. The BSFC values varied from 228.18 to 478.7 g/kW-h. At full load conditions, the best performance of dual fuel engines was recorded (1.7 MPa), where the BSFC was lower by 25.5–50.9% as compared to low load conditions (0.425 MPa). This was due to the higher combustion rate of gaseous fuel as a result of increased combustion temperatures at higher loads.

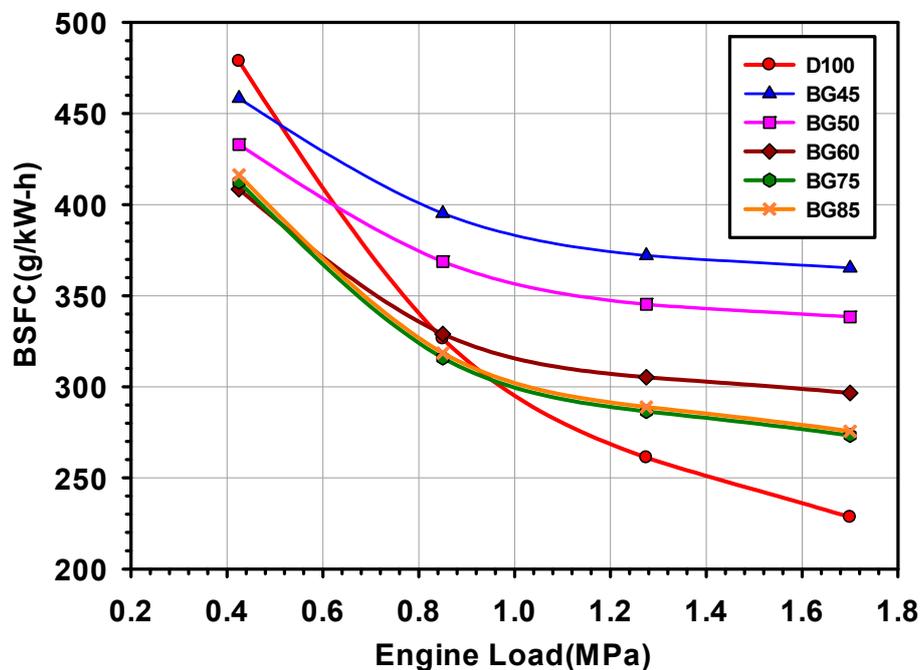


Figure 6. Variations of BSFCs with engine load for D100 and biogas-diesel fuels.

Mustafa et al. [52] reported a similar BSFC trend in biogas-diesel fuels. They obtained that the BSFC of the 49% CO₂ biogas was higher at all loads compared to the 13% CO₂ due to its lower heating value, flame speed, and burning ratio. Moreover, similar findings for biogas-diesel fuel operations were reported by other researchers [29,46,53]. Most of the studies [19,32,54] confirm that the use of biogas increases BSFCs due to their lower energy density and the presence of CO₂.

4.4. Exhaust Gas Temperature (EGT)

Figure 7 illustrates the EGT for diesel and biogas-diesel fuel operations. The exhaust gas temperature with biogas dual fuel operations was found to be slightly higher than that of diesel fuel operation at lower loads. This is due to lower CH₄ and higher CO₂ ratio in biogas which led to lengthening of ignition delay hence causes higher EGT at lower loads [55]. The increase in EGT was in the range of 3.4–18.8% for biogas-diesel fuels as compared to diesel fuel operation at 0.425 MPa. However, at higher loads the EGT of the biogas-diesel fuels was found to be lower. As seen in Figure 7, compared to D100, the biogas-diesel fuels had lower EGT, with the largest decrease being 20.3% for

BG45 at 1.7 MPa of engine load. This is mainly due to the lower concentration of oxygen in biogas-diesel fuel operations which reduced oxygen-rich areas in the combustion chamber causing lower in-cylinder temperatures, hence, decreased EGT. It was found that the lowest EGT was obtained for BG45 i.e., 666.82 °C and the highest EGT was obtained for D100 i.e., 934.4 °C. The EGT for D100, BG75, and BG85 increased by an average of 5.2–26.1% as the engine load was augmented from 0.425 to 1.7 MPa. This is due to increased amount of fuel injected into the combustion chamber at higher loads; hence, the in-cylinder temperature increased. On the contrary, EGT decreased by 7.2%, 8.1%, and 4.6% for BG45, BG50, and BG60, respectively, as the engine load was augmented from 0.425 to 1.7 MPa. This is due to inadequate oxygen as a result of higher CO₂ ratio in the biogas or lower combustion temperature. Moreover the presence higher ratio of CO₂ in biogas absorbs some of the heat from the combustion reaction hence the EGT for the biogas with higher CO₂ percentages decreases [56]. Similar decreases in exhaust gas temperatures for biogas-diesel fuels were reported by Barik and Muragan et al. [29], Duc et al. [51], Barik et al. [46], and Rosha et al. [54].

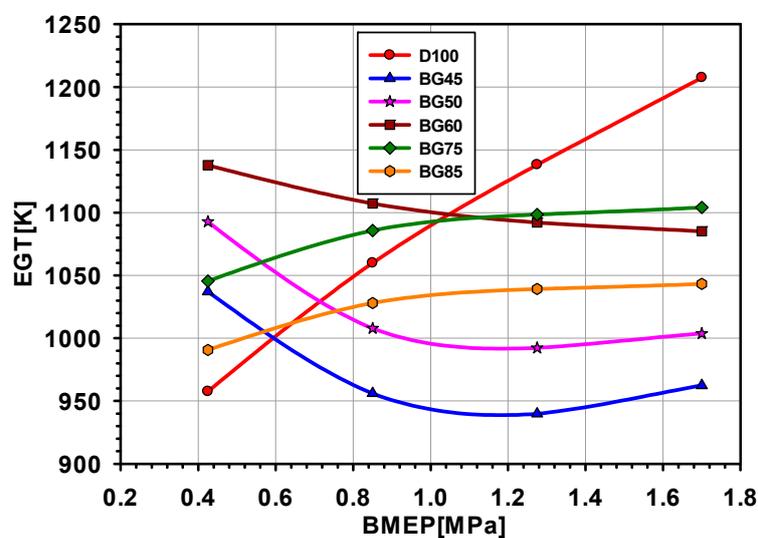


Figure 7. Variations of Exhaust gas temperature (EGT) with load for D100 and biogas-diesel fuels.

4.5. Cylinder Pressure

Figure 8a,b shows the cylinder pressure as a function of crank angle for diesel and biogas-diesel. As presented in Figure 8a peak cylinder pressures of biogas-diesel dual fuel operation were found to be lower than that of diesel fuel operations at lower loads of 0.425 MPa. This is because at lower engine loads, lower admission of fuel energy results in a lower rate of heat release rate (HRR) and consequently diminished cycle pressures and temperatures. In addition, the presence of a notable amount of CO₂ in biogas absorbed the heat release and hence diminished the engine pressure and temperatures. At low load condition (0.425MPa), the maximum peak cylinder pressure was found to be 44.9 bar (at 11.85 °CA aTDC), for D100. The peak cylinder pressure for BG45, BG50, BG60, BG75, and BG85 were found to be lower by about 14.8%, 17.2%, 17.7%, 9.6%, and 3.9%, respectively, than that of D100 at 0.425 MPa of engine load.

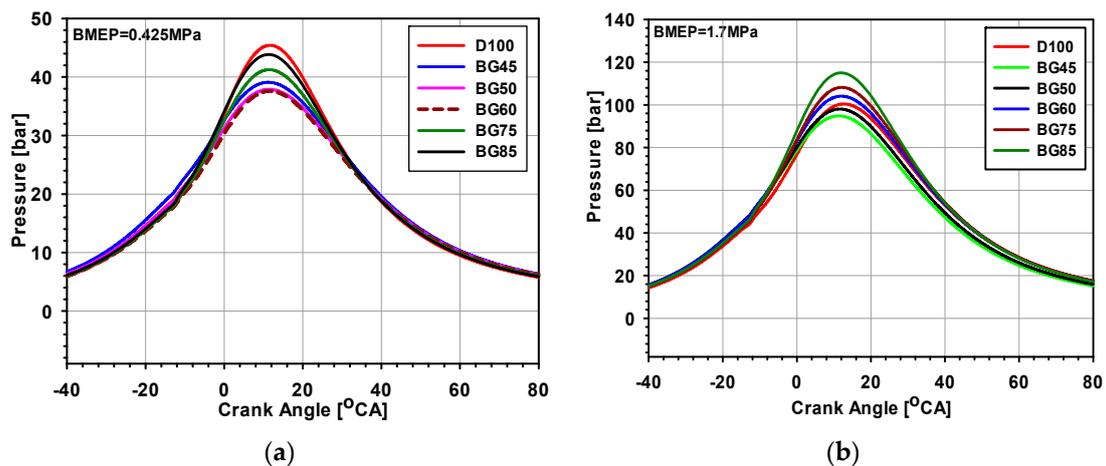


Figure 8. Variations of in-cylinder pressure with crank angle for diesel and biogas-diesel fuels. (a) brake mean effective pressure (BMEP) = 0.45MPa, (b) BMEP = 1.7MPa.

However, as indicated in Figure 8b at higher engine load condition, it was found that the in-cylinder pressure for diesel was 98.5 bar which was higher by 6.8% and 3.7% than that of BG45 and BG50, respectively; while it was lower by 2.3%, 6.6%, and 13.2% than those of BG60, BG75, and BG85, respectively. This rise in the peak cylinder pressure with the biogas-diesel fuels is due to the admission of biogas with the intake-air charge which results in the contraction and dilution of oxygen level, thereby causing longer ignition delay, hence the cylinder pressure increases [36]. According to Liu and Karim [57] the flow of gaseous fuels in diesel engines fundamentally influenced the physical and chemical forms during ignition delay and thus prompts its prolongation. Similar effects were observed by Verma et al. [47], who found higher peak cylinder pressures with biogas dual fuel engines. The cylinder pressure rose with an increasing engine load. The results showed that the increase in the cylinder pressure was about 140–172% for the biogas fuels as the load was augmented from 0.425 to 1.7 MPa.

Figure 9 depicts position of peak pressure for diesel and biogas-diesel. Table 4 shows the injection duration. As shown in Figure 9 the position of peak pressure of biogas-diesel dual fuel operation occurred slightly earlier than that of diesel fuel operation. This is attributed to the CO_2 present in biogas. The flame temperature of diesel can sufficiently induce separation of CO_2 into carbon monoxide (CO) and oxygen [29]. CO is a fast burning gas so that the combustion rate of the biogas-diesel operation is increased due to its presence [33]. Moreover, the availability of an extra amount of oxygen from the separation of CO_2 increases the oxygen level in the blend which makes it burn faster than diesel fuel. For the same running conditions, dual fuel combustion is marked by a rapid and higher release rate of energy and a relatively shorter combustion period as compared to that of diesel fuel combustion. This implies that dual fuel combustion finishes earlier than that of diesel fuel.

Table 4. Combustion parameters of diesel and biogas-diesel dual fuel operation.

Fuel	Injection Timing (°CA bTDC)	Injection Duration (°CA)
BG45	15	13
BG50	15	13
BG60	15	13
BG75	15	14
BG85	15	14
D100	15	14.5

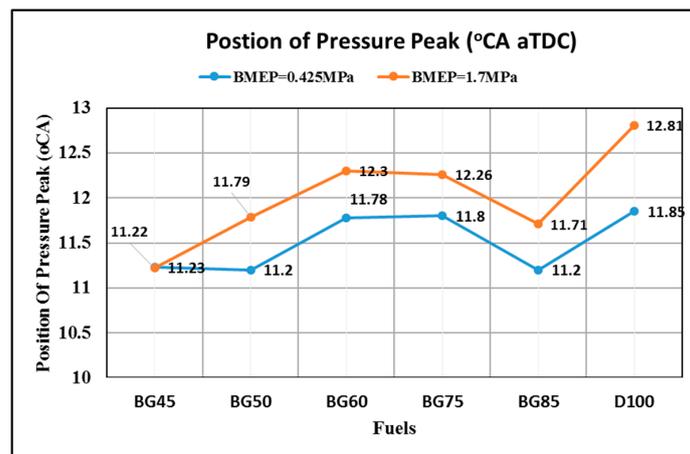


Figure 9. Position of pressure peak.

4.6. Heat Release Rate (HRR)

Figure 10a,b shows the variation of HRR as a function of crank angle for diesel and biogas-diesel fuels. As illustrated in this figure, the HRR increased with the increase in the ratio of CH₄ in the biogas with the increase being higher the higher the ratio of CH₄. As shown in Table 5 and Figure 10a the HRR of diesel fuel operation was found to be higher than those of biogas-diesel fuels at lower loads of 0.425MPa. This is attributed to the decrease in lower heating values (LHVs) of biogas with decrease in CH₄ ratios. At low load conditions (0.425 MPa), the maximum HRR was obtained at 116.8 kJ/°CA (at 6.13 °CA aTDC), 85.3 kJ/°CA (at 10.65 °CA), 84.6 kJ/°CA (at 10.63 °CA aTDC), 86.2 kJ/°CA (at 10.00 °CA aTDC), 97.4 kJ/°CA (at 7.96 °CA aTDC) and 106.2 kJ/°CA (at 6.99 °CA aTDC), for D100, BG45, BG50, BG60, BG75, and BG85, respectively. At lower loads, lower admission of fuel energy led to lower rate of heat release. Moreover, the occurrences of the maximum HRR were found to be delayed in the dual fuel operation, as compared to that of diesel fuel due to higher specific heat of biogas and the presence of CO₂ which caused retarded combustion [19]. At higher loads (1.7 MPa) the HRR of D100 was found to be 243.6 kJ/°CA. The HRR for BG75 and BG85 were 7.1% and 15.7%, respectively, higher than that of diesel fuel operation while, HRR for BG45, BG50, and BG60 were 15.5%, 9.6%, and 0.03%, respectively, lower than diesel fuel operation. The lower heating value of methane is higher than diesel fuel, and both BG75 and BG85 have higher methane concentration which was reflected in their higher HRR. The maximum heat release for D100 occurred at about 10.61 °CA aTDC, whereas, in dual fuel operation, the maximum heat release for BG45, BG50, BG60, BG75, and BG85, occurred at 10 °CA aTDC, 10 °CA aTDC, 10.65 °CA aTDC, 9.01 °CA aTDC, and 7.91 °CA aTDC, respectively, at 1.7 MPa engine load.

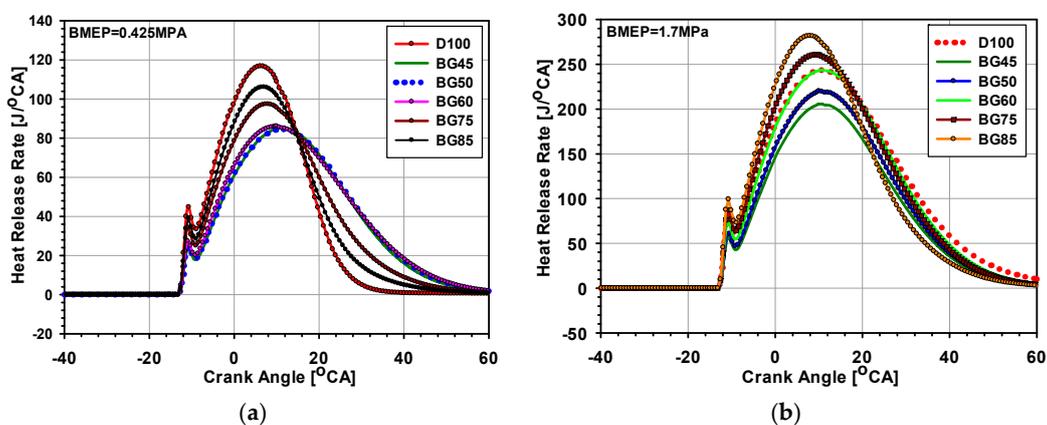


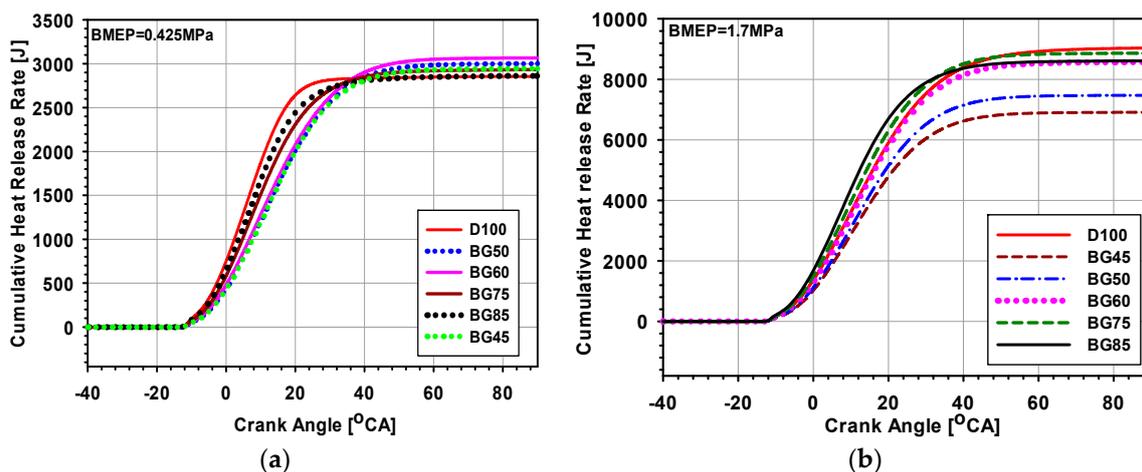
Figure 10. Variations of HRR with crank angle for diesel and biogas-diesel fuels. (a) BMEP = 0.45MPa, (b) BMEP = 1.7MPa.

Table 5. Combustion parameters of diesel and biogas-diesel dual fuel operation.

Load (MPa)	Fuel	Maximum Cylinder Pressure (bar)	PRR (bar/°CA)	Maximum HRR J/°CA	Position of Maximum HRR/°CA
0.425	BG45	38.22	1.07	85.26	10.65
	BG50	37.14	1.07	84.60	10.63
	BG60	36.96	1.14	86.18	10.00
	BG75	40.56	1.34	97.44	7.96
	BG85	43.11	1.49	106.16	6.99
	D100	44.88	1.70	116.82	6.13
1.7	BG45	91.79	2.52	205.78	10.00
	BG50	94.87	2.69	220.17	10.00
	BG60	100.84	3.04	243.51	10.65
	BG75	104.96	3.42	260.81	9.01
	BG85	111.54	3.79	281.80	7.91
	D100	98.49	3.19	243.60	10.61

4.7. Cumulative Heat Release (CHR)

The CHR with respect to engine loads for diesel and biogas-diesel is depicted in the Figure 11a,b. It was observed that at lower engine loads, diesel fuel operation had lower CHR compared to those of biogas-diesel fuels. At 0.425 MPa of engine load, CHR for diesel fuel operation was 2.86 kJ as compared to 2.94, 3.00, 3.07, 2.93, and 2.86 kJ for BG45, BG50, BG60, BG75, and BG85, respectively. Most of the cylinder is filled with biogas at lower engine loads and ignited with a small amount of diesel. As a result, the biogas-diesel fuels showed higher CHR compared to diesel fuel at lower loads. On the other hand, at higher loads, increased input fuel energy led to higher rate of heat release hence the CHR increased. At full engine load (1.7 MPa), CHR for diesel, was found to be 9.04 kJ which was 1.98–30.9% higher than that of biogas-diesel fuel operation. The main reason for lower CHR with dual fuel is that biogas' lower heating value is lower than diesel fuel. The CHR increased with increase in the engine load as a result of increased amount of liquid and gaseous fuel in the combustion chamber.

**Figure 11.** Variation of cumulative heat release with crank angle. (a) 0.425MPa; (b) 1.7MPa.

4.8. Carbon Dioxide

Figure 12 depicts CO₂ emissions as a function of engine load. As illustrated in Figure 12, compared to BG45, the percentage change in CO₂ emissions of the biogas-diesel fuels with (15–50% CO₂ vol.) decreased as level of CO₂ in biogas decreased. The maximum decrease in the change of CO₂ emitted was observed at 41.4% for BG85 at engine load of 0.425 MPa as compared to BG45. This is because

BG45 consists of higher percentage of CO₂ (i.e., 55% by vol.) compared to the other biogas-diesel fuels. At 0.425 MPa load, the fuels BG45, BG50, and BG60 had 68.1%, 65.2%, and 51.5%, respectively, higher CO₂ emissions than those of diesel fuel operation due to their higher CO₂ percentage. BG85 had the lowest CO₂ emissions among the fuels under all running conditions. However, at higher loads (1.7 MPa) it was observed that the biogas-diesel dual fuel mode had slightly lower CO₂ emissions as compared to that of diesel fuel. The increase in CO₂ emissions with regard to diesel fuel operations can be attributed to the increase in combustion temperature that resulted in the oxidation of greater amounts of CO into CO₂. Similar findings were reported by Sahoo [58] and Bora et al. [37] who found higher CO₂ emission under dual fuel operation as compared to diesel fuel operation. In another study Kalasi et al. [1] reported that the CO₂ emission increased by 32%.

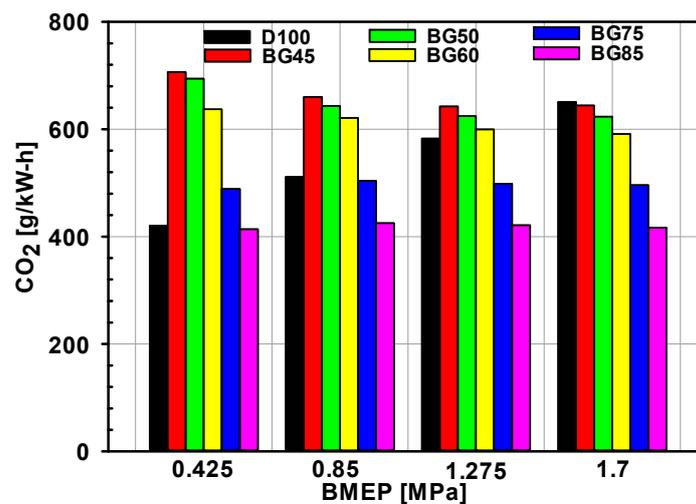


Figure 12. Variation of CO₂ emissions with engine load.

4.9. Total Hydrocarbon (THC)

HC emission is depicted in Figure 13. As illustrated in the Figure 13 it was observed that HC emissions of biogas-diesel (BG-D) dual fuel operations were considerably higher in comparison to that of diesel fuel operations under all the operating conditions. The biogas-diesel fuels had an average of more than 300% higher HC than that of diesel fuel operation. HC emission was observed to be the most minimal for diesel at higher load conditions. The HC emission increased with increasing engine load for the biogas-diesel fuels, while it decreased for the diesel fuel. The injection of biogas caused a rich mixture in the combustion chamber and decreased the concentration of oxygen in air-fuel charge. Besides, biogas has lower burning velocity [30]. These brought about the deficient burning in dual fuel operation hence HC increased. The CO₂ in the biogas-diesel blend absorbed the heat and decreased the in-chamber temperatures which hinders the hydrocarbon oxidation mechanism [29]. Moreover, other sources such as valve-overlapping, inaccurate blending of fluid and vaporous fuel, longer ignition delay of biogas-diesel fuels, impact from crevice volume, and wall quenching also contribute to higher HC [37]. Many scientists have revealed comparative outcomes in HC emission with biogas dual fuel operations [29,38,55]. Bora et al. [45] indicated that HC emissions under dual fuel mode were higher than diesel fuel mode because of lower flame velocity which resulted in higher HC. They also found that HC emissions decreased for higher compression ratios. Kalsi et al. [1] also reported that HC outflows rose with the addition of biogas because of lower in-chamber temperature with the rise of biogas energy share. Similar increases in HC trends in dual fuel mode with biogas can be found in [19,33,51].

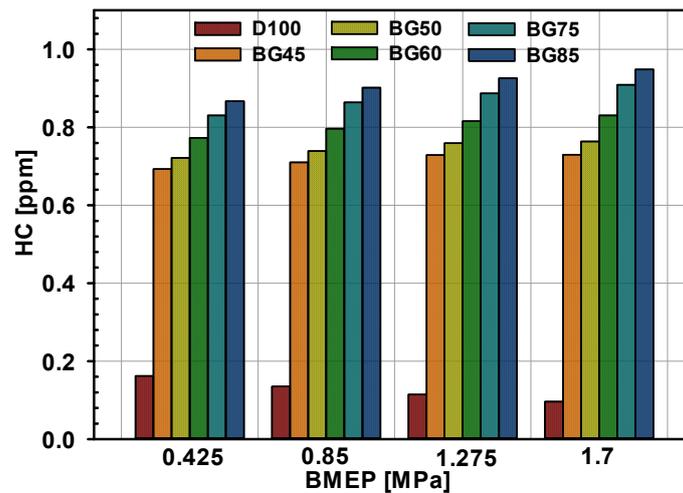


Figure 13. Variations of HC emissions with engine load.

4.10. Nitrogen Oxides

Figure 14 shows NO_x as a function of engine load. The NO_x discharges in dual fuel operation were essentially lower. As seen in Figure 14, the addition of biogas fuels decreased NO_x emissions significantly for all engine loads, the decrease being higher the higher the CO₂ percentage in biogas. As compared to diesel fuel operation, NO_x emissions for BG45, BG50, BG60, BG75, and BG85 decreased an average of 82.5%, 77.7%, 72.7%, 67.7%, and 61.4%, respectively, at 0.425 MPa of engine load. Similar reduction in NO_x emissions was also observed at higher loads. The decrease in NO_x emission is on the grounds that the usage of biogas with air in dual-fuel operation results in the decline of O₂ concentration in the air-fuel mixture which ultimately cut down the NO_x generation rate. Additionally, a high-specific heat CO₂ in biogas leads to a reduction in cylinder temperature by absorbing some amount of heat. Combined impacts of these bring down NO_x discharges in dual fuel operation with biogas. Diesel has a high calorific value compared to biogas [41]. In diesel fuel operation the high in-cylinder temperature caused increased NO_x emission. It was observed that NO_x emissions increased with the decreasing ratio of CO₂ in the biogas-diesel fuel for instance, the change in NO_x emissions was compared to BG45 and showed that the NO_x increase was 27.2%, 55.3%, 84.3%, and 119.6% for BG50, BG60, BG75, and BG85, respectively. This is due to the amount of oxygen and as the ratio of methane in biogas increases it may increase oxygen-rich regions inside the cylinder resulting in greater NO_x formation. The NO_x emission of BG45 was found to be the lowest among the fuels considered. NO_x emissions increased with an increasing engine load. As the load increased, the combustion chamber temperature ascends as more amount of fuel needs to be burnt, consequently, NO_x pollutants increased for all the fuels used with respect to higher engine loads as indicated in Figure 14. This is due to high quantity of fuel and improved combustion at higher loads which led to high peak pressures and temperatures hence NO_x increased [47,55].

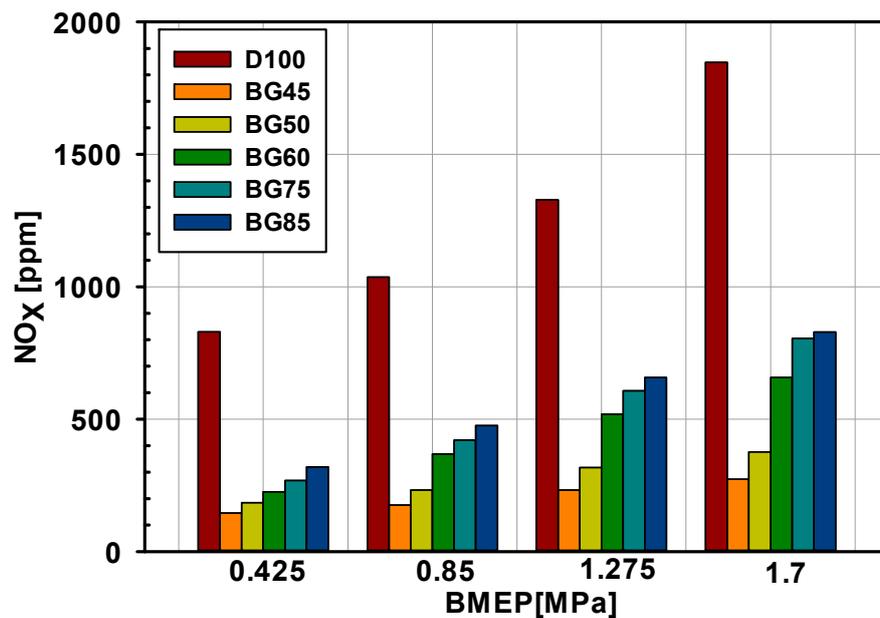


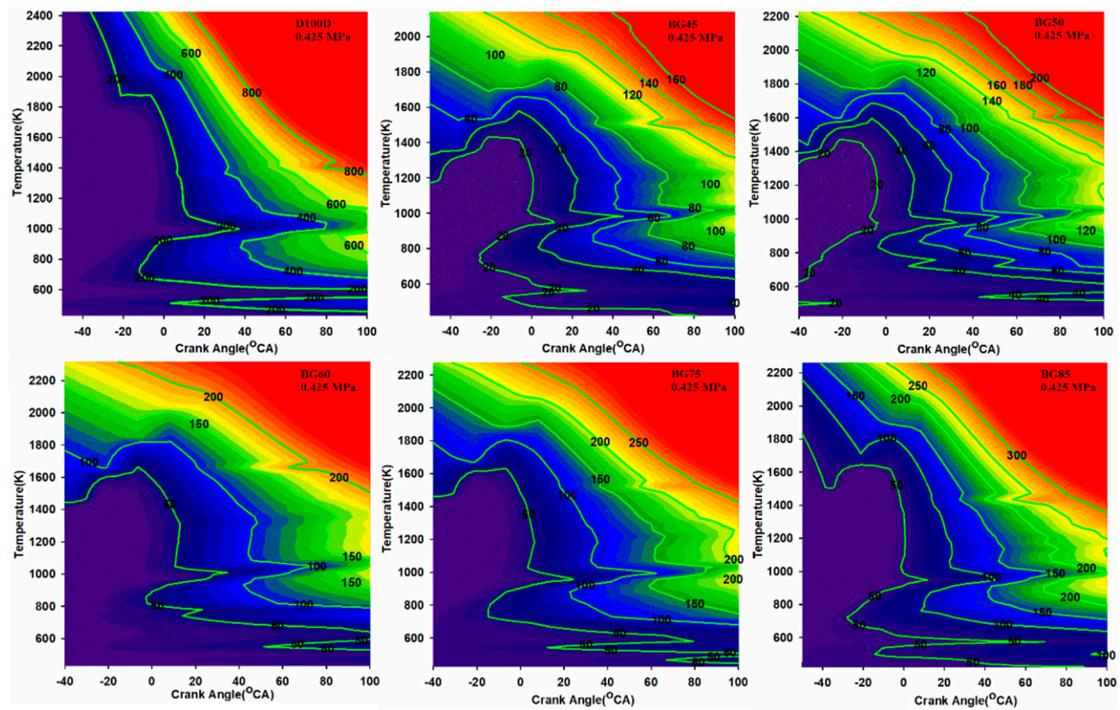
Figure 14. Variation of NO_x emissions with engine load.

The results of this study are in agreement with the finding of Barik and Murugan [29] and Lounici et al. [33]. Verma et al. [55] reported a similar NO_x emission trend where NO_x decreased by an average of 28.7% for biogas-diesel fuel operation as compared to diesel fuel operation. Mustafi et al. [19] also reported that NO_x emissions decreased by about 9–12% for biogas-diesel fueling compared to diesel fueling due to increased CO₂ ratio in biogas. Similar decrease in NO_x emission with biogas-diesel dual fuel operation can be found in [19,32,37].

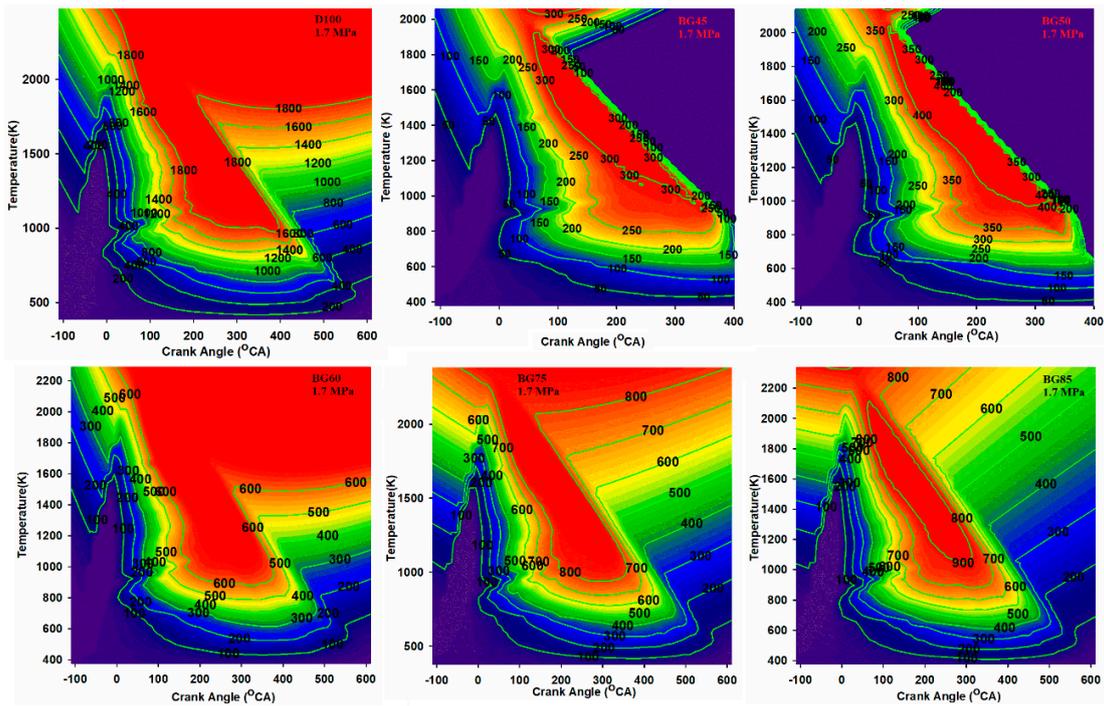
Figure 15a,b shows the NO_x distribution for D100 and biogas-diesel at 0.425 and 1.7 MPa of engine loads. As seen in Figure 15a low NO_x emissions were observed at lower loads, with minimum emissions (i.e., 145.5 ppm) recorded for BG45. NO_x emissions were found to be higher when the engine load increased due to higher combustion temperatures as seen in Figure 15b. The maximum NO_x emission was found to be 1847.1 ppm for D100 and 828.9 ppm for BG85, at 1.7 MPa load.

4.11. Carbon Monoxide (CO)

CO as a function of engine load for diesel and biogas-diesel dual fuel is depicted in Figure 16. It was observed that the CO pollutant of biogas-diesel fuels were lower than that of diesel fuel operation for all the engine loads except for BG75 and BG85 which had slightly higher CO emissions at 1.7 MPa of BMEP. This could be because air-fuel ratios with dual fuel operations are somewhat higher than diesel fuel. At 0.85 MPa engine load, air-fuel ratio for BG45, BG50, BG60, BG75, and BG85 were found to be 12.10, 12.26, 11.8, 11.7, and 11.72, respectively, whereas that of diesel was found to be 11.34. CO emissions decreased gradually when the engine load was augmented due to increased combustion temperature which resulted in a higher conversion of CO to CO₂, hence, CO emissions decreased. Higher CO emissions were observed at lower loads, however at higher loads CO emissions decreased significantly, with minimum discharges recorded at 1.275 MPa engine load for the biogas-diesel fuels. The D100 recorded the highest value of CO at 0.425 MPa engine load. This trend is also reported by Mustafi et al. [21]. The increase of CO at lower loads is due to the inadequacy of oxygen, less time for combustion, and lower combustion chamber temperature which prompted inadequate combustion, hence more CO pollutants while at higher loads CO emissions were reduced due to more complete combustion. Himsar [7] reported that CO pollutants are not impacted by the level of methane in the biogas. They also reported that the CO number increases with increasing biogas flow rate. Similar results can be found in [19,39].



(a)



(b)

Figure 15. NO_x distribution. (a) 0.425 MPa (b) 1.7 MPa.

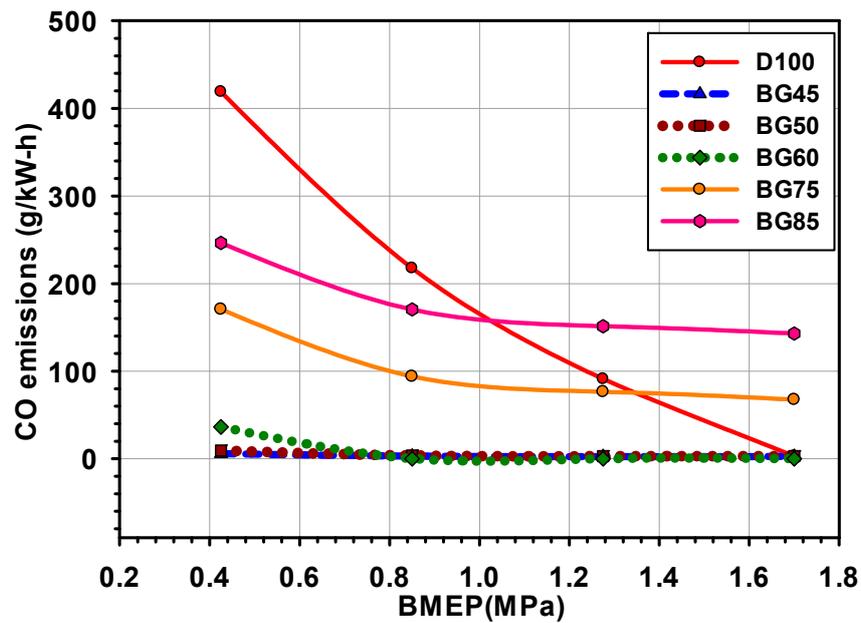


Figure 16. Variations of CO emissions with load.

4.12. Model Validation

We used the test work of Lounaci et al. [33] to verify our numerical simulations. We built Lounaci's dual-fuel model that operates on biogas-diesel (diesel, BG50, BG60, BG70, and BG80) in GT power and performed numerical simulations. Comparisons between the numerical analysis and experimental investigations for brake thermal efficiency, in-cylinder pressure, NO_x , and particulate matter are shown in Figure 17a–d. The results of the simulation were in line with the experimental results, confirming the model's accuracy. The details of the engine used for validation is given below in Table 6.

Table 6. Details of engine used for validation.

Engine Parameters	Values
Engine type	4 stroke
Bore	95.5 mm
Stroke	88.94 mm
Displaced volume	630 cc
Compression ratio	18
Power output	4.5 KW at 1500 rpm
Start of injection	13°bTDC

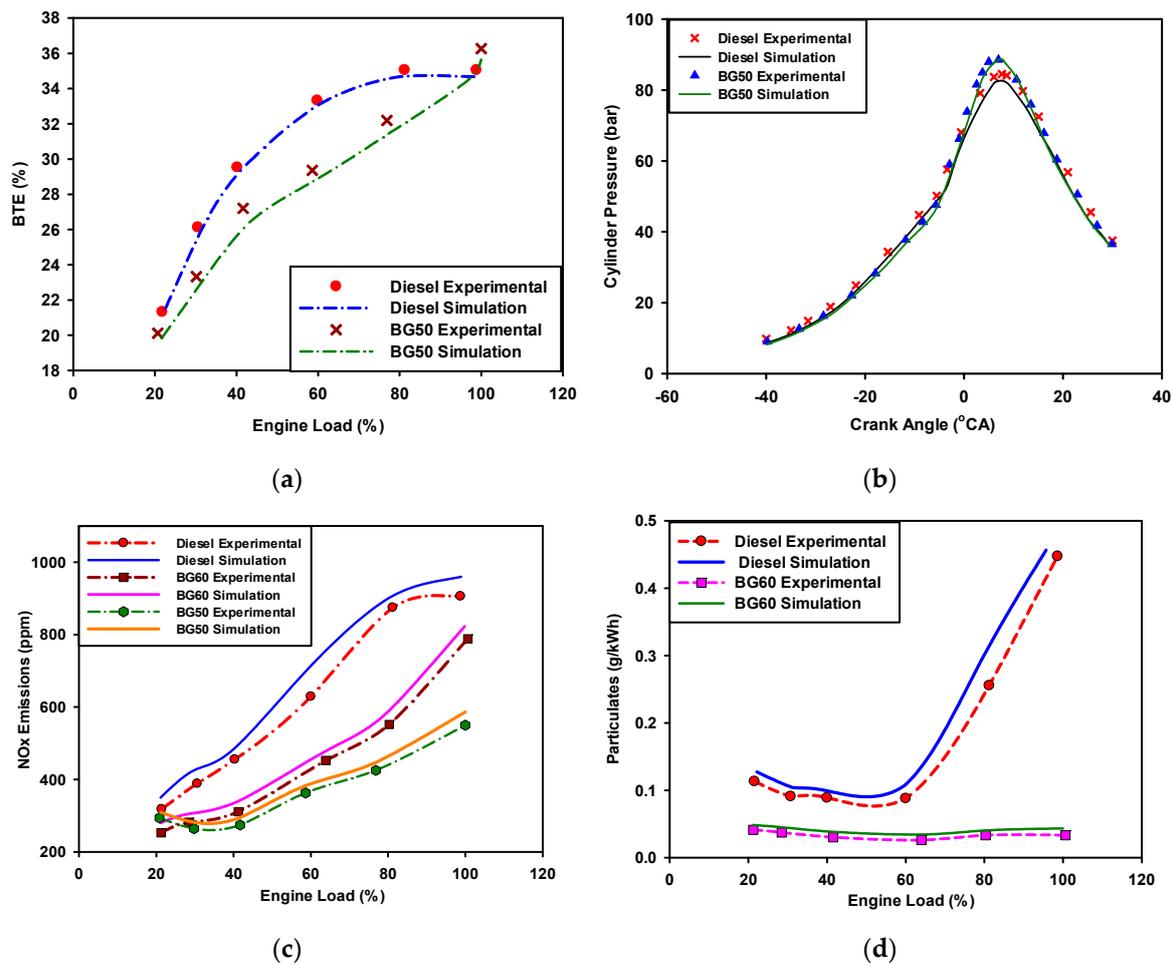


Figure 17. (a) Comparison of BTE between numerical and experimental for BG50 and diesel (b) Comparison of in-cylinder pressure between numerical and experimental for diesel (D100) and BG50. (c) Comparison of NOx emissions between numerical and experimental for diesel (D100), BG50, and BG60. (d) Comparison of particulate matter between numerical and experimental for diesel (D100) and BG60.

As shown in Figure 17d the PM emission of biogas-diesel dual fuel operations was found to be lower in comparison to that of diesel fuel operations under all the operating conditions. Diesel fuel operation had an average of 76% higher PM emission than that of biogas-diesel fuel operation. The PM emission increased with increasing engine load for diesel, while it decreased for the biogas-diesel fuel. Similar decreases in PM emission for biogas-diesel fuels were reported by H.S. Tira [59] and Owczuk et al. [34].

5. Conclusions

In the present work, a numerical investigation was carried out to examine the effects biogas generated from PM and CS as well as enriched biogas on the performance, combustion, and emission of a diesel engine. The following conclusions can drawn:

1. The use of biogas-diesel fuels enhanced the BTE. The BTE values for the biogas with CO₂ ratio of 40–55% of by vol. were on average 3.84% higher than that of diesel fuel operation while those with CO₂ ratio of 15–25% by vol. the BTE were found to be lower by an average of 11.4%. The CO₂ ratio of biogas does not impact BTE considerably.

2. Biogas-diesel provided superior performance in reductions of NO_x emission. NO_x emission for the biogas-diesel was lower than that of diesel fuel operation due to decreased in-cylinder temperature and increase in dilution effect.
3. HC emissions for biogas-diesel fuels increased sharply compared to neat diesel fuel operation under all load conditions. Similarly, the CO₂ emissions of biogas-diesel fuel were higher than those of diesel fuel operations under all load conditions except at 1.7MPa, where the CO₂ emissions of diesel were found to be higher.

Author Contributions: S.A.A. conceived and developed the paper; S.Z. provided guidance and reviews; A.S.T. conducted the experiments; Y.Z. and Y.F. were responsible for project administration; N.A. carried out validation and also helped in writing the paper; A.M. was responsible for methodology. All authors have read and agreed to the published version of the manuscript.

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Nomenclature

Q_{comb}	Total heat released
dQ_n	Apparent heat release rate
Q_{ht}	Heat lost to the cylinder walls
p	In-cylinder pressure
T_g	The gas instantaneous temperature
T_w	Cylinder wall temperature
A	Cylinder heat transfer area
h_g	Coefficient of heat transfer
V	Cylinder volume
γ	Specific heat ratio
BTE	Brake thermal efficiency
CRL	Connecting rod length
CR	Compression ratio
$^{\circ}\text{CA}$	Crank angle in degrees
aTDC	After top dead center
BMEP	Brake mean effective pressure
FIN	Fuel injection nozzle
CO	Carbon monoxide
CO ₂	Carbon dioxide
NO _x	Nitrogen oxides
HC	Hydrocarbon
LHV	Lower heating value
BSFC	Brake specific fuel consumption
BG-D	biogas-diesel
ppm	Parts per million
EGT	Exhaust gas temperature
PRR	Pressure rise rate

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