www.rericjournal.ait.ac.th

Effects of CO_2 ratio in Biogas on Performance, Combustion and Emissions of a Dual-Fuel Diesel Engine

Salman Abdu Ahmed*[1], Song Zhou*, Asfaw Solomon Tsegay[+], Yuanqing Zhu*, Adil Malik*, Naseem Ahmad*, and Zenebe Legese*

Abstract – *In this study, biogas was generated by anaerobic co-digestion process of pig manure and corn straw. The biogas was used as an alternative gaseous fuel in a DI (direct injection) turbocharged diesel engine, in the dual fuel mode with diesel as pilot fuel. Three compositions of biogas generated from pig manure and corn straw: 45BG55CO_2, 50BG50CO_2 and 60BGCO_2 (containing 45%, 50% and 60% of methane (CH_4) by volume respectively) and two compositions of enriched biogas: 75BG25CO2 and 85BG15CO_2 (containing 75% and 85% of CH_4 by volume) were used. The effects of carbon dioxide (CO_2) ratio in biogas on combustion, performance and emission characteristics of the engine in the dual fuel operation were numerically analyzed at four various engine loads (0.425, 0.85, 1.275 and 1.7MPa), and compared with that of diesel fuel operation. The numerical simulations were carried out using GT-Power commercial package. The results showed that the BTE values for biogas-diesel fuel operations were found to be higher compared to that of diesel fuel operation. The BTE was not considerably impacted by CO_2 ratio. The highest BTE value of 38.22% was recorded for 45BG55CO_2 (45% methane and 55% CO_2). The exhaust gas temperatures (EGT) of the biogas-diesel fuels were found to be 8.5-20.3% lower than that of the diesel fuel operation at higher load condition. With respect to emissions, The NO_X (Nitrogen oxides) and carbon monoxide (CO) emissions in the dual fuel operation were found to be lower by about 55.3-83.3% and 73.1-97.4%, compared to that of diesel operation. However, the unburned hydrocarbons (HC) and carbon dioxide (CO_2) discharges of dual-fuel operation were found to be higher by average of 349.7% and 39.8% respectively. 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% CO_2 content biogas.*

Keywords – *anaerobic digestion, biogas, biogas-diesel, diesel engines, emission and performance.*

1. INTRODUCTION

Due to increasing fuel prices, environmental concerns and expanding interest for transportation fuels, the quest for alternative sustainable source of energy with accentuation on alternative fuel sources has been rising. Besides renewable energy plays an important role in cutting down the energy import from other countries and helps in diversifying the means of power generation and guarantees a clean environment. These are the fundamental purposes for searching alternative sources, which are amply accessible and friendly to the environment. Liquefied petroleum gas (LPG), hydrogen, biogas, and producer gas are promising alternative fuels. The conventional internal combustion engines are used in numerous fields such as transportation, power generation and heating. Hydrocarbon fuels are known for their high thermal efficiency however, they produce some noxious emissions such as hydrocarbons (HC), carbon monoxide (CO), carbon dioxide (CO_2), nitrogen oxides (NOx), sulfur oxides (SOx), and particulates. Since gaseous fuels result in low levels of emissions they can viably be used in both spark ignition (SI) and compression ignition (CI) engines.

One of the greatest issues that the world is confronting today is that of environmental pollution, which is causing grave and irreparable harm to the natural world and human society. As clearly indicated in Figure 1, China tops the world in almost all types of air pollution, including sulphur dioxide and nitrogen oxides, as well as carbon emissions. In the last few years it had invested heavily on energy conservation and environmental protection and is getting serious about pollution more than ever; with new green strategies so hard-hitting and broad they can be felt over the world. China as an agricultural country has an enormous potency of biogas. China ranks first in the world in animal manure production and crop residue production. The total annual production of animal manure from large scale centralized farms in China is about 837 million tons per year where 208 million tons (accounting for 24.9%) are pig manure (PM). The crop residue production is estimated over 800 million tons per year of crop residues of which 324.8 million tons per year (accounting 40.6%) are corn straw (CS) [1]-[3].The production of biogas and its utilization in different purposes is very vital for the reduction of pollution and reduce the dependency on fossil fuels.

*College of Power and Energy, Harbin Engineering University, P.O. Box 150001, Nangang district, Harbin, China.

[+]State Key Laboratory of Urban Water Resource and Environment, Harbin Institute of Technology (SKLUWRE, HIT), Harbin 150090, China.

[1]Corresponding author:
Tel: + 8618345156205.
Email: salman2014@hrbeu.edu.cn.

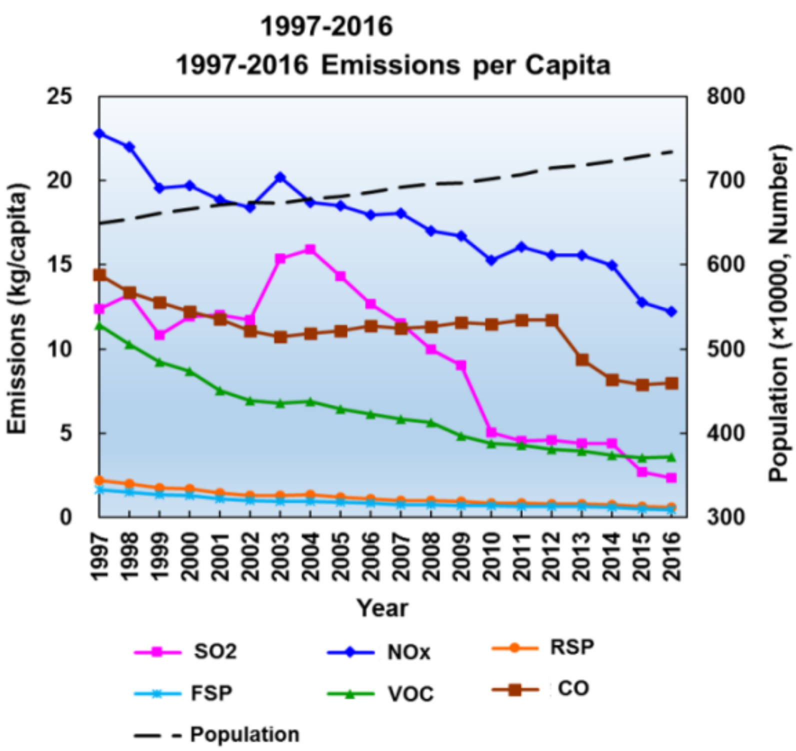


Fig. 1. Change in emissions relative to population [4].

Gaseous fuels are appealing as a result of their high hydrogen to carbon ratio and they have very low emissions when they are used in IC engines. Hydrogen, biogas and producer gas can be acquired from renewable sources [3]. Besides, their great blending attributes with air makes them preferential for internal combustion engines. Biogas is one of those gaseous fuels which is an attractive source of energy. Biogas is a carbon neutral gaseous fuel, resulting in no new addition of greenhouse gases to the environment. It is product of anaerobic digestion that primarily consists of methane and carbon dioxide. The quality of biogas in terms of composition varies depending on biomass, precursors, additives and the conversion process. Methane is the primary constituent of biogas, and the extent shifts from feedstock to feedstock. Table 1 gives the biogas yield and methane level of some normally utilized feed stocks [5]-[7]. In general, biogas contains 50-75% methane (CH_4), 25-45% carbon dioxide and rest including traces of hydrogen sulfide (H_2S), nitrogen (N_2) and hydrogen (H_2) and typically has a calorific value of 21-24 MJ/kg [8]. The typical biogas yield from anaerobic digestion (AD) of pig manure is 0.34 m^3/kg (dry matter), which is approximately equal to 0.28 m^3/kg VS (content of methane in biogas 65%, and 75% VS/TS ratio) [9].

Two techniques can be utilized to work a CI engine with biogas. The first technique is changing the CI engine into spark engine and fuel it with biogas only and the second one is biogas-diesel dual fuel engine [10], [11]. Biogas cannot be utilized to run a CI engine due to its high self-ignition temperature but, it can be used in a CI dual-fuel approach. Its low cetane number being a gaseous fuel makes it suitable for CI engine in dual fuel mode. The main advantage of a dual fuel engine is that, it can run with a wide assortment of fluid and vaporous fuel with no significant engine alterations [10], [12]. The high anti-knock properties of biogas compared with conventional fuel makes it an applicable fuel for dual-fuel engines. However, the presence of CO_2 reduces thermal efficiency due to prolonged ignition delay and decreased flame temperature [13]. Moreover, the CO_2 present has some negative influence towards some parameters such volumetric energy density, fuel conversion efficiency, and the combustion enthalpy [5]. The presence of CO_2 could impact the combustion process (burning velocity), which might result in incomplete combustion prompting HC discharges and reduced engine efficiency [14]. Carbon dioxide acts as inert gas and impacts the burning velocity of the in-cylinder charge, thereby resulting in incomplete combustion; that maybe cause the increase of BSFC [15]. Nathan *et al.* [16] reported that the CO_2 in the biogas suppressed the high rate of release which is common in HCCI engines. However; they found that improved efficiency and low levels of NO_X and soot. In another study Makareviciene *et al.* [17] evaluated the impact of the CO_2 concentration in biogas on the performance and exhaust emissions of a diesel engine. Their findings revealed that lower pollutant levels were observed when the engine was operated with the EGR system. Lounici *et al.* [18] reported that the high proportion of CO_2 present in biogas significantly reduces NO_X and PM.

Table 1. Biogas yield and percentage of methane [5].

Feed straw	Biogas yield, m^3/kg	Methane, %
Cattle dung	0.297	55
Pig manure	0.4	65
Corn straw	0.341	51
Municipal solid waste	0.308	60
Spent wash	0.65	58.7
Leaves	0.21	58
Wheat straw	0.432	59

The use of biogas fuel for the transport application is becoming increasingly important. Biogas can be upgraded to the quality of natural gas and used in the same natural gas vehicles (NGVs) [19]. The biggest biogas manufacturers as fuel in 2016 were Germany, Sweden, Switzerland, UK and the US. The transition to biogas in the transport industry, particularly in China, France, UK and Scandinavia, has been strongly encouraged [20]. Both heavy and light-duty cars can use biogas. Light-duty cars can normally be operated on natural gas or biogas without any adjustment, while heavy duty vehicles operating alternate with biogas or natural gas without the closed loop controls may require adjustment. Sweden is one of the few countries in the world with a national standard for biogas to be used as a vehicle fuel. According to Bhatia [21] there are more than 4000 vehicles in Sweden running on natural gas and biogas. There is a significant increase in public transport cars powered on biogas like buses and waste trucks [19]. Several European towns are exchanging their buses with biogas-powered engines [19]. 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 CO_2 emissions by 75% to 200% compared with fossil fuels [19]. When fluid residues are used as organic fertilizers (subsequent to mineral fertilizers), it can prevent emissions of CH_4 by landfilling or storing manure and further save GHGs [20].

A lot of studies have been carried out on the usage of biogas in a diesel engine in a dual fuel mode using biogas-diesel, biodiesel-biogas and diesel-biogas-hydrogen. 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 2.

Biogas is turning into a potential source of energy in various nations over the globe since it can be utilized to fuel cars in the transport sector and has also been utilized for warming purposes and power generation [37]. Most of the previous works carried out in this area were on naturally aspirated engines either on a single or twin cylinder engines .There are a few studies on the effects of biogas on turbocharged multi cylinder dual fuel engines. Moreover, this engine requires less modification to change it to dual fuel engine. These are the main reasons behind our selection of this engine. According to the author's knowledge, the impact of CO_2 ratio in biogas on the performance and exhaust emissions on a six cylinders turbocharged engine has not been thoroughly studied. In perspective of the above setting, the target of the present work was to investigate the impact of CO_2 ratio in biogas on the performance, combustion and emissions characteristics of a six cylinder CI diesel engine under dual fuel mode with biogas (BG) as primary fuel and diesel as pilot fuel.

1.1 Background

Biofuels are produced by domestically available organic feedstock. Anaerobic digestion (AD) is considered as one of the most dynamic process to transform organic matter into a useful form of energy, *i.e*. biogas. The organic material when confined in a place in the absence of oxygen, give rise to a large number of bacteria, which assimilate and break down organic matter to produce methane gas as a major by-product [38]. The AD can be applied to treat waste such as wastewater, sludge and municipal solid waste (MSW) [39]. It can also be used to treat animal manure [40], energy crops [41], organic food waste [42], microalgae [43] and agricultural residues [44]. The AD can be considered one of the most vital methods to convert complex organics into biogas [45], [46].

The organic matter conversion to biogas follow four main conversion phases namely; hydrolysis, acidogenesis, acetogenesis and methanogenesis [47], as shown in Figure 2. During hydrolysis stage complex polymeric organic matter including carbohydrates, proteins and fats transforms into simple organic monomers by the action of hydrolytic bacteria. The monomers such as sugar, amino acids and fatty acids are then converted into volatile fatty acids (VFAs) under the action of fermentative bacteria during second stage called as acidogenesis. During the third phase acetogenic bacteria transforms VFAs into acetic acid and hydrogen (H_2) gas. Methanogenic bacteria transform acetic acid and H_2 into methane (CH_4) and carbon dioxide (CO_2) [48]. The quality of biogas in terms of composition varies depending on biomass, precursors, additives and the conversion process.

Continent wise global biogas production contribution from Global Bioenergy Statistics 2017 by World Bioenergy Association (WBA) is shown in Figure 3(a). Biogas as a renewable source of energy is an emerging sector globally with consecutive increment in the production capacity over the years. Figure 3(b) represents the regional breakdown, not only reflecting the overall increment but also every region is showing growth over the years, which is a great motivation for scientists and investors for the biogas augmentation utilizing all the available technologies to pursue state-of-the-art solutions for biogas production.

Table 2. Reviews of research works on the usage of biogas.

Researcher	Engine Type/No of Cylinders	Fuels used	Pressure	HRR	BTE	NO_X	HC	CO
E. Porpatham et al. [22]	4.4 kW at 1500 rpm /01	Biogas	↑	↑	↑	↑	↑	↑
Park et al. [13]	117.6kW SI gas Engine /06	N2 + Biogas	X	X	↑	↓	↑	X
Park et al. [13]	117.6Kw SI gas Engine /06	H2+Biogas	X	X	↓	↑	↓	X
Ramesha et al. [23]	10hp/ dual fuel /(01)	Biogas +biodiesel	↑	↑	X	↓	↑	↑
Yoon and Lee [24]	46kW at 4000rpm/ 04	Diesel+biogas and Biodiesel +Biogas	↓	↑	X	↓	↑	↑
Makareviciene et al. [17]	66kW/ dual fuel /04	Biogas +Diesel	X	X	↓	↓	↑	↑
Bora et al. [25]	3.5kW/1500rpm/01	Biogas +Diesel	X	X	↓	↓	X	↓
Mustafi et al. [26]	32.6Nm at 1800 rpm /01	NG+Diesel and Diesel+Biogas	↑	X	↑	↓	X	↑
Ambarita [10]	4.41kW at 2600rpm/01	Diesel +Biogas	X	X	↑	x	↑	↑
Nathan et al. [16]	3.7kW at 1500rpm /01	biogas–diesel HCCI	↓	↓	↓	↓	↑	↑
Ibrahim et al. [27]	18.24kW at 3600rpm/02	biogas +diesel PPCCI	X	X	↑	↓	↓	X
Yilmaz [28]	48kW/04	biogas+Diesel	↑	↑	X	**NE**	↓	X
Bora and Saha [29]	3.5kW at 1500rpm/01	Biogas+biodiesel	X	X	↑	↓	↑	↑
Zhang et al. [30]	10.3kW/01	biogas +hydrogen	↑	X	X	↑	X	X
Karen et al. [31]	20kW at 3000rpm/02	Diesel+ biogas	↑	X	↑	X	X	X
Pattanaik et al. [32]	3.78kW at 1500rpm)/04	Biodiesel+ biogas	X	X	↓	↓	↑	↑
Barik and Murugan [5]	4.4kW at 1500rpm/01	Diesel +Biogas	↑	↑	↓	↓	↑	↑
Bora and Saha [33]	3.5kW at 1500rpm/01	Biodiesel+ biogas	↑	↑	↑	↑	↓	↓
Kalsi and Subramanian [34]	7.4kW at 1500rpm/01	Biodiesel+ biogas	X	↓	↑	↓	↑	↑
Barik et al. [35]	4.4kW at 1500rpm)/01	Biodiesel+ biogas	↓	X	↑	↑	↓	↓
Verma et al. [36]	4.4kW at1500rpm /01	Biogas +diesel +Hydrogen	↑	↑	X	↑	↓	↓

↑=Increased ↓=Decreased NE=not effected , HRR=Heat release rate , BTE=brake thermal efficiency, X:study not conducted

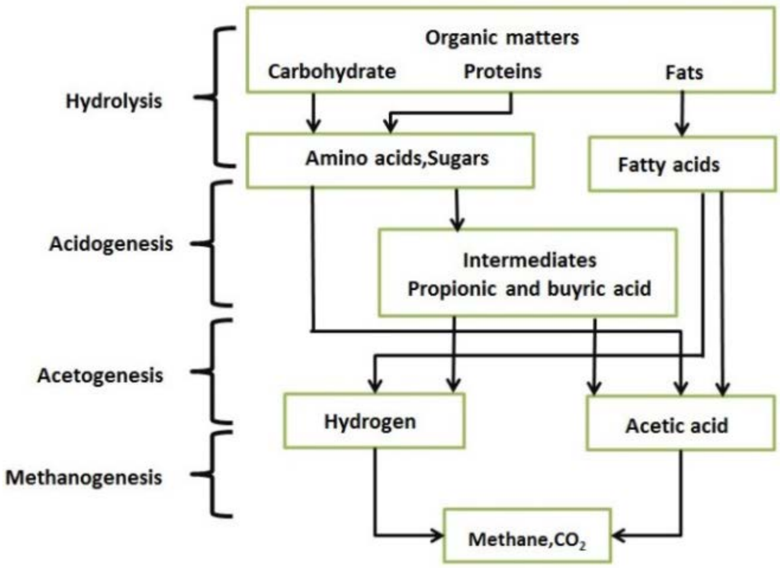


Fig. 2. Common major sequential processes during anaerobic digestion [49].

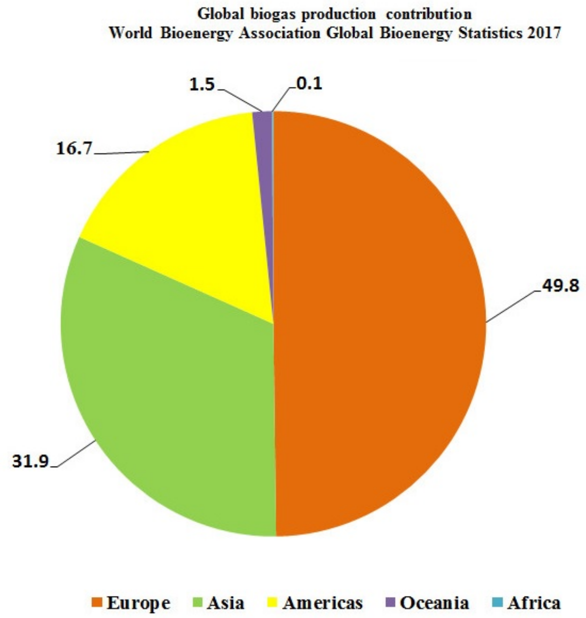


Fig. 3(a). Global biogas production contribution [50].

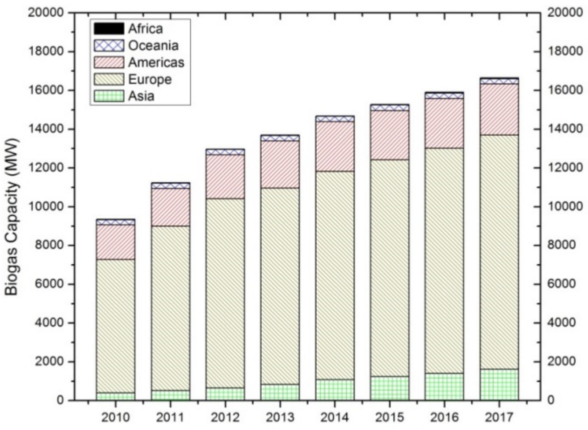


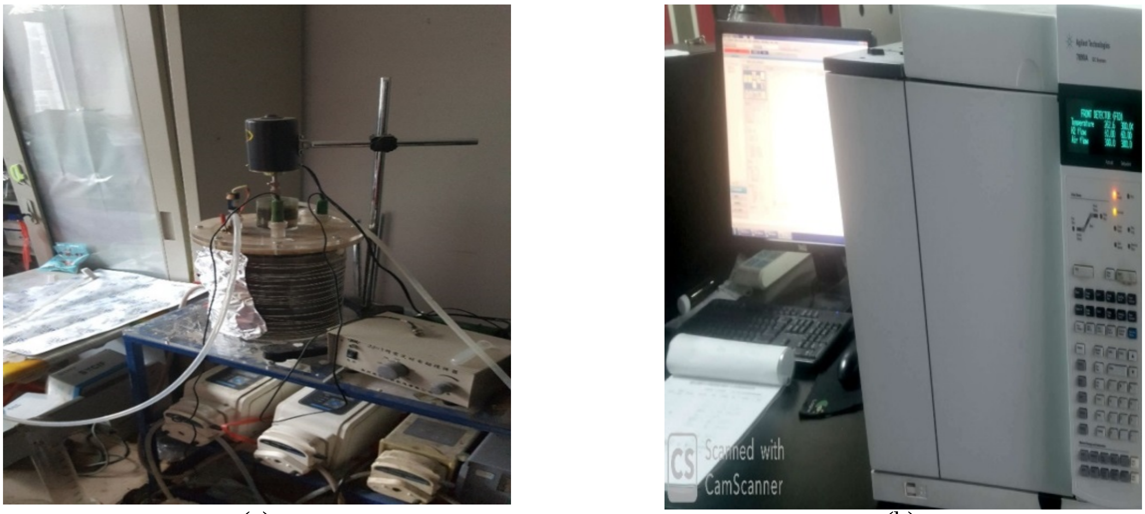
Fig. 3(b). Regional breakdown of biogas [51].

2. MATERIALS AND METHODS

2.1 Experimental Setup for BG Production

The experimental setup consists of two anaerobic continuous stirred tank reactor (CSTR) for anaerobic co-digestion of pig manure and corn straw. The reactors with a total volume of 5.0 L and a working volume 4.5L made of Plexiglas. The reactor temperature was maintained at 35 ± 1 ℃. The hydraulic retention time (HRT) over the whole experimental period was 30 days. To avoid photolysis of antibiotics, the reactor was covered with aluminum foil. The reactor was equipped with a mechanical stirrer to ensure the proper mixing. And these reactors were controlled with reactor cap opening for feeding, pH, ORP electrode, and collecting gas tubes.

The biogas used in this study 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 CSTR reactors (4.5 m^3 working volume) and co-digested in anaerobic environment to produce biogas as shown in Figure 4(a).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, USA) (Figure. 4(b)) fitted with a thermal conductivity detector (TCD) and helium as the carrier gas was used to measure the CH_4 - CO_2 content in the biogas. The CO_2 percentage in biogas was intermittently measured using a Fyrite gas analyzer (Bacharach Inc.) as per the method indicated by the maker. Three different biogas compositions were chosen with CH_4 to CO_2 ratio of 45:55, 50:50 and 60:40. Further the biogas was enriched to increase its CH_4 to 75 and 85% by volume.



(a) (b)

Fig. 4. (a) Anaerobic CSTR reactor; (b) Gas chromatography machine.

The weather condition changes of the region didn't affect the co-digestion system because the reactors were kept in a room with a room temperature by using the hot water conditioning system (winter season) and mechanical air conditioning system (summer season) to maintain the room temperature of 21-24 ℃. So, the room temperature was always similar thoughout the year, what matters to the co-digestion process was the reactor operating temperature applied on it to maintain the mesophilic (35 ± 1℃) temperature condition by a temperature measuring and controlling instrument.

There was no environmental pollution during the conversion of the organic matter to biogas, since anaerobic co-digestion system was carried out with the absence of air (mainly oxygen), and it wasn't exposed to the atmosphere. However there existed unpleasant smell during mixing of the pig manure and the corn straw at the pre-treatment stage (2-3hrs) before inoculated to the digestor.

Figure 5(a) shows the daily biogas yield with respect to the digestion time. The average daily biogas production during the 110 days' digestion time was observed to be 2000 ml/day per kg of PG (pig manure) and CS (corn straw). Figure 5(b) depicts the percentage of methane and CO_2 over 110 days of the digestion period. The authors were able to produce a 0.06m^3 biogas from 1 kg of pig manure biomass.

2.2 Test Fuels

The fuels used in this study are diesel and biogas generated from pig manure and corn straw with CH_4:CO_2 ratio (45:55, 50:50 and 60:40) as well as enriched biogas (75:25 and 85:15). They are designated as follows, 45BG55CO_2 refers to 45% methane gas and 55% CO_2 by volume, 50BG50CO_2 refers to 50% methane gas and 50% CO_2 by volume, 60BG40CO_2 refers to 60% methane gas and 40% CO2 by volume, 75BG25CO_2 refers to 75% methane gas and 25% CO_2 by volume and 85BG15CO_2 refers to 85% methane gas and 15% CO_2 by volume and D100 refers to neat diesel fuel. The properties of diesel and biogas are summarized in Table 3.

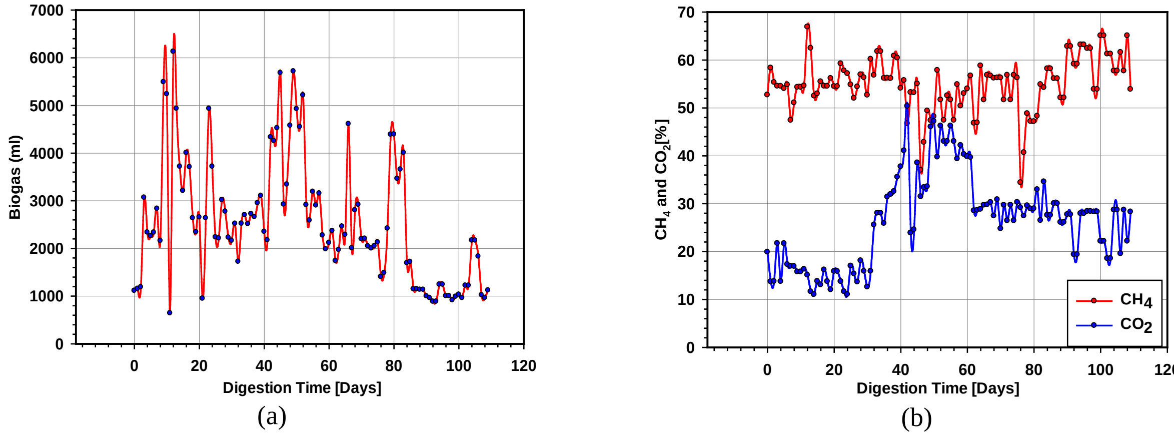


Fig. 5. (a) Daily biogas yields with respect to the digestion time. (b) the contents (%) of methane and carbon dioxide

Table 3. Fuel properties.

Fuel properties	Diesel	45BG55CO2	50BG50CO2	60BG40CO2	75BG25CO2	85BG15CO2
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[16, 36]		
Stoichiometric A/F (mass)	14.5	7.76*	8.62*	10.32*	12.93*	14.6*
Octane number	-			130[52]		
Auto ignition temperature	210			650[33, 37]		
Cetane number	45-55			-		

*Calculated values

2.3 Equipment Set-Up

The engine used for this study is six cylinders, direct injection and turbocharged engine. The application used in the current work is GT-Power which is an extensively-used 1D simulation package for engine modeling and analysis. It is based on one-dimensional gas dynamics representing the flow and heat transfer in the pipes and other components of an engine system. It is designed applicable to all different kinds of internal combustion engines. The engine mentioned above was modeled by using different blocks and interconnections that represent the engine layout. The following input data are required to establish the model: the engine geometric data, the intake and exhaust valve profiles, the compressor and turbine performance maps, the constants of the engine sub-model (combustion, heat transfer, and friction), the engine operating point (load/speed), and the ambient conditions.

The engine model/setup includes high-pressure common rail fuel injection system (injector with 8 hole, 0.25 mm diameter, injection pressure of 2500bar), turbocharger unit, intercooler unit and throttle valve that controls the mass flow rate of the air-biogas mixture. As shown in figure 6 the air flow rate and the biogas mass flow rate were controlled by a sensor connected to their respective control valves depending on the engine load. The injection system main engine parameters (engine speed, crank angle and mass flow rate of fuel and air) were controlled by the engine control unit (ECU). The cylinder pressure was measured by pressure sensor in the ECU. The HC, CO and NO_X emissions from diesel combustion and biogas-diesel fuels were recorded by the emission sensors in the ECU. Specifications are shown in Table 4 and the engine setup are shown in Figure 6.

Table 4. Engine specifications.

Engine parameters	Values
Rated brake power	298[kW]
Bore x stroke x connecting rod length	119x175x300[mm]
Total displacement	11.7[L]
Cylinder configuration	6 in-line
Compression ratio	13
Turbocharger	1 unit
Fuel injection nozzle	8 holes
Pilot fuel injection timing	15^{o}bTDC

The turbine and compressor efficiency maps are shown in Figures 7(a) and 7(b). The waste gate is coupled on the exhaust gas turbocharger system, so as to adjust the inlet gas flow rate of turbine. The waste gate controller was used to meet the desired boost pressure. In this study the boost pressures were 1.9, 1.9, 1.75 and 1.1bar for 0.425, 0.85, 1.275 and1.7MPa of BMEP respectively. The desired boost pressures for 25-100% engine loads are shown in Figures 8(a) and 8(b).

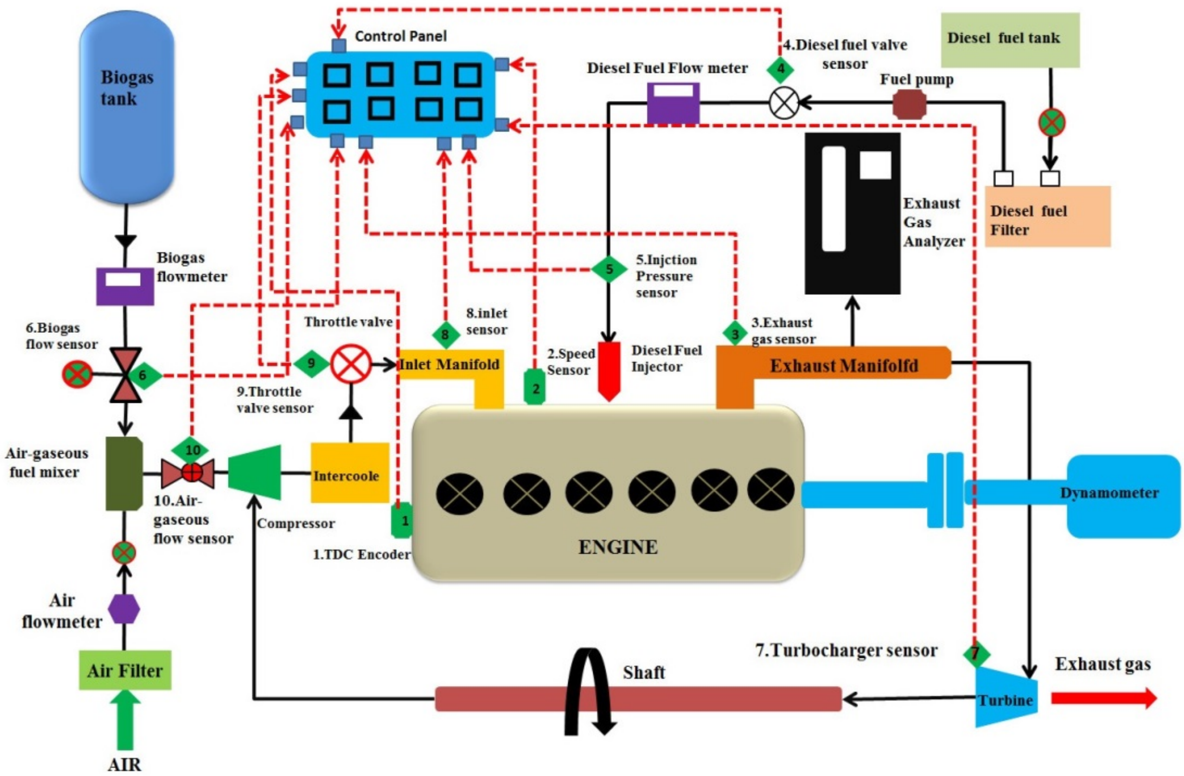


Fig. 6. Engine set up.

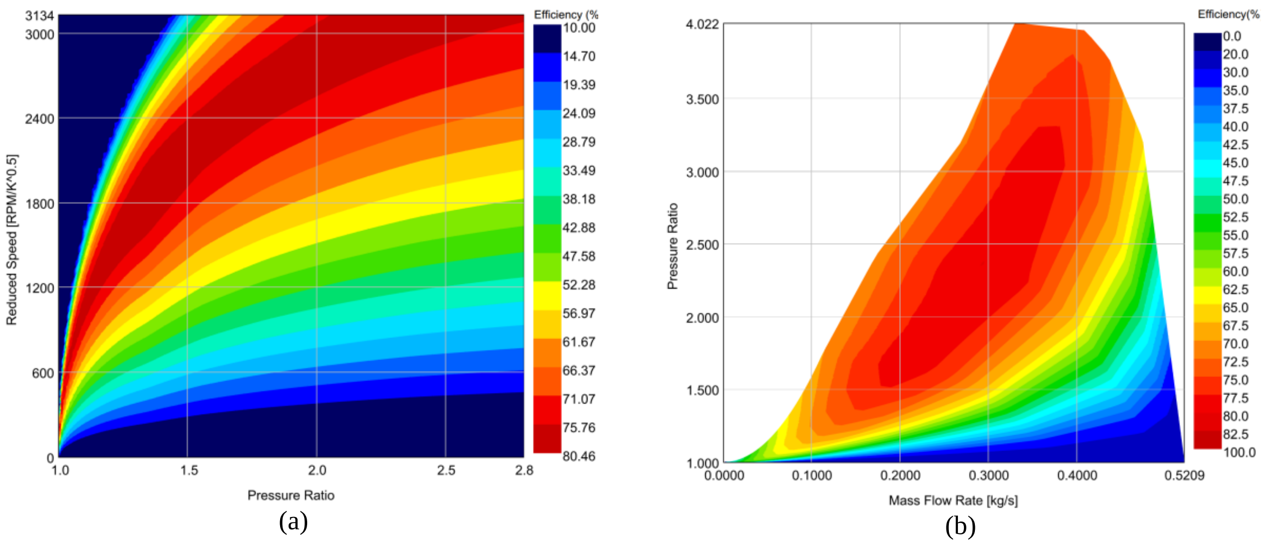
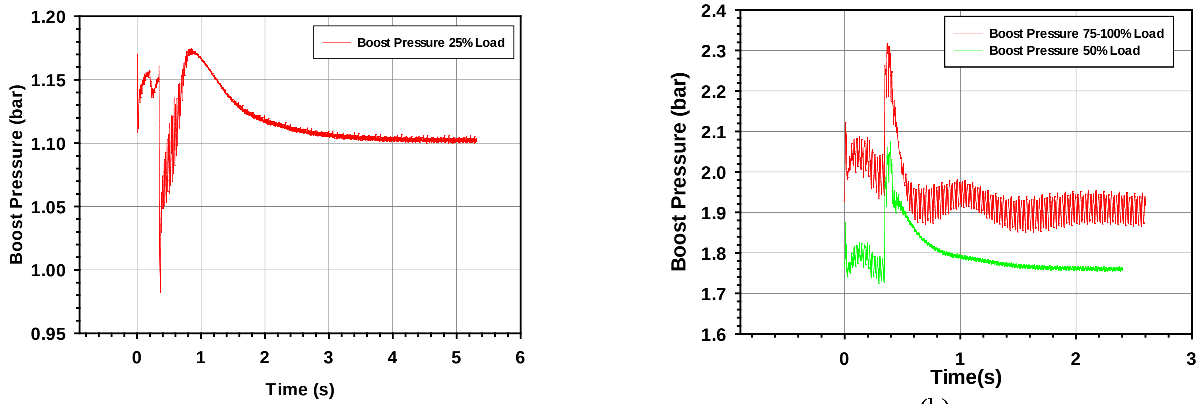


Fig. 7. (a) Turbine efficiency map, (b) Compressor efficiency map.



(a) (b)

Fig. 8. (a) boost pressure from boost-sensor a) 25% load b) 50-100% load.

The GT-power has built in functions to calculate emissions based on the fuels used and exhaust gas composition. The nitric oxide formation was calculated based on the extended Zeldovich [53] mechanism and are given by Equations 1, 2 and 3 [53], [54]:

$$N_2 + O \underset{k_1^-}{\overset{k_1^+}{\rightleftarrows}} NO + N \qquad (1)$$

$$N + O_2 \underset{k_2^-}{\overset{k_2^+}{\rightleftarrows}} NO + O \qquad (2)$$

$$N + OH \underset{k_3^-}{\overset{k_3^+}{\rightleftarrows}} NO + H \qquad (3)$$

Where k^+ and k^- are given as follows [53]:

$$k_1^+ = 7.6x10^{13} \exp(-\frac{38000}{T})$$

$$k_1^- = 1.6x10^{13}$$

$$k_2^+ = 6.4x10^{9} T \left[\exp(-\frac{3150}{T})\right]$$

2.5 *Numerical Simulation*

In order to analyze the impacts of CO_2 on engine performance simulations were carried out with carbon dioxide fractions ranging from 15% up to 55%. The engine was run at a constant speed of 1800 rpm and engine loads of 25%, 50%, 75%, and 100% corresponding to 0.425, 0.85, 1.275 and 1.7MPa brake mean effective pressure (BMEP) respectively. Simulations were performed in single-fuel mode and dual-fuel mode. Simulations were carried out initially using diesel fuel to generate the reference line data, then further simulations were performed for biogas fuels (15% up to 55% CO_2 by vol.) using diesel as pilot fuel.

3. PROBLEM FORMULATION (GOVERNING EQUATIONS)

Different parameters are defined to model the performance, emissions and combustion of the given engine .The rate of heat release at each crank angle is given by Equation 4 [53]:

$$\frac{dQ_n}{d\theta} = \frac{dQcomb}{d\theta} - \frac{dQ_{ht}}{d\theta} = \left(\frac{\gamma}{\gamma - 1}\right) P \frac{dV}{d\theta} + \left(\frac{1}{\gamma - 1}\right) V \frac{dP}{d\theta} \qquad (4)$$

Where dQht/dt is given by Equation 5 as follows:

$$\frac{dQ_{ht}}{dt} = h_g A(T_g - T_w) \qquad (5)$$

The Woschni heat transfer model was used to calculate the in-cylinder gas to wall heat transfer coefficient and is given by Equation 6 [53]:

$$h_g = 130 * \left[B^{-0.2} * P^{0.8} * u(t)^{0.8} * T_g^{-0.53}\right] \qquad (6)$$

The in-cylinder pressure of the model was given by Equation 7 [53]:

$$\frac{dP}{d\theta} = \frac{\gamma - 1}{V}\left[\frac{dQ_{comb}}{d\theta} - \frac{dQ_{ht}}{dt}\right] - \gamma \frac{P}{V}\frac{dV}{d\theta} \qquad (7)$$

The ignition delay period was simulated based on the empirical formula developed by Hardenberg and Hase [53] and is given as follows by Equation 8:

$$\tau_{id}(CA) = (0.36 + 0.22\bar{S}_p)\exp\left[E_A\left(\frac{1}{RT} - \frac{1}{17{,}190}\right)\left(\frac{21.2}{p - 12.4}\right)^{0.63}\right] \qquad (8)$$

Where Sp is the mean piston speed and R is the universal gas constant, E_A is the apparent activation energy ,T and p are charge temperature and pressure during the delay.

Volume of gaseous fuel that will replace inducted air is obtained on volumetric basis by using Equations 9 and 10 [5], [31], [32], [50]:

$$\%CH_4 = \frac{Vol_{CH_4}}{Vol_{CH_4} + Vol_{air}} \qquad (9)$$

$$\%CO_2 = \frac{Vol_{CO_2}}{Vol_{CO_2} + Vol_{air}} \qquad (10)$$

The BTE, BP and BSFC for dual fuel biogas-diesel mode are given by Equation 11, 12 and 13 as follows [33], [35], [55]:

$$BTE_{dual} = \frac{B_P}{\dot{m}_D * LHV_D + \dot{m}_{BG} * LHV_{BG}} * 100 \qquad (11)$$

$$BP = \frac{2\pi * N * T}{60 * 1000} \quad (12)$$

$$BSFC = \left(\frac{\dot{m}_D + \left(\frac{LHV_{BG}}{LHV_D} \right) * \dot{m}_{BG}}{1000} \right) \quad (13)$$

Where B_P is the brake power, N is engine speed, T is engine torque m_D is the diesel mass flow rate, m_{BG} is the biogas mass flow rate, LHV_D is the diesel low heating value and LHV_{BG} is the biogas low heating value.

4. RESULTS AND DISCUSSIONS

4.1 Model Validation

In order to validate the numerical simulations the experimental work conducted by Lounaci *et al.* [18] was used. Lounaci's model which is a single cylinder, four stroke, naturally aspirated direct injection engine which uses biogas-diesel dual fuels (diesel, BG50, BG60, BG70 and BG80) was built in GT power and numerical simulations were carried out. Figure 9(a) and (b) show the comparisons between the numerical analysis and experiments for brake thermal efficiency and in-cylinder pressure. The simulation results were in good agreement with the experimental results, thus confirming the accuracy of our model.

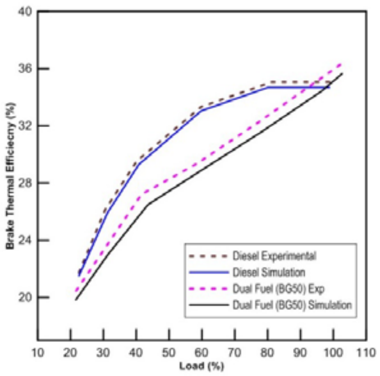
4.1 Engine Performance

4.1.1 BTE

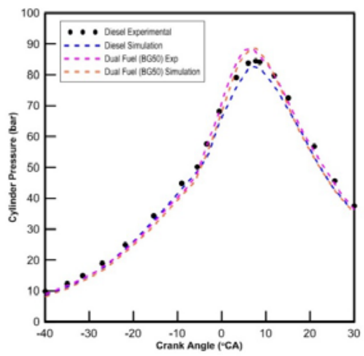
Figure 10 shows the BTE for D100 and the biogas with various ratios of CO_2 as a function of engine load. It was found that the biogas-diesel fuels had better BTE compared to that of diesel fuel operation .The highest increase in BTE obtained was 65.3% at 0.45MPa for $45BG55CO_2$ as compared to diesel fuel operation. at 1.7MPa, the $45BG55CO_2$ fuel achieved the highest BTE *i.e.* 38.22%. The higher BTE with $45BG55CO_2$ may be due to the dissociation of CO_2 into CO and O_2, providing a fast-burning mixture which improves combustion. Moreover as shown in Figure 11(a) and (b) $45BG55CO_2$ had lower heat transfer coefficient which indicated that there were lower heat losses to cylinder walls during combustion, this resulted in a higher BTE when $45BG55CO_2$ was used. The best results in terms of BTE for all the fuels were obtained at 1.7MPa. The BTE of diesel fuel operation was found to be 36.7% whereas, the BTE of 45BG55CO2, $50BG50CO_2$,$60BG40CO_2$, $75BG25CO_2$ and $85BG55CO_2$ was 38.22%, 38.15%, 37.84%, 34.29% and 30.67%, respectively, at BMEP of 1.7MPa . The CO_2 content of biogas didn't influence the BTE significantly. The BTE for all the fuels increased with the increase in load due to increased cylinder temperature at relatively higher loads [50] .The increase in cylinder pressure and heat release rate at higher loads improved the BTE. The results of this study are in agreement with the finding of Feroskhan and Ismail [52].

4.1.2 BSFC

Figure 12 shows the brake specific fuel consumption (BSFC) as a function of engine load for diesel and biogas-diesel fuels. As seen in the Figure 12, the BSFC for the biogas-diesel fuels were found to be lower than that of diesel fuel operation at 0.425MPa engine load. The largest decrease was 14.7% observed at 0.425MPa when $60BG40CO_2$ was used. Compared to that of diesel fuel operation at higher loads the BSFC was found to be higher for dual fuel operation, under all engine running conditions. At 1.7MPa of engine load, the BSFC for diesel fuel operation was found to be 228.4g/kW-h which was lower by about 19.7-59.9% than those of dual fuel operation. The increase in BSFC for the biogas fuels at higher loads compared to that of diesel fuel operation is credited to the low calorific value of biogas which necessitates that a more prominent measure of biogas be expended so as to deliver comparable power as diesel. The best performance of the dual fuel engine was recorded at full load conditions (1.7MPa), where the BSFC was lower by 25.5-50.9% as compared to low load conditions (0.45MPa). This was attributed to the higher rate of combustion of gaseous fuel at higher load due to higher combustion temperatures. The BSFC values varied from 228.18g/kW-h observed on D100 at 1.7MPa engine load, to the maximum value of 478.7g/kW-h, observed for D100 at 0.425MPa of engine load. Similar results were obtained by Mustafa et al. [56].



(a)



(b)

Fig. 9 (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.

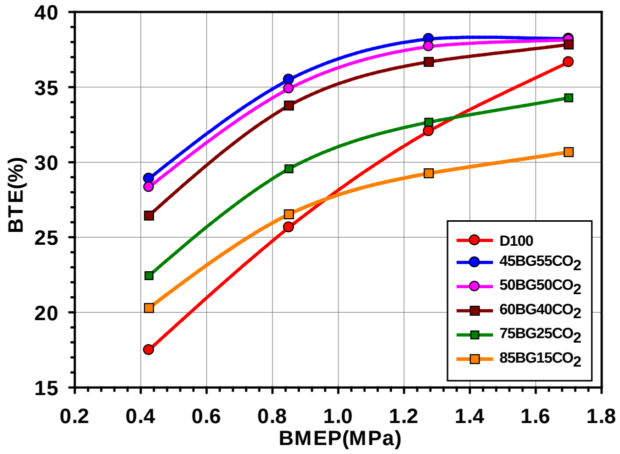


Fig. 10. Variations of BTE for D100 and biogas-diesel fuel.

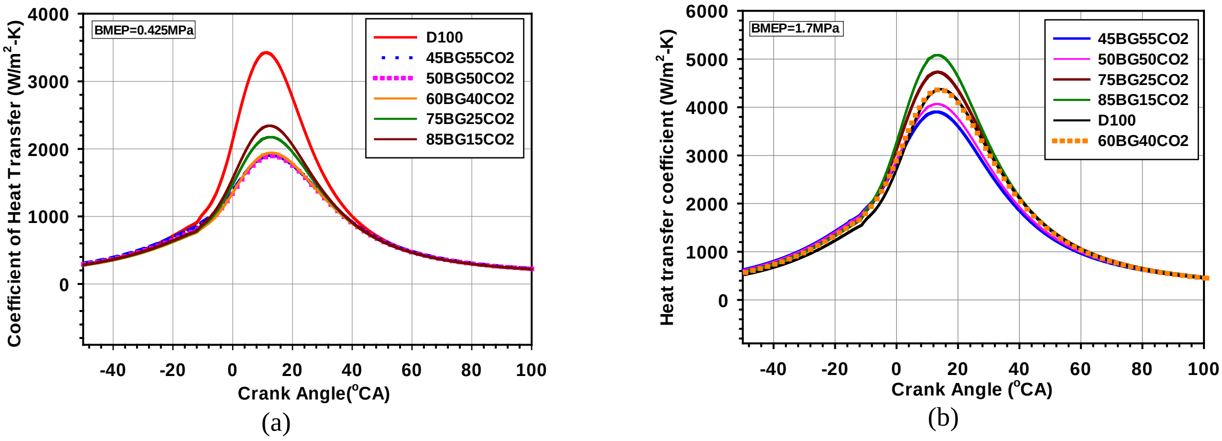


Fig. 11. Variations of HTC for D100 and biogas-diesel fuel a) BMEP=0.425MPa b)BMEP=1.7MPa.

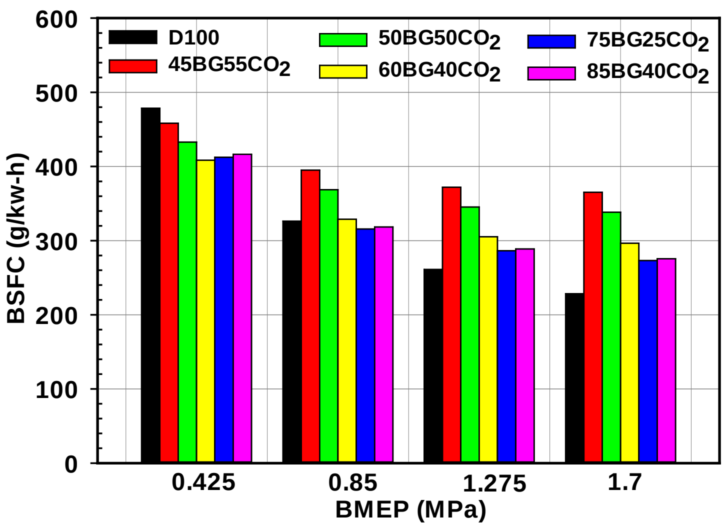


Fig. 12. Variations of BSFCS with engine load for D100 and biogas-diesel fuels.

4.1.3 Exhaust gas temperature (EGT)

Figure 13 illustrates the EGT for diesel and biogas-diesel fuel operations. The exhaust gas temperatures with biogas dual fuel operations were found to be higher than that of diesel fuel operations at lower loads .This is because lower methane and higher CO_2 concentrations in biogas which causes prolongation of ignition delay hence causes higher EGT at lower loads. The increase in EGT was in the range of 3.6-18.8% for biogas-diesel fuels as compared to diesel fuel operation at 0.45MPa. As seen in Figure 13, compared to diesel fuel, the biogas-diesel fuels (45BG55CO_2 and 50BG50CO_2) had lower EGT, with the largest decrease being 20.3% for 45BG55CO_2 at 1.7MPa of engine load. This decrease in EGT in dual fuel mode can be attributed to bulk combustion near TDC and reduced diffusion phase combustion. It was found that the lowest EGT was obtained for 45BG55CO_2 i.e.666.82^{o}C and the highest temperature was obtained for diesel i.e. 934.4^{o}C.The EGT for D100, 75BG25CO_2 and 85BG12CO_2 increased by an average of 5.2-26.1% as the engine load was augmented from 0.425MPa to 1.7MPa. 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 45BG55CO_2, 50BG50CO_2 and 60BG40CO_2 respectively as the engine load was augmented from 0.425MPa to 1.7MPa. This is due to the presence higher ratio of CO_2 in biogas absorbs some of the heat from the combustion reaction hence the EGT for the biogas with higher CO_2 percentages decreased. The presence of CO_2 had prolonged the ignition delay (ID). It is well known that CO_2 has a dilution effect which absorbs the heat energy, and decreases the local flame temperature, leading decreasing the EGT. In this study, the dilution effect of CO_2 was more prevalent than the prolonged ID hence; EGT was lower for 45BG55CO_2 at higher loads. However at lower loads the ID effect of CO_2 was more prevalent than the CO_2's dilution effect hence the EGT of 45BG55CO_2 was found to be higher than of diesel fuel operation.

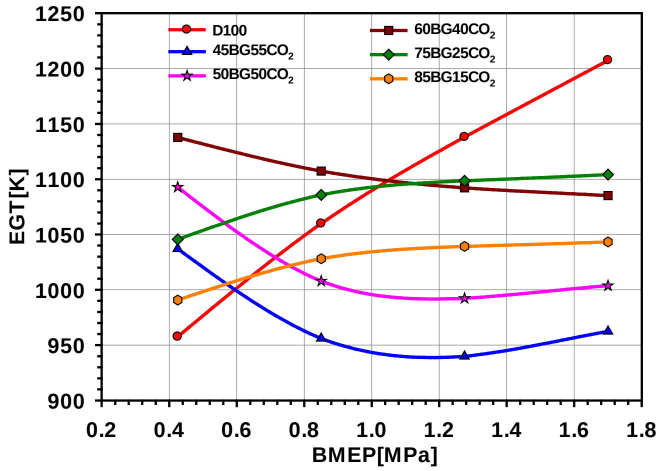


Fig. 13. Variations of EGT with load for D100 and Biogas-diesel fuels.

4.2. Combustion

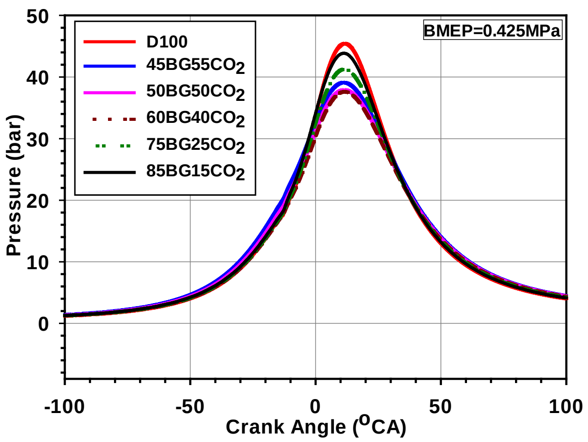
4.2.1 Cylinder Pressure

Figures 14 shows the cylinder pressure with respect to the crank angle for diesel and biogas-diesel dual fuel with various CO_2 percentages. As presented in Figures 14(a) and (b) peak cylinder pressures of biogas-diesel dual fuel operation were found to be lower than that of diesel fuel operations at lower loads of 0.425MPa and 0.85MPa.However 45BG55CO_2 and 85BG15CO_2 had marginally higher pressure than that of diesel fuel operation at an engine load of 0.85MPa. This is because at lower engine loads, lower admission of fuel energy leads to lower rate of HRR and consequently diminished cycle pressures and temperatures. In addition, the presence of a notable amount of CO_2 in biogas absorbed the heat release and hence diminished the engine pressure and temperatures. As shown in Table 5 the peaks of cylinder pressures for diesel occurred late after top dead center (ATDC) compared to those of biogas-diesel fuel operation. As depicted in Figure 14(a) at low load condition (0.425MPa), the maximum peak cylinder pressure was found to be 44.9bar (at 11.85◦CA aTDC), for D100 and was 3.9-17.2% higher than those of biogas-diesel fuel operations.

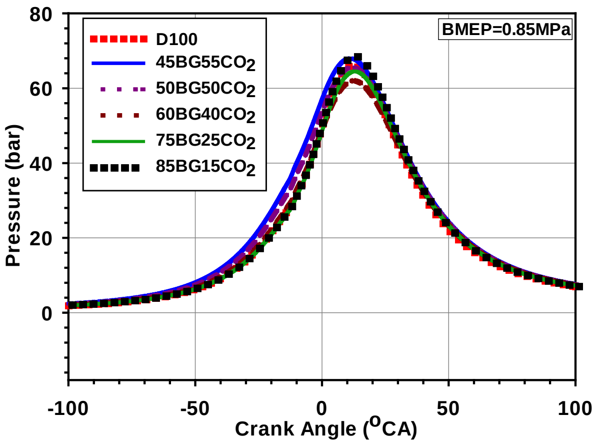
At BMEP of 1.275MPa the biogas-diesel fuels showed slightly higher cylinder pressure compared to that of diesel fuel operation. As indicated in Figure 14(d), it was found that the in-cylinder pressure for diesel was higher by 6.8% and 3.7% than that of 45BG55CO_2 and 50BG50CO_2 respectively; while it was lower by 2.3, 6.6 and 13.2% as compared to those of 60BG40CO_2, 75BG25CO_2 and 85BG15CO_2 respectively. The increase in pressure with biogas dual fuels is attributed to their higher ignition delays compared to that of diesel fuel operation .According to Liu and Karim [57], the influx of vaporous fuels in diesel engines fundamentally influences the physical and chemical forms during ignition delay and thus prompts its prolongation. Similar effects were observed by Verma *et al.* [36], who found higher peak cylinder pressures with biogas dual fuel engines. The cylinder pressure increased with an increasing engine load. The results showed that the increase in the cylinder pressure was about 140-172% for the biogas fuels when the engine load was increased from 0.425 to 1.7 MPa. This is due to the turbocharged and cooled induction of biogas with the intake-air charge, which caused longer ignition delay, hence increased the in cylinder pressure.

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

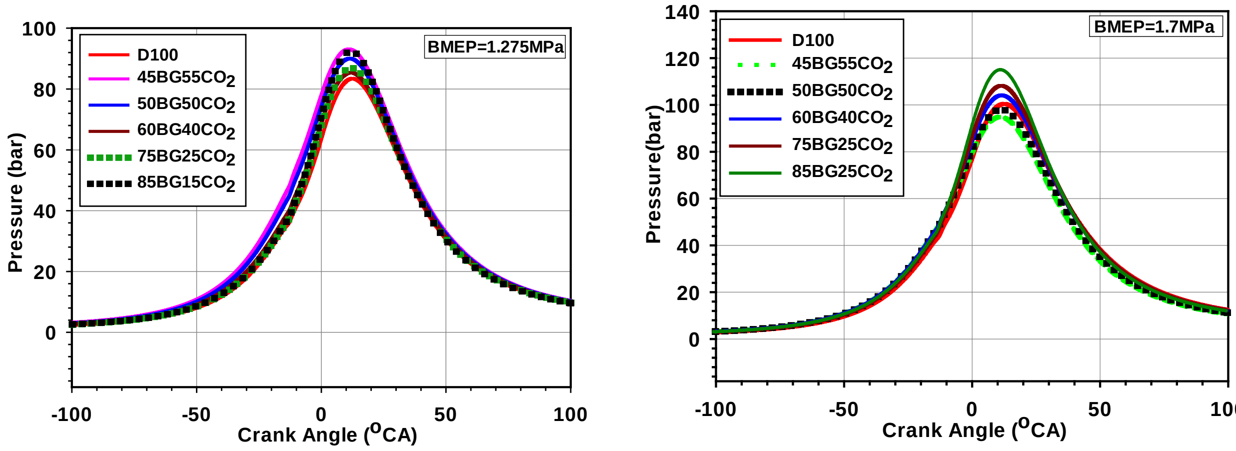
Load (Mpa)	Fuel	Maximum Cylinder Pressure(bar)	Position of pressure Peak/°CA	MMRP (bar/°CA)	Maximum HRR J/°CA	Position of Maximum HRR/°CA
0.425	45BG55CO2	38.22	11.23	1.07	85.26	10.65
	50BG50CO2	37.14	11.20	1.07	84.60	10.63
	60BG40CO2	36.96	11.78	1.14	86.18	10.00
	75BG55CO2	40.56	11.80	1.34	97.44	7.96
	85BG15CO2	43.11	11.20	1.49	106.16	6.99
	D100	44.88	11.85	1.70	116.82	6.13
0.85	45BG55CO2	66.04	11.24	1.79	142.73	10.00
	50BG50CO2	63.86	11.21	1.79	142.84	10.65
	60BG40CO2	60.52	11.75	1.81	141.12	10.62
	75BG55CO2	63.11	11.74	2.03	151.32	8.66
	85BG15CO2	67.12	11.75	2.26	164.34	7.70
	D100	65.20	12.37	2.31	162.70	7.71
1.275	45BG55CO2	89.98	11.23	2.45	197.98	10.00
	50BG50CO2	87.26	11.21	2.45	199.63	10.00
	60BG40CO2	82.94	11.74	2.47	197.30	10.00
	75BG55CO2	84.68	11.71	2.74	206.44	9.22
	85BG15CO2	90.02	11.73	3.04	223.49	7.56
	D100	82.04	12.30	2.77	202.06	9.58
1.7	45BG55CO2	91.79	11.22	2.52	205.78	10.00
	50BG50CO2	94.87	11.79	2.69	220.17	10.00
	60BG40CO2	100.84	12.30	3.04	243.51	10.65
	75BG55CO2	104.96	12.26	3.42	260.81	9.01
	85BG15CO2	111.54	11.71	3.79	281.80	7.91
	D100	98.49	12.81	3.19	243.60	10.61



(a)

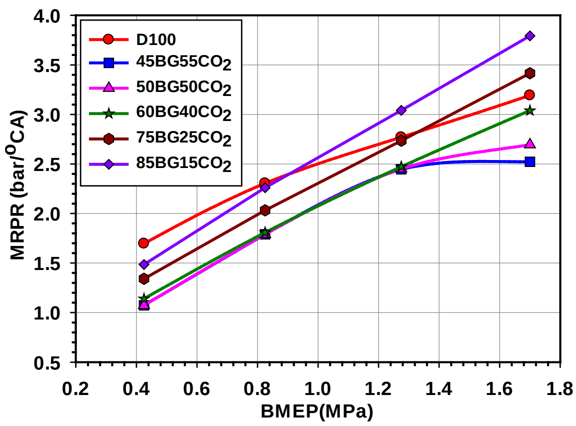


(b)



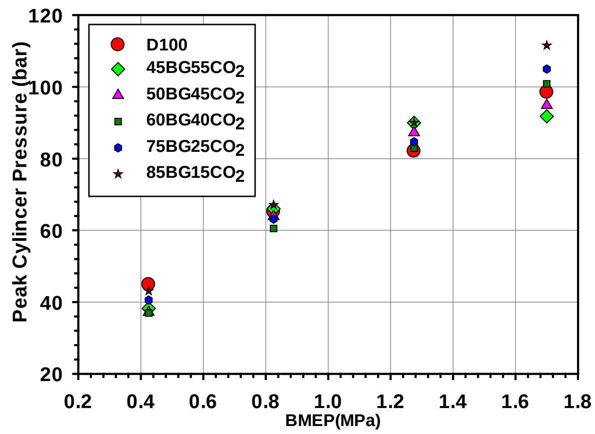
(c) (d)

Fig. 14. Variations of in-cylinder pressure with crank angle for diesel and biogas-diesel fuels (a) BMEP=0.45MPa, (b) BMEP=0.85MPa, (c) BMEP=1.275MPa and (d) BMEP=1.7MPa.



(a)

Fig. 15. (a) Variation of MRPR with load.



(b)

Fig. 15. (b)Variations of peak cylinder pressure with load.

Figure 15 (a) shows the maximum rate of pressure rise (MRPR) as a function of engine Load. As seen in Figure 11(a) the MRPR increased when the load increased for all the fuels given. The diesel fuel operation showed higher MRPR as compared to those of biogas-diesel dual fuel operation at 0.425MPa and 0.85MPa. However, at 1.7MPa of engine load 75BG25CO_2 and 85BG15CO_2 showed higher MRPR than that of diesel fuel operation.

The variation of the maximum cylinder pressure as a function of engine load for biogas-diesel and diesel fuel operations is depicted in Figure 15(b). As seen in Figure 15(b) the maximum pressure increased with an increase in engine load. The main reason behind the increase can be explained as follows, when the load increased, it implies that more fuel is burned hence the cylinder pressure for all the fuels increased. At lower engine loads of 0.425 and 0.85MPa the highest peak pressures of 44.8 and 67.1bar were recorded for D100 and 85BG15CO_2 fuels respectively. Biogas has a poor combustion at lower loads. At higher loads (1.7MPa), the highest pressure was found to be recorded 111.5bar for 85BG15CO_2.The higher methane concentration of 85BG15CO_2 was responsible for the higher pressure.

4.2.2. Heat release rate (HRR)

Figure 16 shows the variation of HRR as a function of crank angle for diesel and biogas-diesel fuels. As illustrated in these Figures, HRR increased with the increase of the percentage of methane in the biogas dual fuel operation with the increase being higher the higher the Ratio of methane in the biogas. As shown in Table. 5 the HRR of diesel fuel operation was found to be higher than those of biogas-diesel fuel with 25-55% (CO_2 by vol.) at lower loads of 0.425MPa and 0.85MPa. This is attributed to the decrease in lower heating values (LHVs) of biogas with decrease in methane concentrations. At low load condition (0.45MPa), the maximum HRR was obtained at 116.8kJ/^{o}CA (at 6.13^o CA aTDC), 85.3kJ/^{o}CA (at10.65^{o}CA), 84.6kJ/^{o}CA (at10.63^{o}CA aTDC) 97.4kJ/^{o}CA (at 7.96^o CA aTDC) and 106.2kJ/oCA (at 6.99^{o}CA aTDC), for D100, 45BG55CO_2, 50BG50CO_2, 60BG40CO_2, 75BG25CO_2 and 85BG15CO_2, respectively. At lower engine loads, lesser admission of fuel energy led to lower rate of heat release. At lower loads, the occurrences of the maximum HRR were found to be delayed in the dual fuel operation, as compared to that of diesel fuel. This is because of higher specific heat of biogas and the presence of CO_2 which caused retarded combustion. At higher loads (1.7MPa) the HRR of 75BG25CO_2 and 85BG15CO_2 were 7.1% and 15.7% higher than that of diesel fuel operation. The lower heating value of methane is higher than diesel fuel, and both 75BG25CO_2 and 85BG15CO_2 have higher methane concentrations that led to higher HRR.

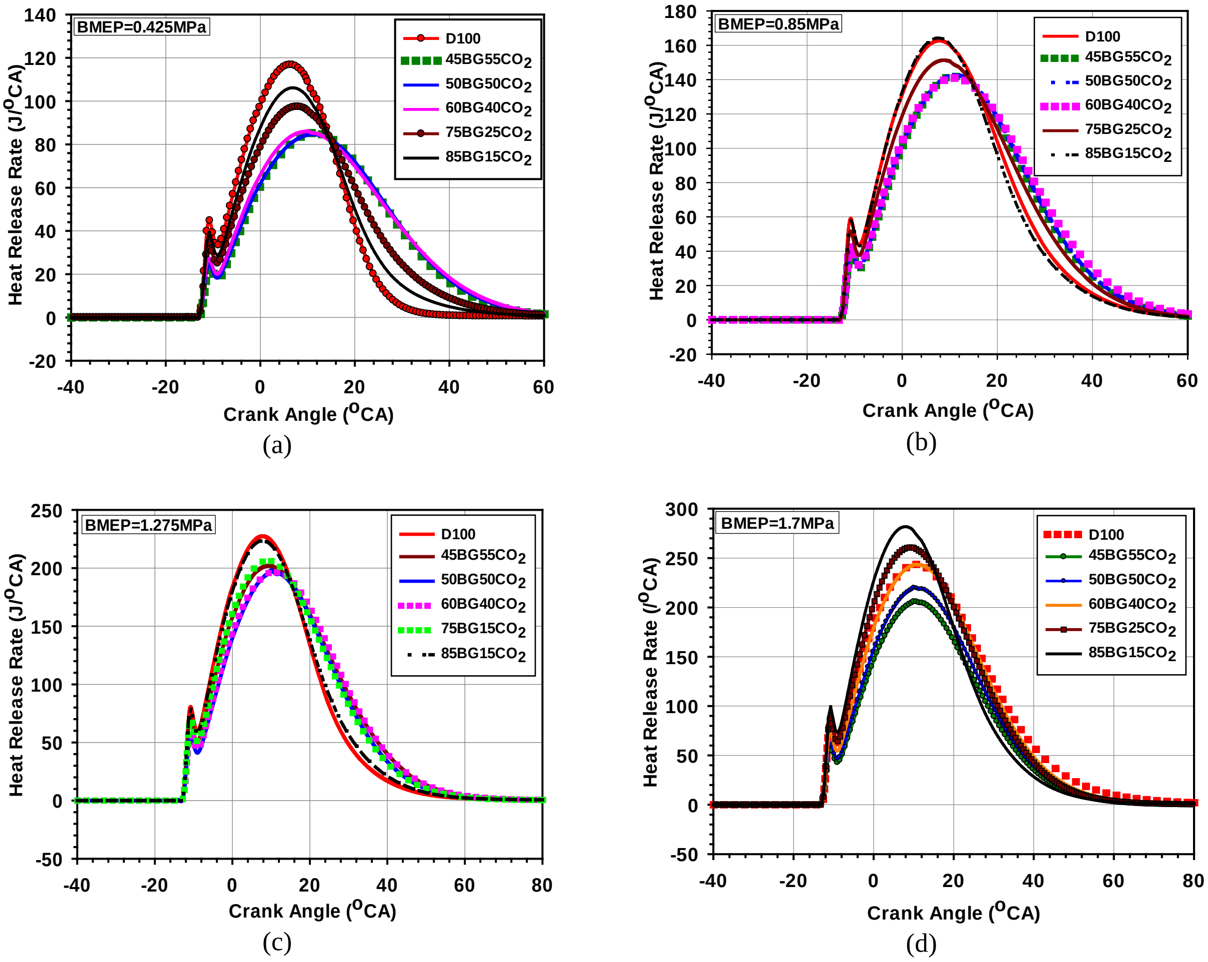


Fig. 16. Variations of HRR with crank angle for diesel and biogas-diesel fuels (a) BMEP=0.45MPa, (b) BMEP=0.85MPa, (c) BMEP=1.275MPa and (d) BMEP=1.7MPa.

4.2.3 Cumulative heat release (CHR)

The cumulative heat release (CHR) as a function of engine loads for diesel and biogas-diesel dual fuel operations are shown in the Figures 17(a) and 17(b) .It was found that at lower engine loads, diesel fuel operation had lower CHR compared to those of biogas-diesel fuels. At 25% of engine load (0.425MPa), CHR for diesel fuel operation was 2.86kJ as compared to 2.94, 3.00, 3.07, 2.93 and 2.86kJ for 45BG55CO_2, 50BG50CO_2, 60BG40CO_2, 75BG25CO_2 and 85BG15CO_2 respectively. At lower engine loads, most of the combustion chamber is filled with biogas and ignited by 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 engine loads, increased input fuel energy led to higher rate of heat release hence the CHR increased. At full engine load (1.7MPa), CHR for diesel, was found to be 9.04kJ which was 1.98-30.9% higher than those of biogas-diesel fuel operation. The lower heating value of biogas as compared to diesel fuel is the reason for lower CHR on dual-fuel operation. The CHR increased with increase in the engine load due to increased amount of liquid and gaseous fuel in the combustion chamber.

4.2.4 Volumetric efficiency ignition delay and equivalence ratio

Figure 18(a) shows the volumetric efficiency for diesel and biogas-diesel fuel operations. It was found that the volumetric efficiency of diesel fuel operation was higher than that of biogas-diesel dual fuel operation. The volumetric efficiency of diesel was found to be 90.8%,whereas, the volumetric efficiency of 45BG55CO_2, 50BG50CO_2, 60BG40CO_2, 75BG25CO_2 and 85BG55CO_2 was 86.5%, 86.9%, 87.1% and 87.3%, respectively, at BMEP of 1.7MPa. The induction of biogas with air through the intake manifold replaced some amount of fresh air which consequently decreased the volumetric efficiency in dual fuel operation.

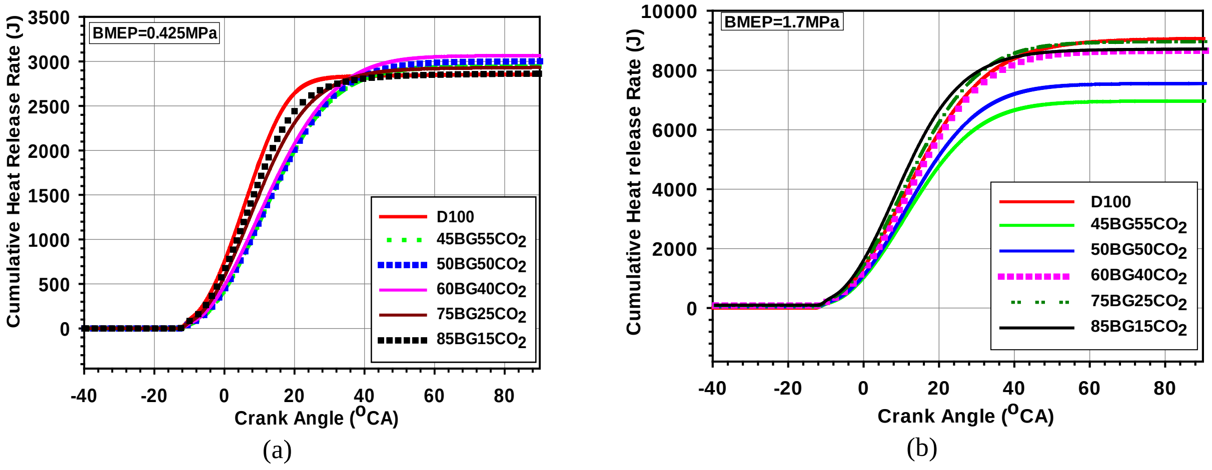


Fig. 17. Variation of cumulative heat release with crank angle a) 0.425MPa b) 1.7MPa.

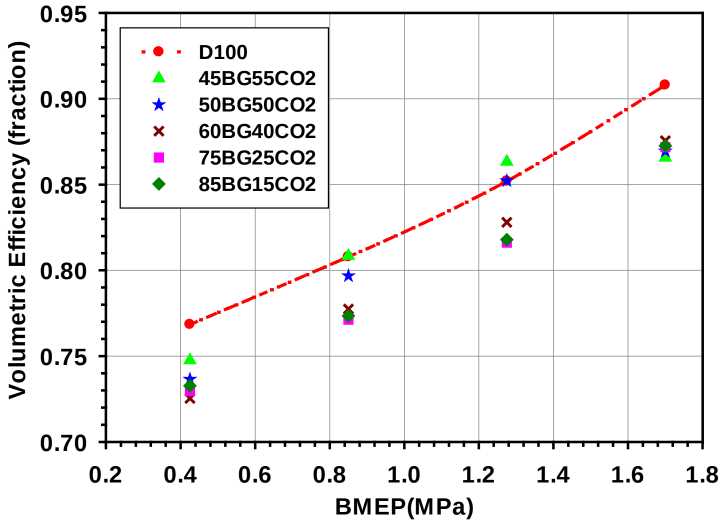


Fig. 18(a). Variation of volumetric efficiency with engine load.

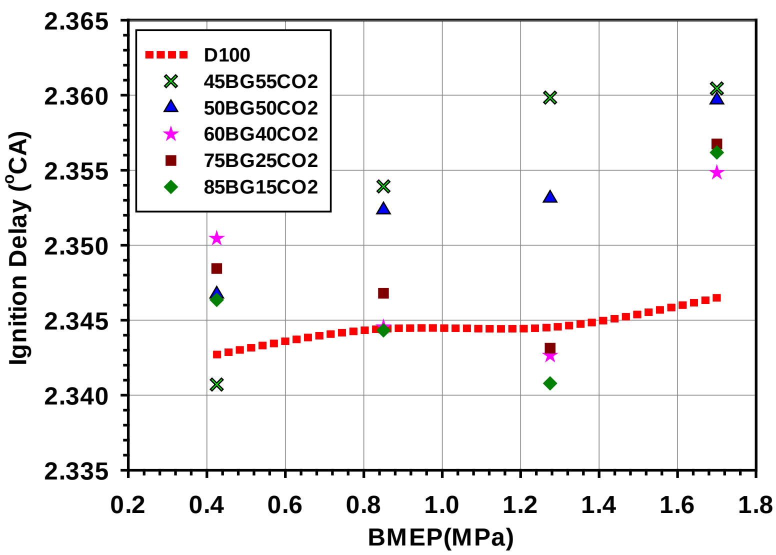


Fig. 18(b). Variation of ignition delay with engine load.

The ignition delay is one of the most important parameters which affect combustion significantly in the premixed phase [53]. Figure 18b shows the variation of ignition delay with engine load. In comparison with the operation of diesel fuel, the ignition delay of dual fuel operation was higher. Highest ignition delay was found with 45BG55CO_2 dual fuel operation as 2.3605CA compared to 2.34CA for diesel fuel operation. This is due to the biogas induction through the intake manifold, decreases the concentration of oxygen in the blend of air-fuel and changes the pre-ignition property of the charge. Furthermore, CO_2 concentration in biogas behaves like a diluent that absorbs a certain quantity of heat and lowers the temperature of the charge. All these effects lead to increasing ignition delays.

The term equivalence ratio is defined as the ratio of the actual air-fuel ratio to the stoichiometric air-fuel ratio [53]. Figure 18(c) shows the variation of the equivalence Ratio with engine load. As indicated in the figure 18(c) the dual-fuel mode combustion displayed higher fuel-air equivalence ratios compared to diesel fuel operation. The equivalence ratio of diesel was found to be in the range of 0.54-0.95, whereas, the equivalence ratio of 45BG55CO_2, 50BG50CO_2, 60BG40CO_2, 75BG25CO_2 and 85BG55CO_2 was 0.66-0.72, 0.71-0.79, 0.82-0.91, 0.99-1.08 and 1.102-1.21 respectively. This is because the biogas is drawn into the engine by the air inlet system, which reduces air volume in the combustion chamber.

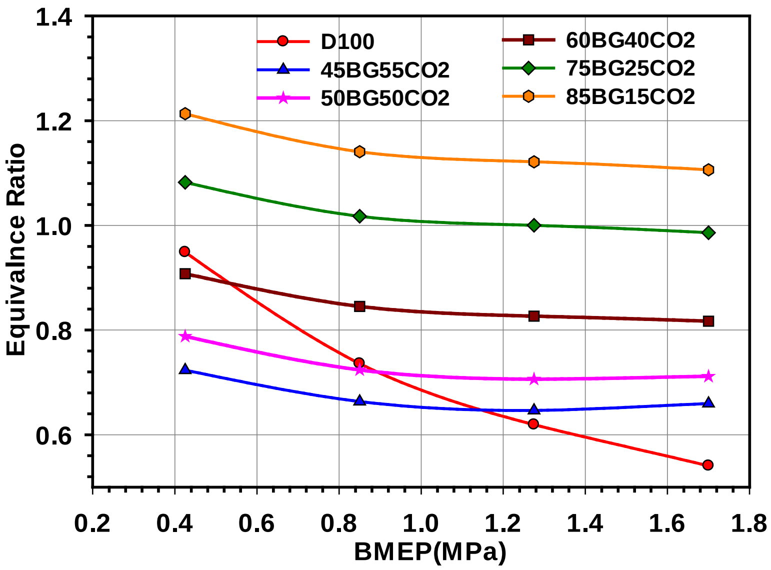


Fig. 18(c). Variation of equivalence ratio with engine load.

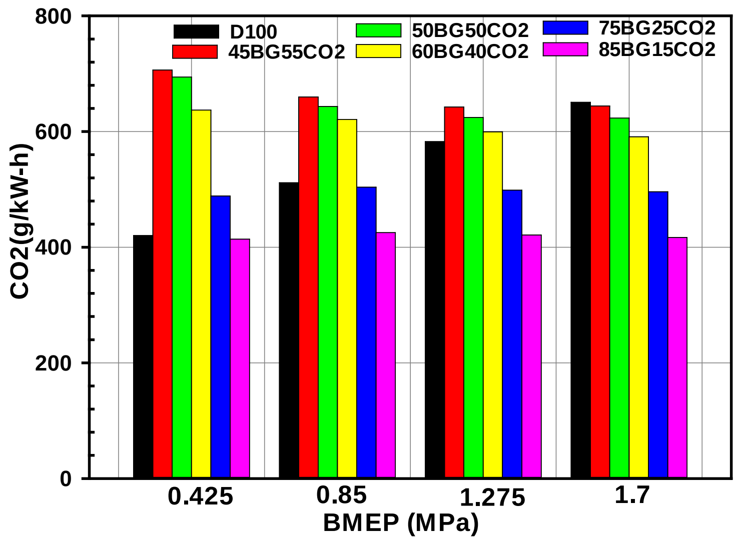


Fig. 19. Variation of CO_2 emissions with engine load.

4.3 Emissions

4.3.1 Carbon dioxide

Figure 19 depicts CO_2 emissions as a function of engine load. As illustrated in Figure 19, compared to 45BG55CO_2, the percentage change in CO_2 emissions of the biogas-diesel fuels (15-50% CO_2 vol.) decreased as level of CO_2 in biogas decreased. The maximum decrease in the change of CO_2 emitted was observed at 41.4% for 85BG15CO_2 at engine load of 0.425MPa as compared to 45BG55CO_2. This is because 45BG55CO_2 contains higher percentage of CO_2 (*i.e.* around 55% by volume) compared to the other biogas-diesel fuels. At 0.425MPa load, the fuels 45BG55CO_2, 50BG50CO_2 and 60BG40CO_2 had 68.1%, 65.2% and 51.5%, respectively higher CO_2 emissions than those of diesel fuel operation due to their higher CO_2 percentage. Similar results were reported by Sahoo [58] and Bora *et al.* [25] who found that higher CO_2 emission under dual fuel operation as compared to diesel fuel operation. However, at higher loads (1.7MPa) it was observed that the biogas-diesel mode had lower CO_2 emissions as compared to that of diesel fuel. The increase in CO_2 emissions with regard to diesel fuel operations can be attributed to the increase in combustion temperature that resulted in the oxidation of more amounts of CO into CO_2 [8].

4.3.2. Total hydrocarbon (THC)

HC emission as a function of engine load is depicted in Figure 19. As illustrated in the Figure 19 it was found that as CO_2 percentage in biogas increased, the HC emissions also increased. The biogas-diesel dual fuel operations have considerably higher HC emission in comparison to that of diesel fuel operations under all the operating conditions used. The biogas-diesel fuels had an average of more than 300% higher HC emission than that of diesel fuel operation. The HC emissions increased with increasing engine load for the biogas-diesel fuels while, it decreased for the diesel fuel. The injection of biogas caused rich mixture in combustion chamber and decreased the concentration of oxygen in air-fuel charge. Besides biogas has lower flame velocity [15]. These resulted in the incomplete combustion in dual fuel operation hence the HC emission increased. The CO_2 in the biogas-diesel mixture absorbs the heat and decreases the in-cylinder temperatures which slow down the hydrocarbon oxidation processes [5]. The flame velocity decreased as the high-specific heat charge reduced reaction rates, resulting in the rise of HC emissions .Moreover, other sources such as valve-overlapping, Improper mixing of liquid and gaseous fuel, effect from crevice volume and wall quenching also contribute to higher HC emissions[25]. Many researchers have reported similar results in HC emissions with biogas dual fuel operations [5], [10], [55]. Similar increase in HC trends in dual fuel mode with biogas can be found in Refs [18], [26], [59].

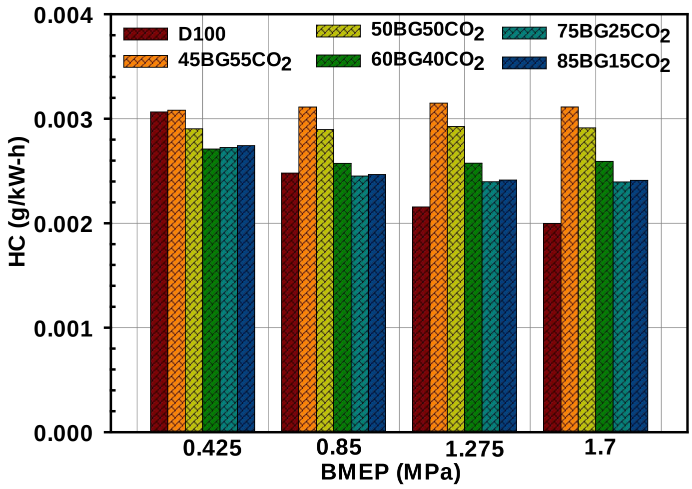


Fig. 19.Variations of HC emissions with engine load.

4.3.3. Nitrogen oxides

Figure 20 shows NOx emissions as a function of engine load. The NOx discharges in dual fuel operation are essentially lower. As seen in Figure 20, the addition of biogas fuels decreased NO_X emissions significantly for all engine loads, the decrease being higher the higher the CO_2 percentage in biogas. As compared to diesel fuel operation, NOx emissions for biogas-diesel decreased an average of 50.5-85.2%. 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 O_2 concentration in the air-fuel mixture (in view of decline in volumetric efficiency) which ultimately cut down the rate of NOx generation. Additionally, the presence of CO_2 in biogas, which has high specific heat, prompts reduced in-cylinder temperature by absorbing some amount of heat. Combined impacts of these reasons bring down NOx discharges in dual fuel operation with biogas. In diesel fuel operation the high in-cylinder temperature caused increased NOx emissions. It was observed that NOx emissions increased with the decreasing ratio of CO_2 in the biogas-diesel fuel for instance, the change in NOx emissions was compared to 45BG55CO_2 and showed that the NOx increase was in the range of 27-37% for 50BG50CO_2 and it was in the range of 55-140% for 60BG40CO_2. The amount of oxygen and the higher methane Ratio in biogas may increase oxygen-rich regions inside the cylinder which cause higher NO_X formation. The NOx emission of 45BG55CO_2 was found to be the lowest among the fuels considered.

NOx emissions increased with an increasing engine load. As the engine load increased, the in-cylinder temperature also increased hence, NOx emissions increased with respect to higher engine loads as indicated in Figure 20. The increase in NO_X emissions at higher loads is due higher combustion temperature which resulted in greater NO_X generation. The results of this study are in agreement with the finding of Barik and

Murugan [5] and Lounici *et al.* [18]. Verma *et al.* [55] reported similar NOx emission trend in biogas-diesel dual fuel using three compositions of biogas (containing 93%, 84% and 75% of CH_4 by volume).They also obtained that NO_X emission decreased by an average of 28.7% for biogas-diesel fuel operation as compared to diesel fuel operation. Similar decrease in NO_X emission with biogas-diesel dual fuel operation can be found in [17], [25], [26].

Figures 21 (a) and (b) show the NO_X distribution for D100 and biogas-diesel at 0.425MPa and 1.7MPa of engine loads. As seen in Figure 17(a) low NO_X emissions were observed at lower loads, with minimum emissions (*i.e.* 145.5ppm) recorded for 45BG55CO_2. As seen in the Figure 21(b), NO_X emissions were found to be higher when the engine load increased due to higher combustion temperatures. The maximum NOx emission was found to be 1847.1ppm for D100 and 828.9ppm for 85BG15CO_2, at 1.7MPa load.

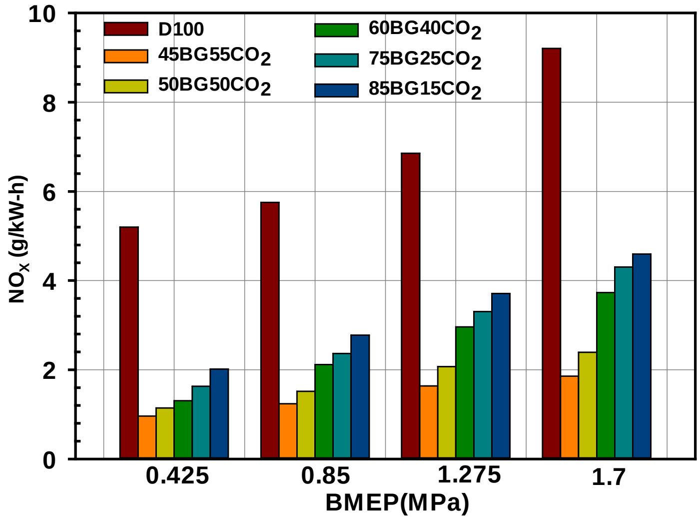
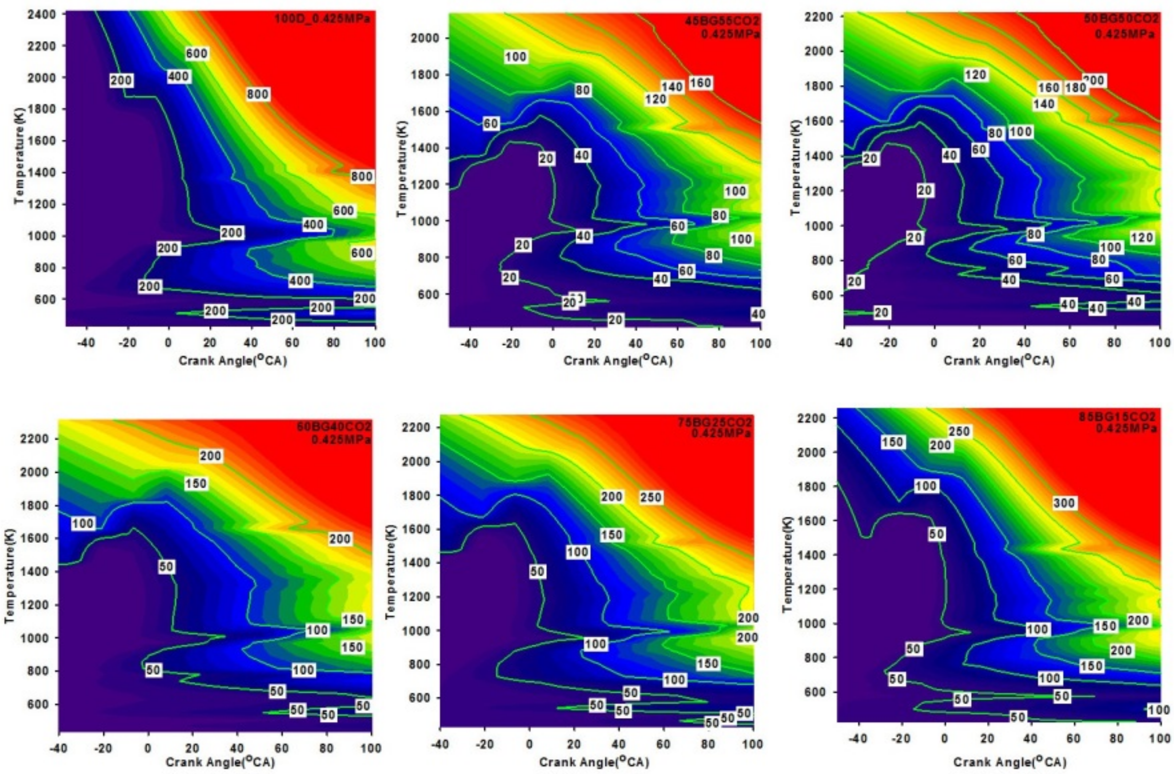
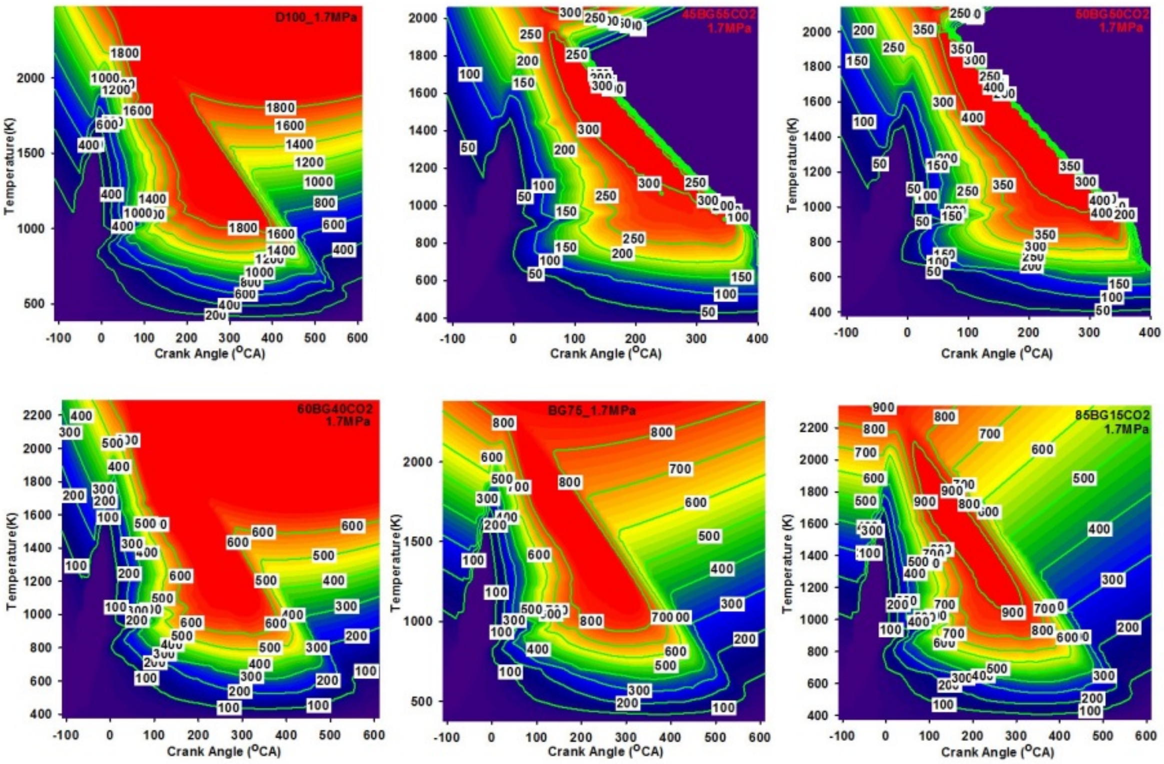


Fig. 20. Variation of NO_X emissions with engine load.



(a)



(b)

Fig. 21. (a) NO_X distribution for BMEP=0.425MPa 4 (b) NO_X distribution for BMEP=1.7MPa.

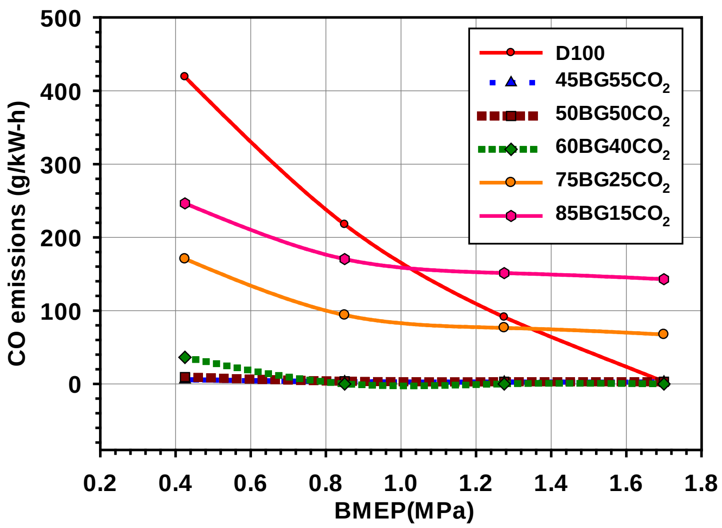


Fig. 22. Variations of CO emissions with load for diesel and biogas-diesel.

4.3.4. Carbon monoxide (CO)

Carbon monoxide as a function of engine load for diesel and biogas-diesel dual fuel is depicted in Figure 22 .It was observed that the CO emissions of biogas-diesel fuels were lower than that of diesel fuel operation for all the engine loads except for $75BG25CO_2$ and $85BG15CO_2$ had slightly higher CO emissions at 1.7MPa of BMEP. This might be due to slightly higher air-fuel ratio with dual fuel operation than that of diesel fuel operation. At 0.85MPa engine load, air-fuel Ratio for $45BG55CO_2$, $50BG50CO_2$, $60BG40CO_2$,$75BG25CO_2$ and $85BG15CO_2$ were found to be 12.10, 12.26, and 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 increased. When the engine load increased, the combustion temperature increased which resulted in a higher conversion of CO to CO_2 hence, CO emissions decreased. Higher CO emissions were observed at lower loads, however at higher loads CO emissions decreased significantly, with minimum emissions recorded at 1.275MPa engine load for the biogas-diesel fuels. The D100 recorded the maximum value of CO emissions at 0.425MPa engine load. The increase of CO at lower loads is due to the deficiency of oxygen, lower combustion chamber temperature, and less time for combustion, which led to incomplete combustion, causing more CO emission while at higher loads CO emissions were reduced due to more complete combustion. Similar results can be found in [24].

5. CONCLUSION

In the present work, numerical investigation has been carried out to examine the effects CO_2 percentage in biogas on the performance, combustion and exhaust emission of a diesel engine. Following is concluded:

1. The BSFC for diesel fuel operation was found to be lower than those of biogas-diesel fuel for all engine loads except at 0.45MPa where it had slightly higher BSFC than Biogas-diesel fuel operation. Among the biogas-diesel fuels the lowest BSFC were recorded for 75BG25CO_2 i.e. 273.3g/kW-h where as that of diesel was found to be 228.4g/kW-h.
2. The use of Biogas-diesel fuels improved the BTE. The BTE values for the biogas with CO_2 Ratio of 40-55% of by vol. were on average 3.84% higher than that of diesel fuel operation while those with CO_2 Ratio of 15-25% by vol. the BTE were found to be lower by an average of 11.4%.A maximum of BTE value i.e. 38.22% was recorded for 45BG55CO_2. The CO_2 Ratio of biogas does not impact BTE considerably.
3. Biogas-diesel provided superior performance in reductions of NO_X emission. NOx emission for the biogas-diesel dual fuel operation was found to be lower than that of diesel fuel operation due to decreased in in-cylinder temperature and increase in dilution effect. The NOx emission was found to be reduced by an average of 65-83% for the different biogas-diesel fuel operations. Due to its higher CO_2 Ratio 45BG55CO_2 had the lowest NOx emission among the fuels. The use of biogas-diesel also caused a decrease in the emissions of CO. On an average there was a decrease of CO emission by 97.4%, 73.03%, and 76.51% for 60BG40CO_2, 50BG50CO_2 and 45BG55CO_2, respectively in comparison to diesel fuel operation.
4. The combustion characteristics of diesel fuel combustion and biogas-diesel indicated similar patterns at various engine loads. The peak pressure and heat release rate for biogas-diesel were lower compared to diesel fuel operation. At 0.425MPa and 0.85MPa engine loads, biogas-diesel combustion showed slightly higher peak pressure than those of diesel fuels.
5. The exhaust gas temperatures (EGT) were slightly higher for biogas-diesel combustions compared to diesel fuel operations at lower engine loads. However, at higher loads the EGT for diesel fuel were higher than those of diesel fuel operation due to the decreased charge temperature of biogas-diesel fuels.
6. With the introduction of biogas, the HC emissions for biogas-diesel fuels increased sharply compared to neat diesel fuel operation under all load conditions. The HC emissions of the biogas-diesel fuels were on average 350% greater than diesel fuels. Similarly the CO_2 emissions of biogas-diesel fuel were found to be higher than those of diesel fuel operations under all load conditions except at 1.7MPa, where the CO_2 emissions of diesel were found to be higher. CO_2 increased with increase in CO_2 content in biogas. The CO_2 emission for biogas-diesel fuels were on average 25-35% higher in comparison to diesel fuel operation.

The biogas is a renewable fuel which can be used in dual-fuel mode in the diesel engine without any modification to an engine. It is renewable and inexpensive besides it is easily accessible. Based on the results of this investigation, it can be concluded that the biogas generated from pig manure and corn straw can make a good substitute for diesel fuel. Significant improvements on exhaust emissions and combustion characteristics can be achieved. The engine performance did not deteriorate with up to 45% CO_2 content biogas.

NOMENCLATURE

Q_{comb}	Total heat released
dQ_n	Apparent heat release rate
Q_{ht}	Heat lost to the cylinder walls
P	Cylinder pressure
T_g	The gas instantaneous temperature
T_w	Cylinder wall temperature
A	Cylinder heat transfer area
h_g	Heat transfer coefficient
V	Cylinder volume
u	Characteristic velocity
B	Cylinder bore diameter
γ	Ratio of specific heats
BTE	Brake thermal efficiency
^oCA	degrees of crank angle
aTDC	After top dead center
BMEP	Brake mean effective pressure
CO	Carbon monoxide
CO_2	Carbon dioxide
NO_X	Nitrogen oxides
HC	Hydrocarbon
BSFC	Brake specific fuel consumption
ppm	Parts per million
EGT	Exhaust gas temperature
MRPR	Maximum rate of pressure rise

REFERENCES

[1] Jiang D., Zhuang D., Fu J., Huang Y., and Wen K., 2012. Bioenergy potential from crop residues in

China: Availability and distribution. *Renewable and Sustainable Energy Reviews* 16(3): 1377-1382.

[2] Tian Y., 2012. Potential assessment on biogas production by using livestock manure of large-scale farm in China. *Transactions of the Chinese Society of Agricultural Engineering* 28(8): 230-234.

[3] Wang Y., Li G., Chi M., Sun Y., Zhang J., Jiang S., and Cui Z., 2018. Effects of co-digestion of cucumber residues to corn stover and pig manure ratio on methane production in solid state anaerobic digestion. *Bioresource Technology* 250: 328-336.

[4] Environmental Protection Department, 2017. Data & Statistics, Hong Kong Air Pollutant Emission Inventory. [On-line serial], Retrieved February 12, 2019 from the World Wide Web: https://www.epd.gov.hk/epd/english/environmentinhk/air/data/emission_inve.html.

[5] Barik D., and Murugan S., 2014. Investigation on combustion performance and emission characteristics of a DI (direct injection) diesel engine fueled with biogas–diesel in dual fuel mode. *Energy* 72: 760-771.

[6] Subramanian K., Mathad V.C., Vijay V., and Subbarao P., 2013. Comparative evaluation of emission and fuel economy of an automotive spark ignition vehicle fuelled with methane enriched biogas and CNG using chassis dynamometer. *Applied Energy* 105: 17-29.

[7] Deeba F., Kumar V., Gautam K., Saxena R., and Sharma D., 2012. Bioprocessing of Jatropha curcas seed oil and deoiled seed hulls for the production of biodiesel and biogas. *Biomass and Bioenergy* 40: 13-18.

[8] Ganzoury M.A., and Allam N.K., 2015. Impact of nanotechnology on biogas production: a mini-review. *Renewable and Sustainable Energy Reviews* 50: 1392-1404.

[9] Kavuma C., 2013. Variation of methane and carbon dioxide yield in a biogas plant. MSc. Thesis. KTH Royal Institute of Technology,Stockholm, Sweden

[10] Ambarita H., 2017. Performance and emission characteristics of a small diesel engine run in dual-fuel (diesel-biogas) mode. *Case studies in thermal engineering* 10: 179-191.

[11] Homdoung N., Tippayawong N., and Dussadee N., 2015. Performance and emissions of a modified small engine operated on producer gas. *Energy Conversion and Management* 94: 286-292.

[12] Henham A., and Makkar M., 1998. Combustion of simulated biogas in a dual-fuel diesel engine. *Energy Conversion and Management* 39(16-18): 2001-2009.

[13] Park C., Park S., Lee Y., Kim C., Lee S., and Moriyoshi Y., 2011. Performance and emission characteristics of a SI engine fueled by low calorific biogas blended with hydrogen. *International Journal of hydrogen energy* 36(16): 10080-10088.

[14] Luijten C., and Kerkhof E., 2011. Jatropha oil and biogas in a dual fuel CI engine for rural electrification. *Energy Conversion and Management* 52(2): 1426-1438.

[15] Bari S., 1996. Effect of carbon dioxide on the performance of biogas/diesel duel-fuel engine. *Renewable Energy* 9(1-4): 1007-1010.

[16] Nathan S.S., Mallikarjuna J., and Ramesh A., 2010. An experimental study of the biogas–diesel HCCI mode of engine operation. *Energy Conversion and Management* 51(7): 1347-1353.

[17] Makareviciene V., Sendzikiene E., Pukalskas S., Rimkus A., and Vegneris R., 2013. Performance and emission characteristics of biogas used in diesel engine operation. *Energy Conversion and Management* 75: 224-233.

[18] Lounici M.S., Loubar K., Tazerout M., Balistrou M., and Tarabet L., 2014. Experimental investigation on the performance and exhaust emission of biogas-diesel dual-fuel combustion in a CI engine. *SAE 2014 International Powertrain, Fuels & Lubricants Meeting.* Birmingham, United Kingdom , 20-23 October. Pennsylvania: SAE international.

[19] Papacz W., 2011. Biogas as vehicle fuel. *Journal of KONES* 18: 403-410.

[20] IRENA, 2017. *Biogas for road vehicles: Technology brief.* Abu Dhabi: International Renewable Energy Agency.

[21] Bhatia S., 2014. *Advanced renewable energy systems,(Part 1 and 2).* New York:WPI Publishing.

[22] Porpatham E., Ramesh A., and Nagalingam B., 2012. Effect of compression ratio on the performance and combustion of a biogas fuelled spark ignition engine. *Fuel* 95: 247-256.

[23] Ramesha D., Bangari A.S., Rathod C.P., and Samartha C R., 2015. Experimental investigation of biogas-biodiesel dual fuel combustion in a diesel engine. *Journal of Middle European Construction and Design of Cars* 13(1): 12-20.

[24] Yoon S.H., and Lee C.S., 2011. Experimental investigation on the combustion and exhaust emission characteristics of biogas–biodiesel dual-fuel combustion in a CI engine. *Fuel Processing Technology* 92(5): 992-1000.

[25] Bora B.J., Saha U.K., Chatterjee S., and Veer V., 2014. Effect of compression ratio on performance, combustion and emission characteristics of a dual fuel diesel engine run on raw biogas. *Energy conversion and management* 87: 1000-1009.

[26] Mustafi N.N., Raine R.R., and Verhelst S., 2013. Combustion and emissions characteristics of a dual fuel engine operated on alternative gaseous fuels. *Fuel* 109: 669-678.

[27] Ibrahim M.M, Narasimhan J.V., and Ramesh A., 2015. Comparison of the predominantly premixed charge compression ignition and the dual fuel modes of operation with biogas and diesel as fuels. *Energy* 89: 990-1000.

[28] Yilmaz I., and Gumus M., 2017. Investigation of the effect of biogas on combustion and emissions of TBC diesel engine. *Fuel* 188: 69-78.

[29] Bora B.J., and Saha U.K., 2015. Comparative assessment of a biogas run dual fuel diesel engine with rice bran oil methyl ester, pongamia oil methyl ester and palm oil methyl ester as pilot fuels. *Renewable Energy* 81: 490-498.

[30] Xin Z., Jian X., Shizhuo Z., Xiaosen H., and Jianhua L., 2013. The experimental study on cyclic variation in a spark ignited engine fueled with biogas and hydrogen blends. *International Journal of Hydrogen Energy* 38(25): 11164-11168.

[31] Cacua K., Amell A., and Cadavid F., 2012. Effects of oxygen enriched air on the operation and performance of a diesel-biogas dual fuel engine. *Biomass and Bioenergy* 45: 159-167.

[32] Pattanaik P., Nayak C., and Nanda K., 2013. Investigation on utilization of biogas & Karanja oil biodiesel in dual fuel mode in a single cylinder DI diesel engine. *International Journal of Energy and Environment* 4(2): 279-290.

[33] Bora J., and Saha U K., 2016. Experimental evaluation of a rice bran biodiesel–biogas run dual fuel diesel engine at varying compression ratios. *Renewable Energy* 87: 782-790.

[34] Kalsi S., and Subramanian K., 2017. Effect of simulated biogas on performance, combustion and emissions characteristics of a bio-diesel fueled diesel engine. *Renewable Energy* 106: 78-90.

[35] Barik D., Murugan S., Sivaram N., Baburaj E., and Sundaram S., 2017. Experimental investigation on the behavior of a direct injection diesel engine fueled with Karanja methyl ester-biogas dual fuel at different injection timings. *Energy* 118: 127-138.

[36] Verma S., Das L., Kaushik S., and Tyagi S., 2018. An experimental investigation of exergetic performance and emission characteristics of hydrogen supplemented biogas-diesel dual fuel engine. *International Journal of Hydrogen Energy* 43(4): 2452-2468.

[37] Chandra R., Vijay V., Subbarao P., and Khura T., 2011. Performance evaluation of a constant speed IC engine on CNG, methane enriched biogas and biogas. *Applied Energy* 88(11): 3969-3977.

[38] Al Seadi T., 2008. *Biogas handbook*. Esbjerg: Syddansk Universitet.

[39] Vasco-Correa J., Khanal S., Manandhar A., and Shah A., 2018. Anaerobic digestion for bioenergy production: Global status, environmental and techno-economic implications, and government policies. *Bioresource Technology* 247: 1015-1026.

[40] Bidart C., Fröhling M., and Schultmann F., 2014. Livestock manure and crop residue for energy generation: macro-assessment at a national scale. *Renewable and Sustainable Energy Reviews* 38: 537-550.

[41] Lönnqvist T., Silveira S., and Sanches-Pereira A., 2013. Swedish resource potential from residues and energy crops to enhance biogas generation. *Renewable and Sustainable Energy Reviews* 21: 298-314.

[42] Zhang Z., O'Hara M., Mundree S., Gao B., Ball S., Zhu N., Bai Z., and Jin B., 2016. Biofuels from food processing wastes. *Current Opinion in Biotechnology* 38: 97-105.

[43] Park J., Lee J., Sim J., and Lee J-H., 2009. Production of hydrogen from marine macro-algae biomass using anaerobic sewage sludge microflora. *Biotechnology and Bioprocess Engineering* 14(3): 307.

[44] Mushtaq K., Zaidi A., and Askari J., 2016. Design and performance analysis of floating dome type portable biogas plant for domestic use in Pakistan. *Sustainable Energy Technologies and Assessments* 14: 21-25.

[45] Feng Y., Zhang Y., Quan X., and Chen S., 2014. Enhanced anaerobic digestion of waste activated sludge digestion by the addition of zero valent iron. *Water Research* 52: 242-250.

[46] Holm-Nielsen B., Al Seadi T., and Oleskowicz-Popiel P., 2009. The future of anaerobic digestion and biogas utilization. *Bioresource Technology* 100(22): 5478-5484.

[47] Christy M., Gopinath L., and Divya D., 2014. A review on anaerobic decomposition and enhancement of biogas production through enzymes and microorganisms. *Renewable and Sustainable Energy Reviews* 34: 167-173.

[48] Mao C., Feng Y., Wang X., and Ren G., 2015. Review on research achievements of biogas from anaerobic digestion. *Renewable and Sustainable Energy Reviews* 45: 540-555.

[49] Feng R., Zaidi A., Zhang K., and Shi Y., 2019. Optimisation of Microwave Pretreatment for Biogas Enhancement through Anaerobic Digestion of Microalgal Biomass. *Periodica Polytechnica Chemical Engineering* 63(1): 65-72.

[50] Kummamuru B., 2016. WBA global bioenergy statistics 2017. *World Bioenergy Association*,Sweden.

[51] IRENA. 2017. Renewable Energy Statistics 2017. [On-line serial], Retrieved January 20, 2019 from the World Wide Web: https://www.irena.org/publications/2017/Jul/Renewable-Energy-Statistics-2017.

[52] Feroskhan M., and Ismail S., 2016. Investigation of the effects of biogas composition on the performance of a biogas–diesel dual fuel CI engine. *Biofuels* 7(6): 593-601.

[53] Heywood B., 1988. I*nternal Combustion Engine Fundamentals*.NewYork: McGraw-Hill.

[54] Ferguson R., and Kirkpatrick A., 2015. *Internal Combustion Engines: Applied Thermosciences*. New Jersey :John Wiley & Sons.

[55] Verma S., Das L., and Kaushik S., 2017. Effects of varying composition of biogas on performance and emission characteristics of compression ignition engine using exergy analysis. *Energy Conversion and Management* 138: 346-359.

[56] Karagöz M., Sarıdemir S., Deniz E., and Çiftçi B., 2018. The effect of the CO2 ratio in biogas on the vibration and performance of a spark ignited engine. *Fuel* 214: 634-639.

[57] Liu Z., and Karim G., 1998. An examination of the ignition delay period in gas-fueled diesel engines.

Journal of Engineering for Gas Turbines and Power 120(1): 225-231.

[58] Sahoo B., 2011. Clean development mechanism potential of compression ignition diesel engines using gaseous fuels in dual fuel mode. PhD. Thesis. Indian Institute of Technology, Guwahati, India.

[59] Duc P. M, and Wattanavichien K., 2007. Study on biogas premixed charge diesel dual fuelled engine. *Energy Conversion and Management* 48(8): 2286-2308.