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Effects of charge preheating on the performance of a biogas-diesel dual fuel CI engine

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ABSTRACT

This study explores the viability of preheating the intake charge as a means to enhance the performance of a compression ignition (CI) engine operated on biogas and diesel in dual fuel mode. Biogas is the primary fuel, which is mixed with air in the intake, preheated and inducted into the engine. This mixture is compressed and subsequently ignited by means of pilot diesel injection within the cylinder. The influence of charge temperature and biogas flow rate on brake thermal efficiency, volumetric efficiency, diesel fuel consumption, diesel equivalent brake specific fuel consumption, exhaust gas temperature and overall equivalence ratio are investigated under two speeds and various loads. Charge preheating, low biogas flow rates and high speed operation are observed to enhance brake thermal efficiency. Methane enrichment is effective in improving thermal efficiency at low biogas flow rates. While charge preheating and increasing the biogas flow rate reduce diesel fuel consumption, volumetric efficiency is lowered. Displacement of air by biogas increases the overall equivalence ratio and exhaust gas temperature. Low speed mode is characterised by reduced diesel fuel consumption and high volumetric efficiency. Under low speed operation, biogas can provide more than 90% of the total energy release.

1. Introduction

The rapid depletion of fossil fuel reserves and environmental concerns have prompted various researchers to explore alternative fuel options for internal combustion engines in recent times [1–3]. Biogas is a promising alternative fuel for internal combustion engines due to its low cost and ease of production. The main constituents of biogas are methane (CH_4) and carbon dioxide (CO_2), which are present in the ranges of 55 to 70% and 25 to 50% by volume respectively [4,5]. The presence of CO_2 reduces the ignitability and calorific value [6,7]. This can be partly overcome by methane enrichment – a generic name for various physico-chemical processes used to remove CO_2, thereby increasing the combustible methane fraction of biogas. Biogas can be used in compression ignition (CI) engines in two modes, viz. dual fuel and homogeneous charge compression ignition (HCCI) modes. In both cases, biogas is mixed with the incoming air and inducted into the cylinders. The difference lies in the method used to ignite this combustible mixture. In dual fuel mode, it is achieved by injecting a small quantity of diesel into the cylinder, while in HCCI, the biogas-air mixture is compressed until auto-ignition occurs. Attainment of self-ignition temperature usually requires the use of manifold heating.

The application of biogas in CI engines has been studied over the past few years by various researchers [8–14]. The use of biogas in dual fuel mode reduces brake thermal efficiency, volumetric efficiency and increases Brake Specific Fuel Consumption (BSFC) compared to conventional CI engine mode. This is attributed to the presence of CO_2 which results in low combustion temperature, low flame speed, early occurrence of peak pressure and higher pumping work [11,15–17]. However, diesel fuel consumption can be significantly reduced by employing dual fuel mode. More than 48% savings in diesel fuel consumption has been reported in an IDI engine operated in dual fuel mode with biogas as primary fuel [11]. Fuel conversion efficiencies of diesel-only and biogas-diesel modes were almost same at full load operation, while part load efficiency was lower in the biogas-diesel mode [18]. The drop in brake thermal efficiency can be partly overcome by using methane-enriched biogas, high compression ratio or high load in dual fuel mode [17,19]. It has been observed that biogas containing CH_4 and CO_2 in the ratio close to 7:3 provides high brake thermal efficiency. This is possibly due to the dissociation of CO_2 into CO and O_2, providing a fast-burning mixture which improves combustion [20,21]. Increasing the oxygen content in the intake air from

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21% to 27% has been shown to improve the brake thermal efficiency from 15 to 18% [22].

Sahoo et al. [21] have also observed that biogas with lower CO_2 fraction (0.2 to 0.3) provides lower BSFC. Mustafi et al. [18] have reported that the brake specific energy consumption (BSEC) of the dual fuel mode is comparable to that of conventional CI engine operation. Brake thermal efficiency and volumetric efficiency are observed to drop at higher biogas flow rates [23,24]. The drop in volumetric efficiency is caused by displacement of fresh air in the intake by biogas [25]. Similar trends have been reported by Barik and Sivalingam [23,26].

Up to 30% diesel fuel substitution on energy basis has been reported at full load in a study conducted on a biogas-diesel dual fuel engine with manually controlled biogas flow rate and governor-controlled pilot fuel injection [23]. Luijten and Kerkhof [25] have reported up to 35% diesel substitution using biogas with methane fraction of 0.7. Methane enrichment has been shown to enhance diesel fuel substitution [13].

Considering the benefits outlined above, methane-enriched biogas (75% CH_4 and 25% CO_2 by volume) is utilised in dual fuel mode in the present work. The calculated properties of the constituents of biogas are listed in Table 1. As there is a scarcity of literature related to the use of charge preheating in biogas-diesel dual fuel engines, this study focuses on investigating the effects of intake temperature, biogas flow rate and methane fraction of biogas on various performance parameters.

2. Experimental setup

A schematic and photograph of the experimental setup are provided in Fig. 1. A conventional four stoke, water cooled, CI engine is modified to operate in dual fuel mode. The engine governor allows the operating speed to be set to a constant value between 1500 and 2200 rpm. The specifications of the engine rig are given in Table 2. Biogas is replicated by mixing methane and CO_2 (called simulated biogas) which are stored in two separate cylinders. CH_4 and CO_2 are supplied via flow control valves and specifically calibrated glass-tube rotameters. This helps in regulating the CH_4:CO_2 ratio as well as the flow rates. Biogas with 75% CH_4 and 25% CO_2 has been used for the present study. The mixture is fed into the incoming air stream. A 2.4 kW nichrome resistance heater is wound around the intake manifold and insulated externally in order to heat the air-biogas mixtures to temperatures as high as 120 °C. The test setup is equipped with a rope-brake type mechanical dynamometer, a burette with stop-watch arrangement, thermocouples and orifice meter to measure the engine brake torque, diesel flow rate, exhaust gas temperature and air flow rate respectively. The accuracy of the instruments and uncertainty analysis of the output parameters [27] are provided in Tables 3 and 4 respectively.

Table 1
Properties of biogas constituents.

Property		Value	
		Methane (CH_4)	Carbon dioxide (CO_2)
Composition (% by volume)		75	25
Lower calorific value (MJ/kg)		50	–
Density at 1 atm (kg/m^3)	35 °C	0.73	1.99
	60 °C	0.67	1.83
	80 °C	0.63	1.72
	100 °C	0.59	1.63
Stoichiometric air–fuel ratio (kg of air/kg of fuel)		17.24	–

3. Result and discussions

3.1. Performance characteristics

The performance of the dual fuel engine is studied by varying the biogas flow rate (Q_{bg}) and intake temperature (T_{in}) in the ranges 0 to 16 L/min and 35 to 100 °C respectively. Initially, performance tests are conducted at the operating speed (n) of 1900 rpm (referred to as high speed mode) for the full range of applied torque with biogas-diesel fuelling. The operating parameters for the high speed mode are listed in Table 5, constituting100 cases (5 values of torque, 5 biogas flow rates and 4 intake temperatures). Methane fraction is maintained at 75% in these cases. Subsequently, the governor setting is changed to n = 1550 rpm (referred to as low speed mode) for minimum diesel injection and tests are performed at the constant torque (T) of 5.5 Nm. Methane-diesel as well as biogas-diesel fuelling are used to study the effect of methane enrichment. While methane flow rate (Q_{CH4}) is varied in the range 6 to 12 L/min, carbon dioxide flow rate (Q_{CO2}) is varied from 0 to 4 L/min providing methane fractions of 100% and 75%. The operating parameters for the low speed mode are summarised in Table 6, constituting 18 cases (3 biogas flow rates each for 100% and 75% methane fraction, with 3 intake temperatures). The effects of the operating parameters on various performance indices are discussed below.

3.1.1. Equivalence ratio

Fig. 2(a) shows the effect of biogas flow rate and charge preheating on overall equivalence ratio under high speed mode at a constant applied torque of 11 Nm. The displacement of fresh air during biogas induction results in an increase in overall equivalence ratio. Fig. 2(b) shows that an increase in torque causes a rise in equivalence ratio. This is attributed to the rise in the quantity of injected diesel. As charge preheating reduces the density of both biogas and air, the net effect on equivalence ratio is insignificant. Fig. 2(c) compares the overall equivalence ratio for biogas-diesel and methane-diesel modes under low speed mode. As biogas displaces more air than methane, equivalence ratio is higher. This effect is more pronounced for lower flow rates (6 + 0 and 6 + 2 L/min).

3.1.2. Volumetric efficiency

Fig. 3(a) depicts the relationship between volumetric efficiency, biogas flow rate and intake temperature under high speed mode with a constant torque of 11 Nm. Results show that increasing the biogas flow rate causes a reduction in volumetric efficiency due to the displacement of fresh air. The same trend is available in few literatures [13,26]. With increase in temperature, there is a significant drop in the volumetric efficiency. This is due to the reduction in air density which results in lower air intake. Up to 6% reduction in volumetric efficiency is observed when temperature is raised from 35 to 100 °C. Fig. 3(b) reveals the engine torque has relatively less effect on volumetric efficiency.

The variations of volumetric efficiency with intake temperature for methane-diesel and biogas-diesel operation under low speed mode are shown in Fig. 3(c). While charge preheating is again found to cause a drop in volumetric efficiency, the effect is relatively lower in biogas compared to methane. In general, the low speed mode provides slightly higher volumetric efficiency compared to the high speed mode, most likely due to better filling of charge and lower operating temperatures.

3.1.3. Diesel fuel consumption

Fig. 4(a) shows diesel fuel consumption for various biogas flow rates and intake temperatures for a constant torque of 11 Nm

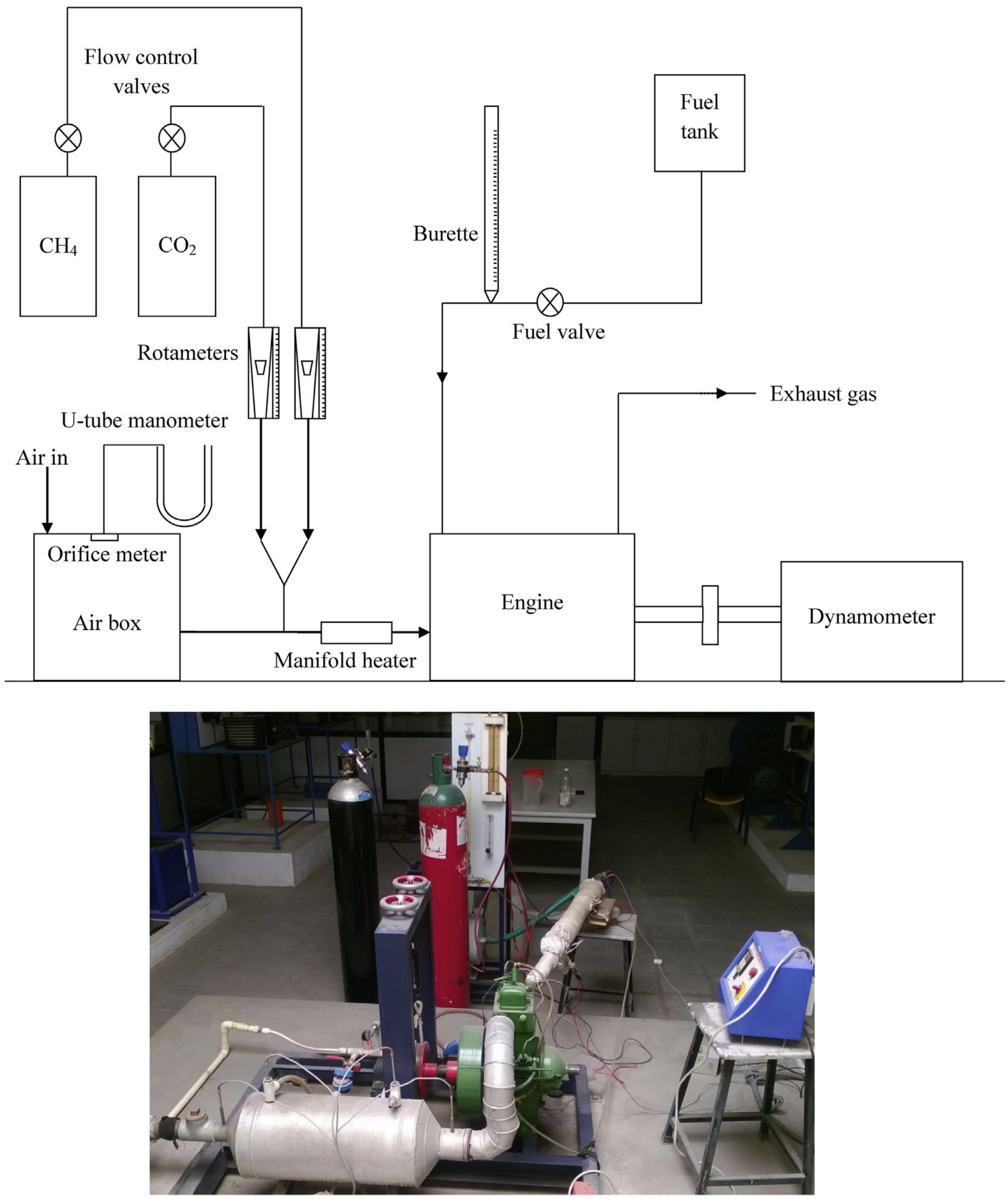

Fig. 1. Experimental setup.

Table 2
Engine specifications.

Parameter	Value
Engine	Kirloskar AV1XL model
Bore	0.0875m
Stroke	0.080m
Cubic capacity	481 cc
Peak pressure	73.6 bar
Number & arrangement of cylinders	1-Vertical
Working cycle	4-stroke diesel
Combustion principle	Compression ignition
Compression ratio	17:1
Maximum power	5.97 kW
Maximum torque	25 Nm
Rated speed	2200 rpm
Injection system	Direct injection
Number of nozzle holes	3
Injector opening pressure (bar)	210
Fuel injection timing	28° bTDC

Table 3
Details of measuring devices.

Quantity measured	Measuring device	Least count
Flow rate of CH_4	Rotameter (calibrated for methane)	0.5 L/min
Flow rate of CO_2	Rotameter (calibrated for CO_2)	0.5 L/min
Torque	Mechanical dynamometer	0.1 kg
Air flow rate	Orifice meter and differential water manometer	1 mm
Diesel flow rate	Burette	0.2 ml
Temperature	Thermocouples	0.1 °C
Speed	Digital tachometer	10 rpm

under high speed mode. As the biogas flow rate is increased, it contributes more towards the input fuel energy, so that the governor mechanism cuts down the diesel supply. However, at low intake temperatures, some amount of biogas remains incompletely

Table 4
Error analysis.

Output parameter	% of error (±)
Brake thermal efficiency	4%
Volumetric efficiency	3%
Diesel fuel consumption	1%
Overall equivalence ratio	3%
Brake specific fuel consumption	4%

Table 5
Details of engine operating parameters in high speed mode (n = 1900 rpm).

Torque, T (Nm)	Biogas and component flow rates, Q_{bg} (Q_{CH4}+Q_{CO2}) (l/min)	Intake temperature, T_{in} (°C)
0	0 (0 + 0)	35
5.5	4 (3 + 1)	60
11	8 (6 + 2)	80
16.5	12 (9 + 3)	100
22	16 (12 + 4)	–

Table 6
Details of engine operating parameters in low speed mode (n = 1550 rpm).

Methane fraction (% by volume)	Biogas and component flow rates, Q_{bg} (Q_{CH4}+Q_{CO2}) (l/min)	Intake temperature, T_{in} (°C)
100	6 (6 + 0)	80, 90, 100
	9 (9 + 0)	
	12 (12 + 0)	
75	8 (6 + 2)	
	12 (9 + 3)	
	16 (12 + 4)	

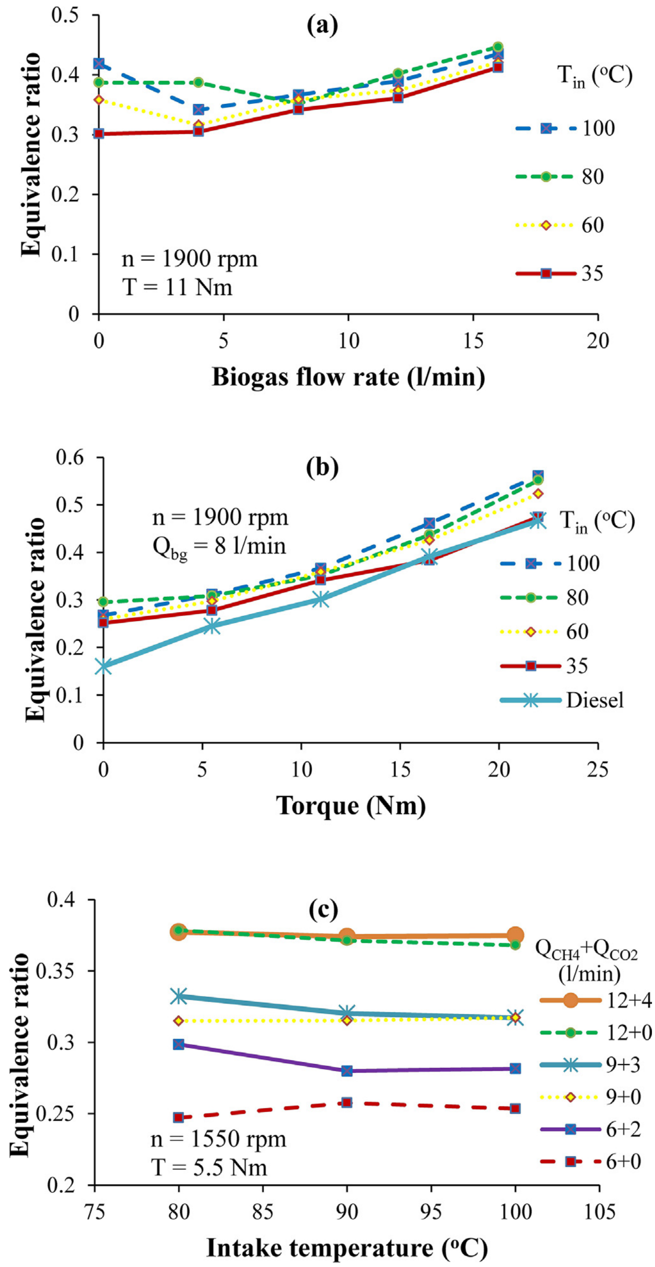

Fig. 2. Effect of (a) biogas flow rate, (b) torque and (c) intake temperature on overall equivalence ratio.

burned. Charge preheating makes air-biogas mixtures more combustible and hence provides a slight reduction in diesel fuel consumption. Charge preheating is detrimental in diesel-only operation, as more diesel is injected to compensate the reduction in air intake. As expected, diesel fuel consumption increases with engine torque as shown in Fig. 4(b).

Fig. 4(c) illustrates the effect of intake temperatures on diesel fuel consumption for biogas-diesel and methane-diesel operation under low speed mode. Compared to methane-diesel operation, biogas-diesel operation entails greater diesel fuel consumption at low gas flow rates (6 + 0 and 6 + 2 L/min) owing to the negative effect of CO_2 on methane combustion. For higher flow rates, the effect of CO_2 is insignificant, as the elevated temperatures support methane combustion. While operating at 12 + 0 and 12 + 4 L/min, biogas accounts for more than 90% of the total energy release.

3.1.4. Brake specific fuel consumption

BSFC is calculated based on total equivalent diesel fuel consumption, i.e. the diesel quantity which has the combined energy content of biogas and diesel in dual fuel mode. Fig. 5(a) illustrates the variation of BSFC with biogas flow rate and intake temperature at the applied load of 11 Nm at 1900 rpm. The least fuel consumption per unit work output occurs at low biogas flow rate and high intake temperature. Charge preheating provides 10 to 20% improvement in BSFC. High biogas flow rate causes an increase in BSFC, confirming earlier findings [15,26].

Fig. 5(b) shows the variations of BSFC with torque and intake temperature at the constant biogas flow rate of 8 L/min under high speed mode. Energy conversion efficiency and hence BSFC improve with increase in torque.

The effect of charge preheating on BSFC at a constant torque of 5.5 Nm under low speed mode is depicted in Fig. 5(c). Low biogas flow rate with methane enrichment shows better BSFC. At high biogas flow rate, the effect of biogas composition on BSFC is negligible.

3.1.5. Brake thermal efficiency

Fig. 6(a) illustrates the variation of brake thermal efficiency with intake temperature and biogas flow rate at the applied load of 11 Nm under high speed mode. It can be observed that at all intake temperatures, highest brake thermal efficiency is obtained with the biogas flow rate of 4 L/min, reaffirming earlier findings from a non-preheated engine [17,26]. This can be attributed to the fact that the supplied biogas mixes with the available excess air and undergoes combustion. This provides optimum utilisation of the excess air, which otherwise has only a detrimental effect due to flow losses. Higher biogas flow rates are accompanied by a displacement of air, affecting the mixture formation and the

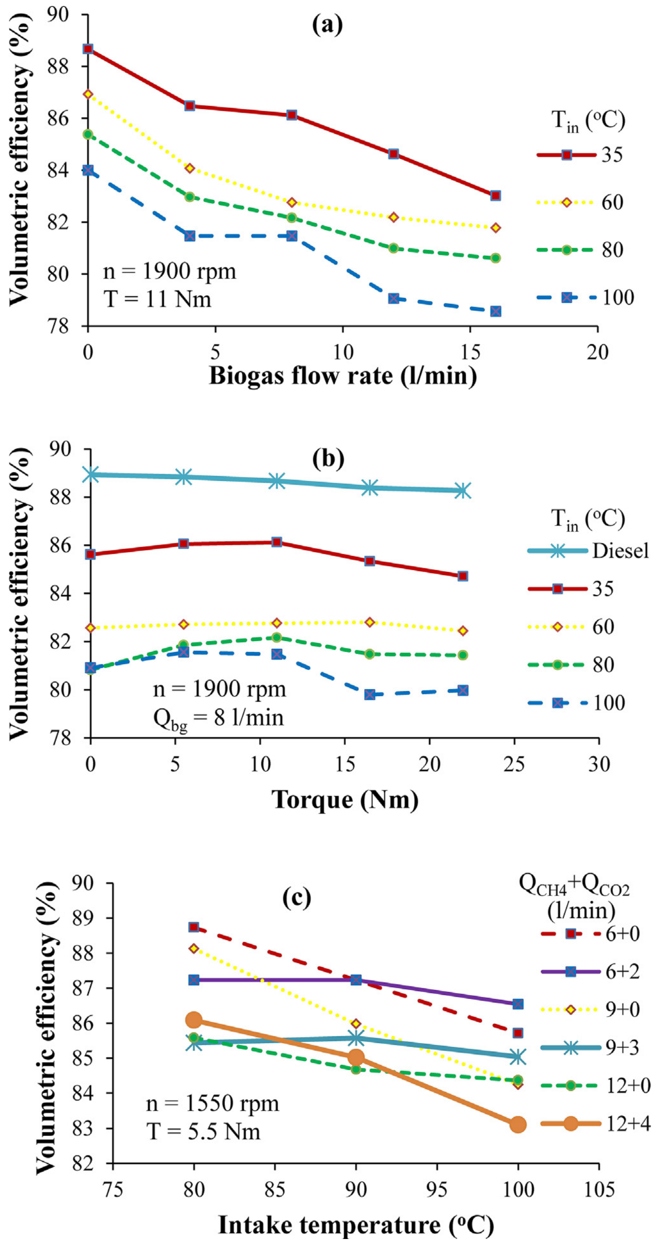

Fig. 3. Effect of (a) biogas flow rate, (b) torque and (c) intake temperature on volumetric efficiency.

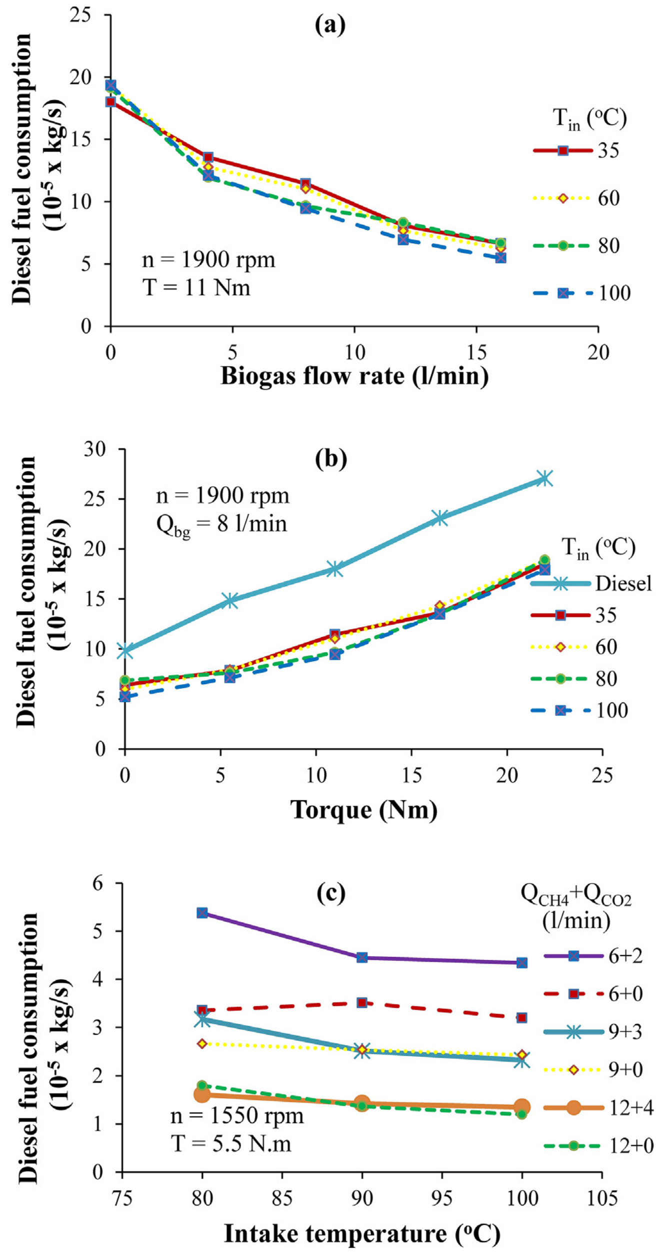

Fig. 4. Effect of (a) biogas flow rate, (b) torque and (c) intake temperature on diesel fuel consumption.

completeness of combustion. This is reflected in the fall in thermal efficiency at higher biogas flow rates. In the case of diesel-only operation (0 L/min biogas flow rate), charge preheating causes a reduction in air density and flow rate, leading to the formation of relatively richer mixtures, reducing the brake thermal efficiency. However, in biogas-diesel dual fuel mode, the brake thermal efficiency is found to increase significantly with charge preheating. This may be because biogas combustion can proceed to a greater degree of completion at elevated temperatures.

Fig. 6(b) depicts the variations of the brake thermal efficiency with torque and intake temperature at the constant biogas flow rate of 8 L/min under high speed mode. It is seen that the improvement in thermal efficiency due to charge preheating is more significant at higher loads.

The effect of charge preheating on thermal efficiency at a constant torque of 5.5 Nm under low speed mode is depicted in Fig. 6(c). Both methane-diesel and biogas-diesel modes are considered here with different flow rates of the gaseous fuels. For clarity, the flow rates of CH_4 and CO_2 are shown separately in the legend. For instance, the entry 6 + 2 indicates a mixture of 6 L/min CH_4 and 2 L/min CO_2. While the brake thermal efficiencies at low speed are lower than those of high speed, primarily due to higher coolant losses, charge preheating is again observed to improve the thermal efficiency. Also, a comparison of the 6 + 0 L/min (methane-diesel) and 6 + 2 L/min (biogas-diesel) cases shows that thermal efficiency is improved by methane enrichment. However, methane enrichment shows no significant effect at higher flow rates (e.g. 12 + 0 and 12 + 4 L/min).

3.1.6. Exhaust gas temperature

Fig. 7(a) depicts the relationship between exhaust gas temperature and biogas flow rate at different intake temperatures under high speed mode with a constant torque of 11 Nm. Results show a rise in exhaust gas temperature with increase in biogas flow rate.

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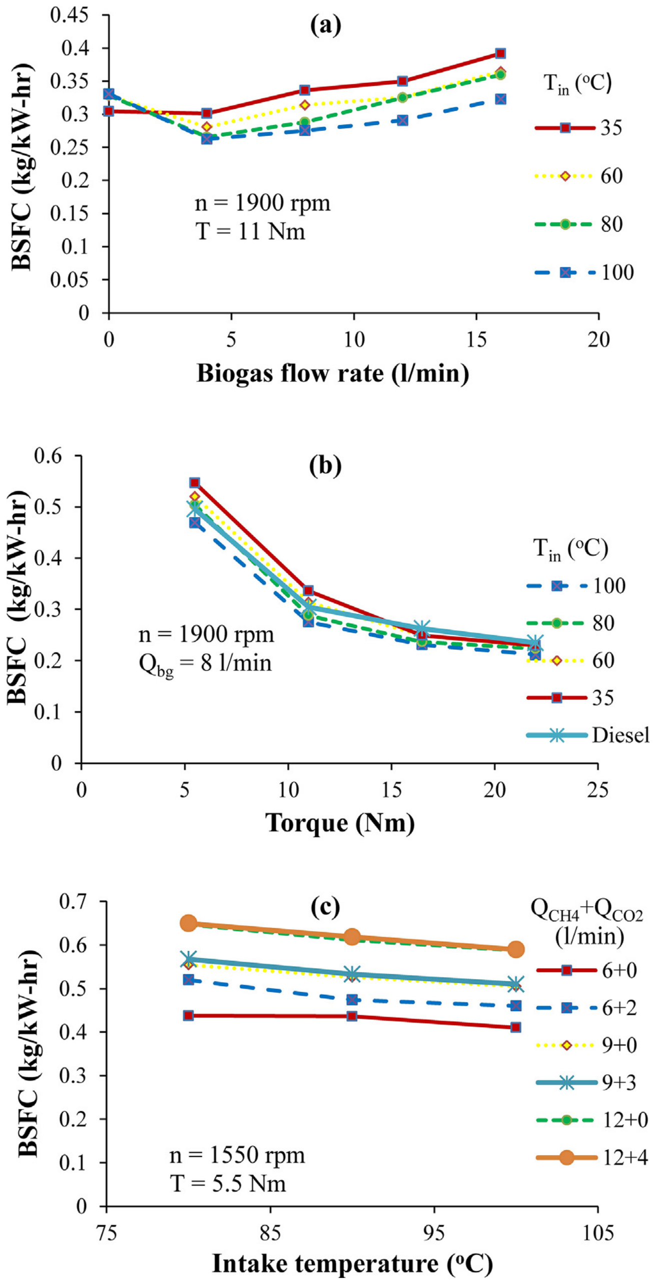

Fig. 5. Effect of (a) biogas flow rate, (b) torque and (c) intake temperature on diesel equivalent BSFC.

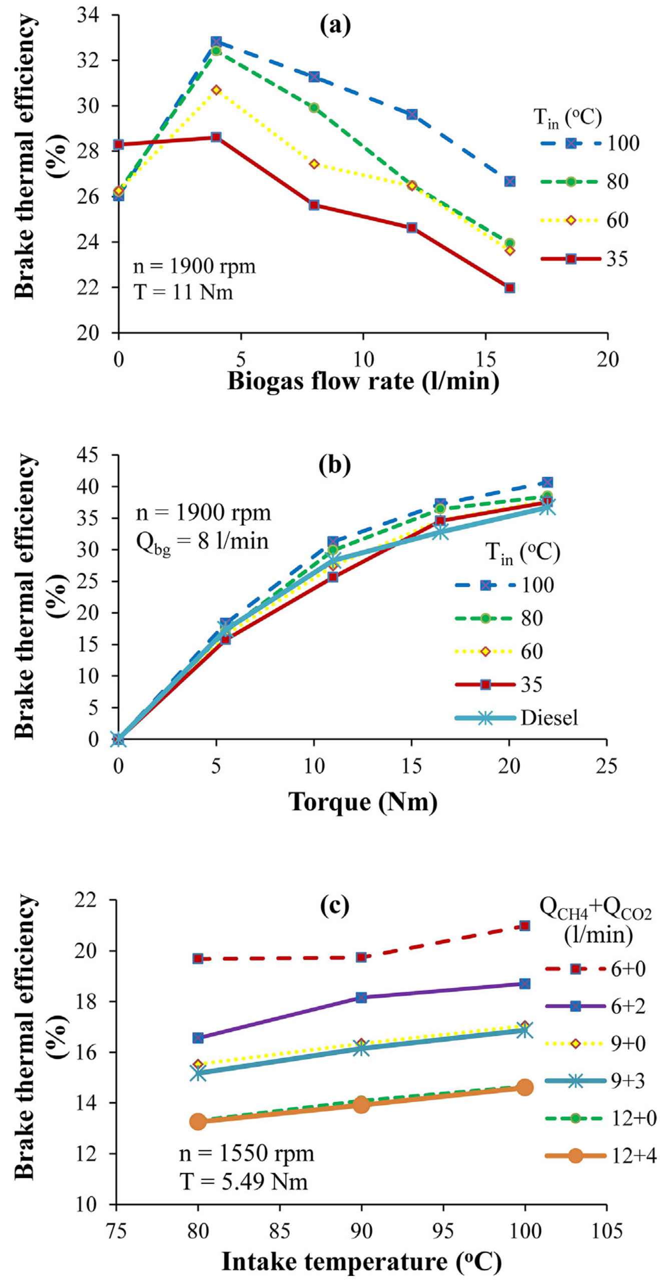

Fig. 6. Effect of (a) biogas flow rate, (b) torque and (c) intake temperature on brake thermal efficiency.

Dual fuel mode is characterized by the displacement of air. Reduction in excess air results in mixtures which are closer to stoichiometric. Further, the calorific value of methane exceeds that of diesel. This leads to greater energy release during combustion and hence higher exhaust temperatures. As expected, a rise in intake temperature produces a corresponding increase in exhaust temperature. Increased diesel fuel consumption at higher loads is responsible for hotter exhaust gases (see Fig. 7(b)).

The effect of charge preheating on exhaust gas temperature for methane-diesel and biogas-diesel modes under low speed mode is shown in Fig. 7(c). Compared to methane, exhaust gas temperatures are lower for biogas as the CO_2 absorbs part of the energy released. Temperatures in this mode are lower than those of the high speed mode.

4. Conclusions

This study has shown that careful choice of the intake parameters such as charge temperature, biogas flow rate and composition can effectively control various performance indices in a dual fuel engine. Preheating the biogas-air mixture in the intake can improve brake thermal efficiency by up to 5% at full load. However, preheating reduces volumetric efficiency by as much as 6% and increases exhaust temperature by up to almost 100 °C. Charge preheating can therefore be a viable option in conjunction with a suitable waste heat recovery system. Low biogas flow rates (e.g. 4 L/min), especially with methane enrichment, are recommended if enhancing thermal efficiency is the main objective. On the other hand, high biogas flow rates ensure significant diesel fuel savings at the expense of thermal and volumetric efficiency. Likewise, high

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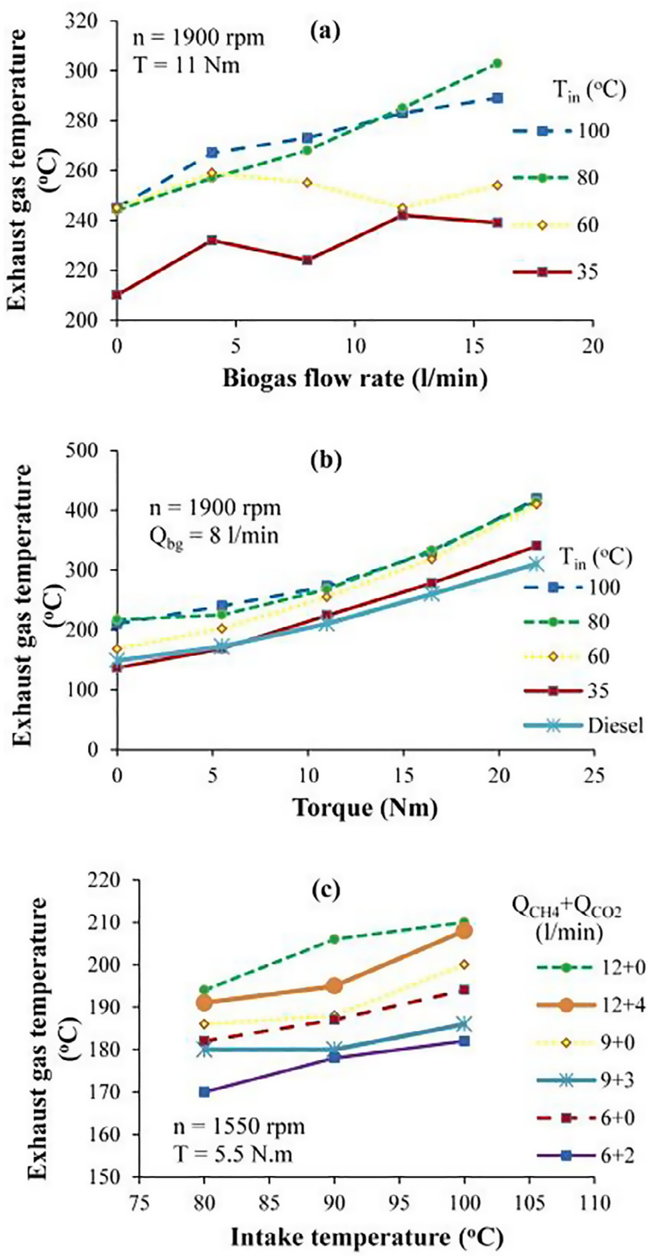

Fig. 7. Effect of (a) biogas flow rate, (b) torque and (c) intake temperature on exhaust gas temperature.

speed operation ensures higher thermal efficiency, while low speed operation provides diesel fuel substitution up to 90%.

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