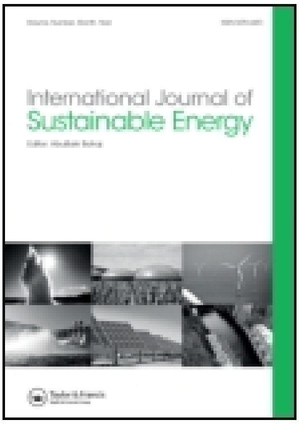

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Performance of a diesel engine in a dual fuel mode using producer gas for electricity power generation

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The paper presents the performance of a diesel engine coupled to a biomass gasifier in the dual fuel operation. The paper addresses a methodology and analysis towards choice of a diesel engine to meet the specific power requirement. Performance evaluation of both the engine and the gasification system is reported. Diesel savings in excess of 75% have been achieved. The specific fuel consumption is about 1 kg of biomass and 46 g of diesel per kilowatt hour of electricity, amounting to an overall efficiency of 22% in the dual fuel compared with 28% in the diesel engine operation. The overall energy balance indicates that the heat loss through the engine exhaust is high in the case of dual fuel mode of operation. The specific energy consumption is 17 MJ/kWh in the dual fuel mode. Engine exhaust emissions, CO and NO_x, in diesel and dual fuel modes are compared.

Keywords: biomass gasification; dual fuel; distributed power generation; producer gas

1. Introduction

One of the crucial factors that ensure and indicate to development in any region is access to electricity. Electricity is considered as an enabling intermediate for ensuring job potential and productivity that brings in economic returns to individuals and families, thus enabling sustainable living with a minimum standard. Centralized electricity generation and distribution are essential components of a modern society, mostly based on hydro resources and thermal routes. The thermal route either through the Rankine or through the Brayton cycle needs a fuel, and its capacity depends on the availability of coal in reasonable proportions and oil in some cases. There are few exceptions using biomass as a fuel in large capacity power generation using the above processes.

According to one of the theories on oil consumption, about 25% of the oil consumption is used for heat and power (Anon 2008). This amounts to about 20 million barrels per day (mbpd) of the current 84 mbpd of oil demand towards heat and power, primarily in developing countries. Based on the projections made, about 5 mbpd is consumed for power generation and therefore the rest 15 mbpd is for the heat application.

There are several isolated communities that receive very little or no electricity (Dasappa 2007). The African region, which accounts for 13% of the world population, generates about 3.1% of the world electricity. Similarly, there are regions such as the Caribbean, South America and

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Asia where diesel-based power generation is predominantly used due to various factors that do not permit central grid to be drawn. These factors further attract attention from isolated areas, such as islands in various continents, the Amazons, etc., where the primary source of energy is based on liquid fossil fuel such as diesel and heavy oil. Any saving in the fossil fuel has immense cost benefits and also causes a reduction in greenhouse gas emissions. Consider a country like Cuba which has large numbers of small diesel generators to meet both base load and peak demand and uses a large amount of diesel. The capacities of these electricity-generating systems range from 50 kW to a few megawatts, depending on the size of the population to be serviced. In some of the installations, such as those in Africa (Dasappa 2007), diesel-based power generation meets only the domestic illumination demand, while other economic activities such as agriculture, fishing, etc., are deprived of the use of electricity due to the expensive oil for power generation.

1.1. *Dual fueling of engine*

An option to save fossil fuel in the existing diesel-based power generation system is to operate on dual fuel operation. Dual fuel operation of an engine can be viewed in two broad categories, depending on the relative amount of gaseous fuel to that of the diesel liquid fuel. The first and the most common category is to inject a relatively small quantity of diesel primarily to provide ignition of the lean mixtures of the gaseous fuel and air. The bulk of the energy release comes from the gaseous fuel component. The second category is associated with the addition of some gaseous fuel to the air of a fully operational diesel engine, well beyond the light load operational range. Ideally, there is a need for optimum variation in the liquid fuel quality in relation to the gaseous fuel supply, so as to provide the best performance for a specific engine under consideration over a range of gaseous fuels (Karim 2003). In the present case, dual fuelling with producer gas, the former option, is chosen. Karim (2003) also highlights the influence of various factors on the performance in the dual fuel mode of operation using different fuels such as methane, ethylene, hydrogen and propane. Sahoo *et al.* (2009) in a review article compile the influence of various engine parameters in a dual fuel engine along with diesel as the primary fuel. Most of the studies are on using natural gas, LPG and methane as the second fuel, and very limited scientific studies are reported on producer gas (Ramadhas *et al.* 2006, Singh *et al.* 2007, Banapurmath and Tewari 2009). Selim (2004) has studied the performance of a dual engine using LPG, natural gas and methane, with focus on various parameters influencing knocking of the engine.

1.2. *Producer gas as fuel*

Producer gas is generated using biomass as a fuel in a thermo-chemical process. Gasification is a process by which solid biomass is converted into a clean gaseous form in a solid bio-residue gasifier. The process involves subjecting the solid biomass to partial combustion under sub-stoichiometric conditions in air, followed by subsequent reduction process resulting in the formation of producer gas, which is composed of H_2, CO, CH_4, CO_2 and rest N_2, with a mean calorific value of 4.7 ± 0.3 MJ/kg. The gas is then piped to an engine generator as a fuel, and electricity generated. The system comprises various elements, a reactor and a cooling and cleaning system. A water treatment plant for recycling the water used for cooling the gas and a biomass drier using an engine exhaust are the standard auxiliaries for the plant operation (Sridhar *et al.* 2005a, 2005b, Dasappa 2007). Bhattacharya (2005) highlights some of the activities in Asia on gasification. Knoef (2005) elaborates on various technology packages using the gasification system for both power and heat applications. Ghosh *et al.* (2004) highlight the performance of an islanded operation of dual fuel using a 5×100 kW system, reporting an average diesel replacement of about 59% with an overall efficiency of 19%. Murari *et al.* (2009) have addressed and tested a single-cylinder water-cooled engine on dual fuel operation with varying hydrogen contents using simulated producer gas from high-pressure cylinders containing different gaseous species. They have reported smooth engine operation in the dual fuel engine using producer gas for different fuel–air equivalence ratios and pilot injection timings. The indicated thermal efficiency is about 37% over a range of injection timings, and a small drop in efficiency was reported with retarded injection timings. Banapurmath and Tewari (2009) have carried out a study to evaluate the performance of a direct injection (DI) single-cylinder compression ignition engine operated on dual fuel mode to evaluate the feasibility of alternative fuels in the form of Honge oil and its methyl ester along with producer gas as a replacement for fossil fuels. They have found that advancing the injection timing from 19° BTDC to 27° BTDC (Before Top Dead Centre), the engine performance improved moderately; they have presented results for 27° BTDC. Maximum diesel replacement of about 70% is reported in the dual fuel operation with diesel and producer gas. They have carried out engine exhaust measurements for different fuel combinations, but have not normalized the results for any meaningful comparison; for example, CO is mentioned in percentage, without oxygen concentration in the exhaust. Ramadhas *et al.* (2006) have presented results from a gasifier coupled to a single-cylinder diesel engine of 5.5 kW. The specific energy consumption (SEC) reported by them for both wood chips and coir pith as fuel is 18 MJ/kWh, while the diesel SEC is about 15 MJ/kWh. They have reported a maximum of 72% diesel substitution at 50% of the rated load. Singh *et al.* (2007) have also reported test results on dual fuel with producer gas and diesel on a three-cylinder engine and an SEC of 11.9 MJ/kWh with a diesel replacement of 22%. The SEC is low due to low diesel replacement, implying operating close to normal diesel.

1.3. *Producer gas as a fuel for engines*

The right choice of the engine becomes an important criterion to meet the specifications of a power plant, i.e. to deliver the power at the rated capacity and the diesel substitution. In order to deliver the designed electrical output, it is important to consider the engine volumetric cylinder capacity to accept the low energy content gas and to deliver the designed output. Further, it is vital to recognize that there is a de-rating on the output of the engine and a fall in the overall efficiency of the engine at high diesel replacement without modifying the engine operating conditions such as injection timings (Baliga *et al.* 1993, Galal *et al.* 2002, Ghosh *et al.* 2004, Ravindranath *et al.* 2004, Murari *et al.* 2009). These are related to combustion processes that occur in the engine cylinder. Detailed investigations into the use of producer gas for gas engines and its implications have been carried out by Dasappa (2001) and Sridhar *et al.* (2005a, 2005b). Dasappa (2001) has analysed and identified the parameters that influence the performance of a gas engine using producer gas with respect to the power output of an engine. These parameters are related to the net energy content in the gas–air mixture, which is lower by about 15%, effect of change in the number of moles between product and reactant to a tune of 10 and about 8% due to the lower adiabatic temperature compared with fossil fuel which influence the peak pressures achieved inside the engine cylinder during combustion. Similar effect on the performance is expected in the dual fuel operation, except that the peak power can still be obtained by increasing the diesel fraction in the dual fuel mode. It is important to highlight that the properties of the producer gas are different compared with those of the other fuels. Table 1 provides the properties of various fuels that influence the engine operation. It is clear that producer gas has different stoichiometric ratios, flame speeds, adiabatic flame temperatures and, more importantly, the reactant-to-product mole ratios. These parameters will have an influence on the overall performance of the engine.

Table 1. Properties of various fuels.

Fuel + air	Fuel LCV (MJ/kg)	Air/fuel at $\Phi = 1$	Mixture (MJ/kg)	Φ, Limit Lean	Φ, Limit Rich	S_L (limit) (cm/s) Lean	S_L (limit) (cm/s) Rich	S_L, $\Phi = 1$ (m/s)	Peak flame temperature (K)	Product/reactant mole ratio
H_2	121	34.4	3.41	0.01	7.17	65	75	0.270	2400	0.67
CO	10.2	2.46	2.92	0.34	6.80	12	23	0.045	2400	0.67
CH_4	50.2	17.2	2.76	0.54	1.69	2.5	14	0.035	2210	1.00
Producer gas	5.00	1.35	2.12	0.47	1.60	10.3	12	0.050	1800	0.87

Galal *et al.* (2002) have also carried out tests using natural gas and diesel as fuels in a dual fuel engine in which the rated output is reduced by about 15% in the dual fuel mode compared with the normal diesel operation. Introduction of gaseous fuel for dual fuel operation has an influence on heat release rates within the engine cylinder, thus influencing the performance (Karim 2003). In the case of dual fuel operation, the overall efficiency, which is defined as the ratio of output power to energy input from the combination of fuels, is in the range of 26–28% (Galal *et al.* 2002) using natural gas as the second fuel, whereas in the case of producer gas, the overall efficiency has been found to be about 20% (Ghosh *et al.* 2004). Murari *et al.* (2009) also report a loss in the power output in the dual fuel operation, and they address this issue by using superchargers. It must be mentioned that unlike natural gas, which is a prepared fuel, the present work addresses generating the gaseous fuel from a solid fuel whose conversion efficiencies are to be accounted for. Thus, for producer gas, it is imperative to consider the input energy to be the sum of energy contents of both biomass and diesel. The cold gas efficiency, which is defined as the ratio of the energy content in the gas to the energy content in biomass, is in the range of 75–78% and has to be factored in the overall efficiency. Based on a detailed study using producer gas in small engines (Baliga *et al.* 1993, Ravindranath *et al.* 2004), the average conversion efficiency from gas to electricity is about 20% in the case of a small capacity naturally aspirated engine.

The paper presents the development of a technology package for electricity generation using biomass gasification and reports results from the performance evaluation of a diesel engine using producer gas generated from the thermo-chemical conversion of biomass as fuel in the dual fuel mode. The present work helps in retrofitting existing diesel engines to substitute fossil fuel, and no modification on the engine is attempted here.

Section 2 presents the design formulation for the technology package and the measurement scheme. In Section 3, the performance evaluation of the diesel engine for establishing the baseline scenario using fossil fuels is presented and the results from the experiments on evaluating the performance of the gasification system in the dual fuel mode are reported. Section 4 is devoted for results and discussion, and the conclusions are drawn in Section 5.

2. Materials and methods

2.1. *The system configuration*

The technology package consists of a gasification system and a diesel engine adapted to operate in the dual fuel mode. Figure 1 shows the schematic diagram of the power generation system configuration. The gasification system was configured to operate on different biomasses as fuels. The other elements of the package are a water treatment plant for closed-loop water recirculation system, wood chip drying using engine exhaust and a fuel preparation system. All the system

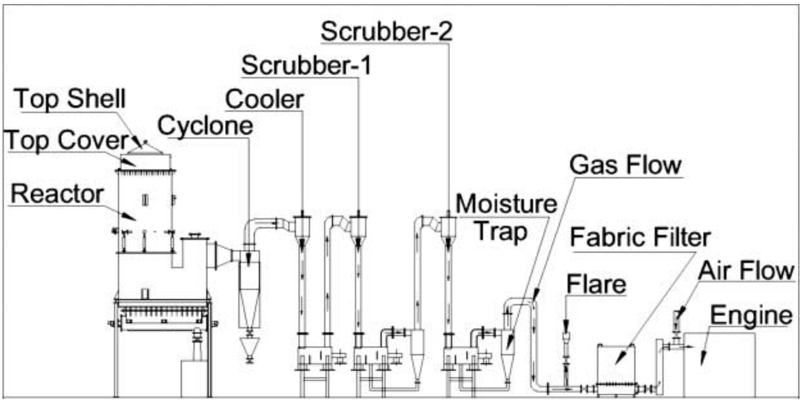

Figure 1. Schematic of the gasification system.

elements were designed to operate on three-phase 220 V and 60 Hz. Details about the gasification technology package are described in Dasappa *et al.* (2004, 2007).

2.2. *Design of the dual fuel power package*

The design requisite is to generate about 50 kW electrical output in the dual fuel mode using producer gas (secondary fuel) along with diesel (primary fuel). The overall efficiency is defined as the ratio of electrical energy output to the energy input in the form of diesel and/or biomass. With an overall conversion efficiency of 20% (using major energy input from biomass-derived fuel along with diesel to electricity), the total energy input from the producer gas and diesel is estimated to be 250 kW. The energy content for the producer gas and air mixture under stoichiometric conditions is in the range of 2.2 ± 0.1 MJ/kg. At 75% diesel substitution, the overall biomass consumption to deliver an electricity output of about 50 kW is estimated at 55 and 3 kg/h of diesel. Using these derived data, the energy and mass balance, as discussed in Section 4, suggests a total cylinder air intake capacity of 325–350 Nm^3/h to deliver about 50 kWe of power output in the dual fuel mode. Thus, the specification of the power generation system using dual fuel comprises a gasification system of 75 kg/h gasifier coupled to a diesel engine whose total air flow rate is about 350 m^3/h to deliver 50 kW electricity.

2.3. *The engine specification*

Arriving from the need to deliver about 50 kW in the dual fuel mode, a naturally aspirated, six cylinders, inline, DI, Isuzu engine of 6.494 l (bore 105 mm and stroke 125 mm) with displacement capacity at 1800 rpm was chosen. The engine is rated at 68.4 kW under standard conditions (298 K and 100 kPa) for a continuous rating, with a compression ratio of 17.5:1. The engine is coupled to a 75 kVA alternator to generate 190–240 V electricity at 60 Hz.

The engine was adapted to operate in the dual fuel mode by modifying the intake manifold to accommodate producer gas flow along with the air flow, thus ensuring that the air–gas mixture enters the engine cylinders through the manifold. The gas from the gasifier after cooling and cleaning enters the engine intake manifold at the venturi-based mixing device. A standard venturi designed to draw both air and gas also helps in the mixing. Actuators linked to the diesel governor and the valves were used in regulating the gas and air mixture. The exhaust line was connected to a biomass drier with a facility to dilute the flue gas to temperatures below 373 K. The entire

Table 2. Specifications of the dual fuel power plant.

Engine	
Engine model	Isuzu engine
Bore × stroke (mm)	105 × 125
Number of cylinders	6, inline, direct injection
Displacement (l)	6.494
Revolutions per minute	1800
Compression ratio (CR)	17.5:1
Rated output (kW)	68.4
Aspiration	Natural
Alternator (kVA)	75
Gasifier system	
Biomass consumption capacity (kg/h)	75
Gas flow rate (m^3/h)	200
Reactor	Open top-down draft
Cooling and cleaning system	Direct water scrubbers and dehumidifier along with fabric filter
Cold gas efficiency (%)	~80
Tar and particulates (mg/m^3)	Less than 10

generator set is housed in an acoustic enclosure with sound levels less than 65 dB, as per the standards.

2.4. *Measurement scheme*

The test conditions were as follows: ambient pressure was 0.92 kPa; density of air was 1.1 kg/m^3 and the ambient temperature during the testing period was 308 ± 3 K. A load bank was connected to test the engine generator set. Measurements on current, voltage, frequency, fuel consumption and exhaust gas analysis were carried out. Temperatures at various locations on the gasifier and engine were measured using a K-type thermocouple. Static pressures were monitored using water tube manometers.

An online gas analyser from SICK Maihak, Germany was used to measure the mole fractions of CO, H_2, CH_4, CO_2 and O_2 in the producer gas. Sample evaluation of the gas for calorific value was carried out using Junker's calorimeter. The diesel consumption was measured at varying load conditions by increasing and decreasing the load. Diesel flow rate was measured using the topping-up method for continuous operation, while instantaneous measurements were made using a separate offline flowmeter. Producer gas and airflow rates were measured using a venturi meter. The exhaust gas was analysed using a Quintox analyser. Except for oxygen measurement, which was a chemical cell, the other sensors used were non-dispersive infrared type.

The specifications of the power plant are presented in Table 2. The power plant was designed to deliver 50 kWe in the dual fuel mode and to achieve about 75% diesel replacement. The term 'diesel replacement' refers to the amount of diesel being replaced by the gas in comparison to diesel consumption in the diesel-alone mode of operation at any given load.

3. Performance testing of the engine in diesel and duel fuel modes

The diesel engine was tested separately for the overall performance to establish the baseline scenario, and similarly, the gasifier system coupled to the engine was also evaluated.

3.1. *Diesel mode*

Figure 2 provides data on the specific fuel consumption (SFC), oxygen concentration in the engine exhaust and the engine efficiency at various loads. It is clear that the SFC is about 0.260 kg/kWh around 50 kW. In the load range of 30–65 kWe, the SFC is 0.255 ± 0.15 kg/kWh. The peak engine efficiency, i.e. fuel to electricity, is about 33% at 48 kWe. The peak load achieved was about 68.4 kWe, amounting to about 74 kW on the engine shaft, accounting for an alternator efficiency of 92%. This load of 74 kW is about 8% higher than the JS DOOO6 rating of the engine set at 68.4 kW for the engine output (Denyo 2006), which is consistent with the 10% overrating facility. It is apparent from the data that under this load condition, the SFC has increased and also the oxygen content in the flue gas has reduced to 2.25%, indicating the inability to accept further fuel. The output per cylinder of about 12 kWe translates to 11 kWe/l of the cylinder volume, which is consistent with the data presented by Heywood (1988).

3.2. *Gasification system performance*

The gasification system was initially evaluated by measuring various parameters such as pressure across various elements, gas flow rates, gas composition and biomass consumption rate along with tar and particulates in the producer gas. The pressure drop across the open top-down draft reactor was in the range of 500–1500 Pa, depending on the flow rate. The scrubbers are ejector-based design, which helps in gaining static pressure; hence, the gas delivered to the engine was at positve pressure, which is in the range of 500–1000 Pa. The cyclone pressure drop was 100 Pa. Under these operating conditions, there was no need for a blower to supply the gas to the flare or to the engine. The system was tested up to 75 kg/h, and gas was flared in a specially designed burner.

Biomass used for the test was *Casuarina* (*Casuarina equisetifolia*), and the properties of the fuels used are presented in Tables 3 and 4. The gas compositions measured using an online gas analyser under operating conditions were found to be $CO = 19 \pm 2\%$, $H_2 = 19 \pm 2\%$, $CH_4 = 1.5 \pm 0.5\%$, $CO_2 = 12 \pm 2\%$ and the rest N_2. The calorific value is in the range of 4.6 ± 0.1 MJ/kg. Tar and particulates were measured using the wet method (Mukunda *et al.* 1994) using anisole as the solvent. Test results indicated that the tar and particulates in the gas

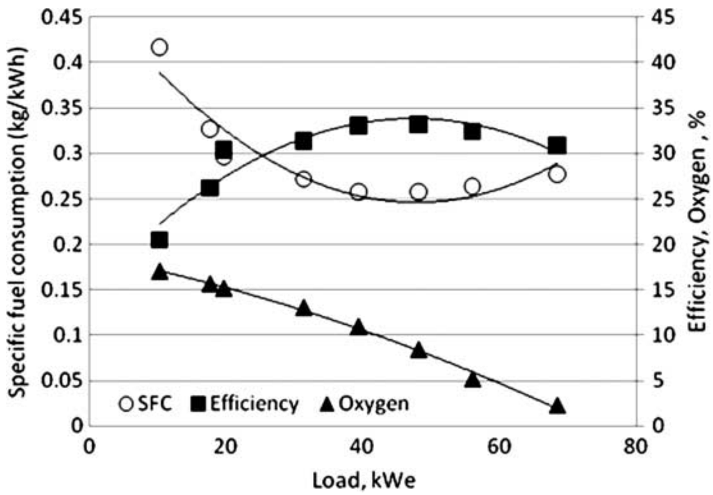

Figure 2. Performance of the engine with diesel alone as fuel.

Table 3. Properties of biomass used.

Ultimate analysis % dry basis	
C	42.83
H	6.236
N	0.124
O	50.39
Dimension of biomass used	~35 × 35 × 35 mm
Density (kg/m^3)	580 ± 20
Bulk density (kg/m^3)	410 ± 20
Volatile matter (% dry basis)	81.28
Fixed carbon (% dry basis)	18.38
Moisture content (%)	10
Ash content (%)	<1

Table 4. Properties of diesel used.

Properties	Tests	Values
Density at 15°C (kg/m^3)	P:16	820–845
Water content (mg/kg)	ISO 12937	Max. 200
Flash point (Abel) (°C)	P:20	Min. 35
Viscosity (kinematic) at 40°C (cSt)	P:25	Min. 2.0, max. 4.5
Total sulphur (mg/kg)	ASTM D:4294	Max. 350
Acidity (total) (mg KOH/g)	P:2	To report
Acidity (inorganic) (mg KOH/g)	P:2	Nil
Ash (%wt)	P:4	Max. 0.01
Cetane index	ASTM D 4737	Min. 46

were found to be 10 ± 2 and 18 ± 4 mg/m^3, respectively. Measurements on the particle size distribution indicate that over 90% of them are below 1 μm. Based on the experience in operating engines over several thousand hours (Sridhar *et al.* 2005a, 2005b, Dasappa *et al.* 2007), the amount of tar and particulates in the gas was found to be acceptable for operating a naturally aspirated engine.

3.3. *Dual fuel operation*

During the typical run, pressure drops across various system elements in the gasification system were recorded, and the data were in the acceptable range. The pressure drop across the reactor was less than 1000 Pa, and the exit temperature was 700 K. Dual fuelling was achieved by allowing the producer gas to enter the engine along with air at intake manifold. Figure 3 provides the nominal operating parameters for the gasification system. In a typical operation for about 10 h, various parameters are monitored. The normalized pressure drop $\Delta p/(1/2\rho\ V^2)$ across different elements of the gasification system is plotted against time at various loads. Δp is the pressure drop across each element, as indicated in Figure 1, and V is the superficial velocity at varying loads. The normalized pressure drop across the reactor gradually reduces, with a small increase in the gas flow rate. This is due to the bed parameters settling down on long duration operation. The pressure drop across the cyclone is nearly constant. It is important to recognize that the pressure data for other elements are positive, indicating an increase in pressure across all these elements. As indicated earlier, the cooler and scrubbers are designed as ejectors, which assist in drawing the gas. With an increase in the gas flow rate, the gain in pressure is lower due to increased fluid friction. There is a steep change in the pressure reading on the time axis, near 200 min, which was

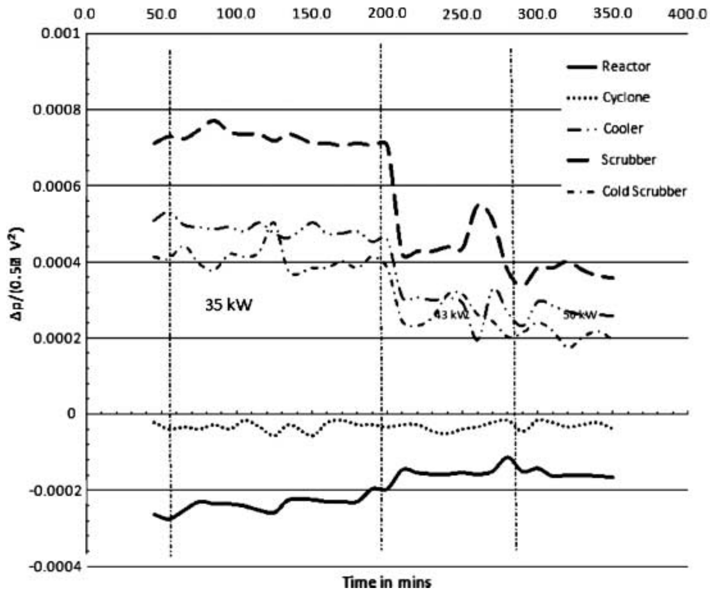

Figure 3. Normalized pressure drop across various elements in the gasifier system.

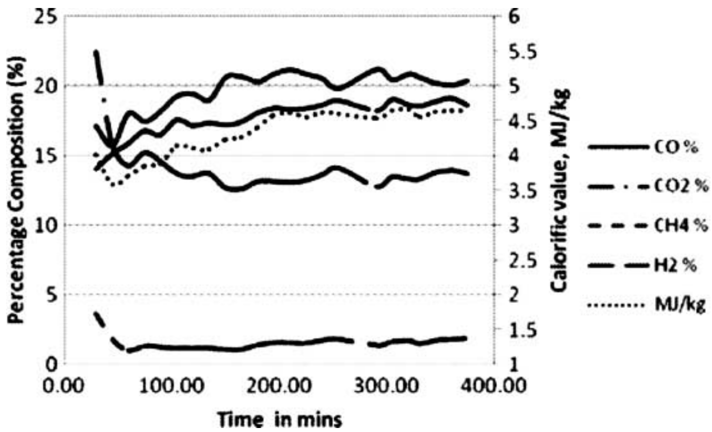

Figure 4. Gas composition and calorific value during a typical dual fuel operation.

due to a change in the gas flow rate and also the ash extraction carried out at this point. These changes have no influence on the performance of the system.

Figure 4 provides the gas composition and its calorific value based on the composition during one of the test runs. It is clear that the calorific value is consistently above 4.5 MJ/kg, except during the startup. The cold gasification efficiency, defined as the energy content in the cooled gas to that of the energy as solid fuel, is in the range of 75–80%.

Table 5 provides the details on the dual fuel operation of the engine. The engine operating parameters such as the injection timing and injection pressures were maintained the same as those of the standard diesel engine. The engine was gradually loaded and allowed to stabilize. Measurements were made on the diesel consumption at various loads. The maximum amount of producer gas (in other words, the lower limit of diesel fuel amount) at each of the engine load was decided based on the engine response with respect to misfire, if any, and the oxygen measurement

Table 5. Diesel replacement at various loads.

Diesel mode		Dual fuel mode			
Load (kWe)	Fuel consumption (l/h)	Load (kWe)	Fuel consumption (l/h)	Fuel replaced (l/h)	Diesel replacement (%)
8.27	5.0	8.3	1.21	3.8	75.9
15.9	6.9	21.2	1.24	6.2	89.7
20.4	7.4	22.9	1.46	5.9	80.3
25.0	7.8	25.6	1.01	6.8	87.2
29.2	9.9	28.8	1.14	8.8	88.5
36.5	11.1	36.2	1.66	9.4	85.1
40.2	12.7	41.5	2.09	10.6	83.6
44.9	14.2	44.1	2.52	11.6	78.3
48.3	15.2	49.9	3.13	12.0	79.3
53.6	17.3	52.4	4.0	13.3	77.0
57.3	18.3	57.4	3.89	14.4	78.7

in the exhaust. At low loads and very high diesel replacement, i.e. in excess of 85%, misfire in the exhaust was obvious, thus suggesting the limiting condition for diesel substitution by producer gas. At high loads, the limiting condition was evident based on the oxygen content in the engine exhaust, and the engine exhaust was found to be both sooty and smoky. Further, loading the engine at this condition reduced the capability of the engine to maintain the speed and affected the frequency of the electricity generated.

4. Results and discussion

4.1. *Mass and energy balance*

In order to explain the experimental results obtained and also to establish the limiting conditions for the performance, a simple mass and energy balance is carried out. It is assumed that the suction efficiency or volumetric efficiency is in excess of 90%, which is true in a well-designed intake manifold.

The primary objective of dual fuelling an engine is to substitute diesel by an alternate fuel, in this case producer gas, thus reducing the fossil fuel consumption. The methodology adapted allows the mixture of air and gas into the engine cylinder and allows the diesel governor to take control on the speed. With an increase in the gas flow rate, the diesel flow rate reduces to maintain the speed of the engine. The relative flow rates of air and gas depend on the extent of diesel substitution aimed at the smooth operation of the engine without any knocking.

In order to establish the necessary relationship between various parameters related to the performance, a simple mass and energy balance across the engine cylinders is considered: actual volumetric flow rate of air, $\dot{m}$, in kg/h = density of air × (cylinder volume × number of cylinders × rpm)/(number of strokes × 60) × η_s, where η_s is the suction efficiency.

In the case of dual fuel mode of operation, the total flow rate is $\dot{m} = \dot{m}_a + \dot{m}_g$, where $\dot{m}_a$ and $\dot{m}_g$ are the mass flow rates of air and gas in the dual fuel mode, respectively. $\dot{m}_a$ is the amount of air available for combustion of diesel and gas in the dual fuel operation. The mass flow rate of gas is obtained from the energy balance by estimating the amount of diesel to be replaced. As the primary objective of dual fuelling is to replace diesel with producer gas, the amount of diesel replaced is to be substituted by an equivalent energy from the producer gas, accounting for the inefficiencies in the overall conversion in the dual fuel mode of operation. If $\dot{m}_d$ is the diesel flow rate in the diesel mode at the rated condition and for 75% diesel substitution, the energy $(0.75 \times \dot{m}_\mathrm{d} \times H_\mathrm{d})$ has to be substituted by the producer gas.

Balancing the two energy flows, the gas flow rate $\dot{m}_\mathrm{g} = (0.75\dot{m}_\mathrm{d}H_\mathrm{d}/h_\mathrm{g})(\eta_\mathrm{d}/\eta_\mathrm{df})$, where H_d and h_g are the heats of combustion of diesel and gas and η_d and η_df are the efficiencies in the diesel and dual fuel modes, respectively.

The amount of air required for diesel combustion is $\dot{m}_\mathrm{a1} = 0.25\dot{m}_\mathrm{d} \times (A/F)_\mathrm{s} = 0.25\dot{m}_\mathrm{d} \times 15$. Similarly, the amount of air required for combustion of $\dot{m}_g$ quantity of producer gas is $\dot{m}_\mathrm{a2} = \dot{m}_\mathrm{g} \times (A/F)_\mathrm{s} = \dot{m}_g \times 1.3$, where $(A/F)_\mathrm{s}$ refers to the stoichiometric requirements for both the fuels independently. The minimum amount of air required for both the fuels to burn is $\dot{m}_\mathrm{min} = \dot{m}_\mathrm{a1} + \dot{m}_\mathrm{a2}$. The difference between $\dot{m}_\mathrm{min}$ and $\dot{m}$, the total air flow into the cylinder, will provide the excess air available in the engine cylinders for combustion.

4.2. *Diesel operation analysis*

As indicated in Figure 2, measurements on the oxygen concentration in the engine exhaust reduce from nearly 17% to about 2.25% with an increase in the load, suggesting the shift from a lean combustion process to a rich condition at higher loads. From the mass balance analysis presented earlier, it is clear that from the values of diesel flow rate, the air flow rate and oxygen content in the exhaust suggest the amount of diesel that can be injected and thus the load. At 68.4 kW, the diesel consumption rate is 18.5 kg/h. Under stoichiometric conditions, the minimum air required for combustion is about 280 kg/h, whereas the actual A/F is 18.2 for diesel operation. At this load, from the mass balance, the excess oxygen is 3.3%, compared with the measured data of 2.3%. This further strengthens the argument on the peak load capability of this engine and the limiting condition. The peak load achieved also indicates the limiting condition on the engine as the oxidizer is nearly fully utilized during combustion with no oxygen left for additional fuel combustion. The peak conversion efficiency of 33% is in the load range of 55 kWe, with oxygen content in the exhaust at 5%.

The energy balance at the peak load condition indicates that the sensible heat equivalent of about 61 kW lost in the exhaust and balance 80 kW is lost to the water used for engine cooling. With the engine efficiency at 33%, the exhaust and the engine cooling account for 29% and 38%, respectively. A fan within the acoustic enclosure appears to have drawn some heat from the exhaust piping, resulting in slightly lower energy in the flue, suggesting that the radiator-based engine cooling load may be slightly underestimated as there is no direct access to measure this quantity.

4.3. *Dual fuel operation analysis*

Table 5 also provides the comparison of diesel and dual fuel modes of operation. At a peak load of 57 kWe, the diesel consumed in diesel and dual fuel modes of operation is 18.3 and 3.9 l/h, respectively, with a saving of 14.4 l/h. This saving translates to about 79% over the diesel operation. Over the entire load range tested on this engine, the diesel savings have been in excess of 75%. The output per cylinder in the dual fuel mode is 9.5 and 8.8 kWe/l of the cylinder volume at diesel replacement of 78%. This is lower by about 20%, compared with the diesel operation due to lower mixer density. The specific diesel consumption is 0.25 kg/kWh in the diesel mode, whereas it is 0.045 kg/kWh in the duel fuel mode.

Figure 5 shows the comparison of SEC in the diesel and dual fuel modes. In the case of dual fuel operation, the SEC is the sum of the energy content from both the fuels, i.e. biomass and diesel to generate a kilowatt hour of electricity and only diesel otherwise. It is clear that in the entire range of loads, the energy consumption in the dual fuel mode has been higher than that in the diesel mode. For the diesel and dual fuel operations, the SEC is nearly constant beyond an electrical

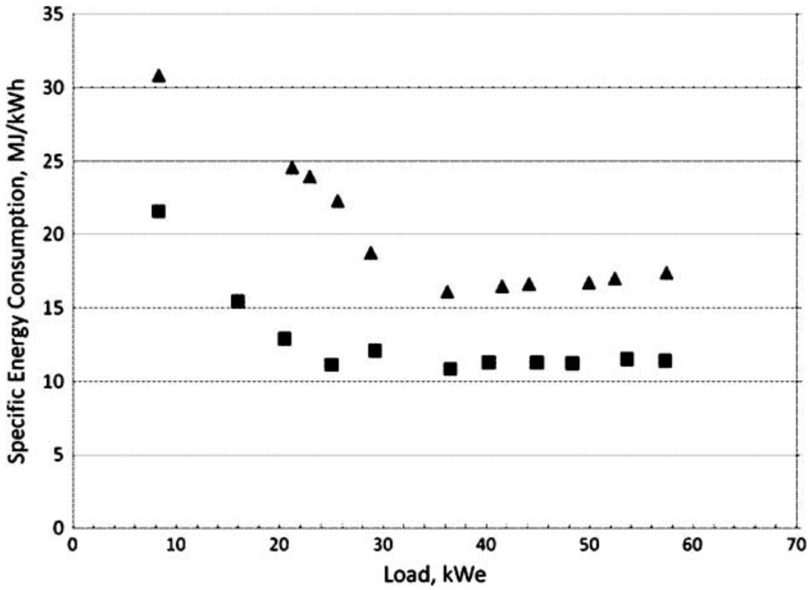

Figure 5. SEC in diesel and dual fuel modes.

load of 30 kWe, with 11 MJ/kWh in the case of diesel mode of operation and 16 MJ/kWh for dual fuel. This value also suggests that the overall efficiency in the dual fuel operation is lower compared with the diesel operation.

Figure 6 compares the present diesel replacement with the normalized maximum achieved load with that of Ghosh *et al.* (2004). It is clear that the present operations have shown a consistent diesel replacement/savings in excess of 75% over the load range considered, while Ghosh *et al.* have achieved an average of 59% at 45% of the rated conditions. The best diesel replacement achieved was 64%. The SEC is 17 MJ/kWh in the dual fuel mode. One of the major reasons for low diesel substitution is due to poor gas quality, meaning the energy content in the producer gas being low, reflecting on the gasifier performance.

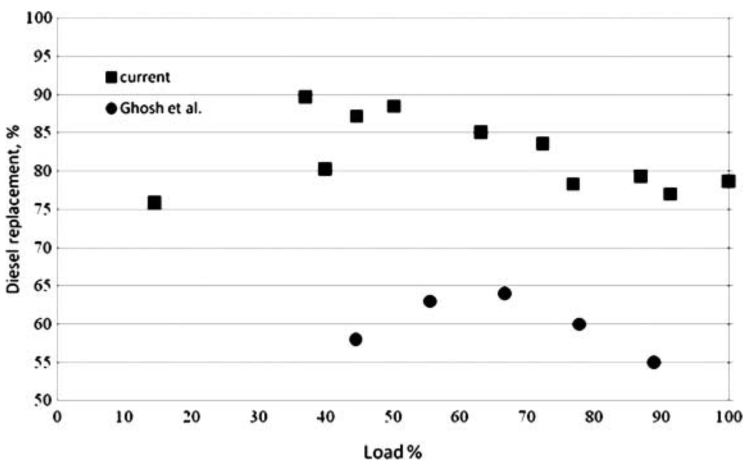

Figure 6. Comparison of diesel replacement with load.

Long duration operations were carried out to establish the biomass consumption rate at different loads. Table 6 highlights the details of the biomass consumption rate and diesel consumption at various loads. The specific biomass consumption at the rated condition is less than 1 kg/kWh, whereas at low loads, it is about 1.5 kg/kWh. Table 6 also presents the data on the efficiency of conversion also an indication of SEC. The overall efficiency is about 22% at 36 kWe. The engine efficiency, defined as energy in the producer gas plus diesel to electricity, is 29% at the same load condition. The measured efficiency is lower by about four points, compared with the diesel operation. This is consistent with the observation by Galal *et al.* (2002), in which they observed a 3% reduction in efficiency in the dual fuel mode (compressed natural gas plus diesel) compared with the diesel mode.

Table 7 analyses the overall energy balance in the diesel and dual fuel modes at peak efficiency. At 43 kWe, the engine exhaust temperature in the dual fuel mode is about 80 K higher than that in the diesel mode. This is consistent with the higher exhaust temperature by about 100 K observed by Galal *et al.* (2002) in dual fuel operation with natural gas. This increase is due to the nature of the combustion process inside the engine cylinder, which manifest itself as increase in the exhaust temperature and also the cooling load on the engine, namely the radiator load. Sahoo *et al.* (2009) report an increase in the exhaust temperature in dual fuel operation and indicate a slight reduction in the exhaust temperture with advancing the injection timing.

Based on the temperature measurements, the difference in the energy outflow in the exhaust gases between the dual fuel mode and the diesel mode is about 17.5 kW. The difference in the input energy in the two modes of operation is about 32 kW, reflecting the inefficiencies in the dual fuel operation. Thus, a net 14 kW was unaccounted. Based on the exhaust gas composition, the energy carried away by unburnt CO is about 6 kW. Arising from the energy balance 8 kW is unaccounted, amounting to 5% of the energy input.

There is an increase in the energy loss in the dual fuel mode through exhaust as well as the cooling water compared with the diesel operation. Karim (2003) highlights that dual fuel operation

Table 6. Overall performance of the engine on dual fuel mode.

Load (kWe)	Fuel consumption (l/h)	Biomass consumption (kg/h)	Specific biomass consumption (kg/kWh)	Overall efficiency from (diesel + biomass) to electricity (%)	Efficiency gas to electricity (%)
8.3	1.21	14.2	1.70	11.7	15.2
21.2	1.24	31.8	1.50	14.6	19.0
23.0	1.46	33.3	1.45	15.0	19.5
25.8	1.01	36	1.40	16.1	20.9
28.6	1.14	33.1	1.15	19.3	25.0
36.2	1.66	35.0	0.97	22.3	28.9
41.5	2.09	40.7	0.98	21.8	28.3
44.1	2.52	43.0	0.97	21.6	28.1
50.0	3.13	48.4	0.97	21.4	27.8
52.4	4.0	50.0	0.95	21.1	27.4
57.4	3.89	57.4	1.00	20.7	26.8

Table 7. Energy balance in the dual fuel mode.

Parameter	Dual fuel mode	Diesel mode	Difference (kW)
Exhaust temperature (K)	730	650	
Electrical output (kWe)	43	43	
Engine output (kW)	46.7	46.7	
Exhaust (kW)	54.7	37.6	17.5
Energy input (kW)	162.3	130.4	32.1

changes the heat release rate pattern, and it also has an influence on the combustion processes. As stated earlier, with optimal injection timing, these discrepancies can be controlled and are not being attempted here to maintain the same diesel engine operating conditions. Karim (2003) provides a detailed analysis on the influence of injection timing on the engine performance. Further, increased amount of heat loss to the cooling water in gas operations is attributed to the engine combustion chamber design. Heywood (1988) indicates that engine geometries such as bowl-in-piston would experience 10% higher heat transfer. In the present case, the heat transfer from the combustion chamber to the coolant water falls well within this range (7–10%). For a compression ignition engine with liquid fuel, combustion is heterogeneous and essentially occurs at multiple ignition sites in a diffusion mode.

4.4. *Emissions*

Table 8 compares the exhaust emissions in the diesel and dual fuel modes. The CO emission measured in engine exhaust gases is 1 g/kWh in the diesel mode and 46 g/kWh in the dual fuel mode at a load of 40 kWe. The NO_x emission measured is 3.1 g/kWh in the diesel mode and 0.8 g/kWh in the dual fuel mode at a load of 40 kWe. Typical standards used in various countries for both CO and NO_x are less than 5 and 9 g/kWh, respectively, for diesel stationary applications (Anon 2006). Further, the guideline for emissions from biomass gasification plants for gas engine applications (Anon 2009) suggests some of the standards used in Denmark and Germany. Denmark and Germany use 3000 and 650 mg/m^3 for the CO emission and 550 and 500 mg/m^3 with 5% oxygen in the exhaust. The measured values of CO at 5% oxygen in the exhaust are 154 and 8454 mg/m^3 for diesel and dual fuel modes, clearly suggesting that the CO levels are higher. Similarly, the NO_x levels are 446 and 165 mg/m^3 for diesel and dual fuel modes, respectively. In the case of dual fuel mode, the CO level is higher compared with the diesel operation, reflecting on the combustion process with both diesel and gas as fuels. The emission level is severe at low loads and high diesel substitution, indicating that the combustion properties related to mixing and flame propagation pose the limit. Murari *et al.* (2009) have addressed the effect of ignition timing on the exhaust emissions in single-cylinder dual fuel operation with simulated producer gas. The variation in ignition timing has been from nearly TDC to 24° BTDC, and they have found that in the range of 10° to 12° BTDC, there exists a valley for both CO and HC and a slightly higher NO_x, an increase from 200 to 400 ppm. It is evident from the compilation of results from the literature by Sahoo *et al.* (2009) that the injection timing has an effect on emissions. Further, the work by Selim (2004) on LPG, natural gas and methane suggests that some of the fuel properties along with injection timing have an influence on emissions. Experiments with varying injection timings are to be carried out for the possible reduction of these emissions along with improvement in performance for producer gas dual fuel operation as a part of future work.

Table 8. Comparison of emission in diesel and dual fuel modes.

Diesel mode			Dual fuel mode		
Load (kWe)	CO (g/kWh)	NO_x (g/kWh)	Load (kWe)	CO (g/kWh)	NO_x (g/kWh)
6	10.50	7.31	8.3	1192.4	0.86
10	6.66	4.86	21.4	243.8	0.25
21	2.66	3.49	23.5	135.1	0.22
30	1.62	3.35	28.9	56.3	0.18
43	0.97	3.13	40.6	46.5	0.79
53	2.41	3.06	50.0	38.5	1.19
60	14.00	2.63			

At present, it is recommended to operate the system in the dual fuel mode at higher loads. It is important to address the necessary requirement of a catalytic converter for meeting the exhaust emission norms. In the dual fuel operation, with gas accounting for 30–40% of the total flow rate, the NO_x level is lower than the diesel operation due to lower combustion temperature in the combustion chambers, reducing both fuel and thermal NO_x.

The challenge has been in establishing the technology package using biomass as a fuel towards generating gaseous fuel of desired quality that can be piped to an internal combustion engine to replace diesel to a significant extent. Biomass being CO_2 neutral, it may have an important role to play in the mitigation options.

5. Conclusions

The paper provides a criterion for designing distributed power generation systems using producer gas as fuel in a diesel engine in the dual fuel mode. The cold gas efficiency of the gasification system is about 80%, with an average calorific value of producer gas being 4.5 MJ/kg. Diesel savings in excess of 75% have been shown to be possible on a commercial diesel engine operating in the dual fuel mode. The SFC is found to be about 1 kg/kWh of biomass along with about 55 ml/kWh of diesel. The technology package highlights the feasibility of decentralized power generation along with environmental benefits. Reduction in the fossil fuel results in a net decrease in CO_2 emission using biomass as fuel.

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