Adaptation of small capacity natural gas engine for producer gas operation

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The manuscript was received on 22 June 2011 and was accepted after revision for publication on 6 September 2011.

DOI: 10.1177/0954406211424678

Abstract: This article addresses the adaptation of a low-power natural gas engine for using producer gas as a fuel. The 5.9 L natural gas engine with a compression ratio of 10.5:1, rated at 55 kW shaft power, delivered 30 kW using producer gas as fuel in the naturally aspirated mode. Optimal ignition timing for peak power was found to be 20° before top dead centre. Air-to-fuel ratio (A/F) was found to be 1.2 ± 0.1 over a range of loads. Critical evaluation of the energy flows in the engine resulted in identifying losses and optimizing the engine cooling. The specific fuel consumption was found to be 1.2 ± 0.1 kg of biomass per kilowatt hour. A reduction of 40 per cent in brake mean effective pressure was observed compared with natural gas operation. Governor response to load variations has been studied with respect to frequency recovery time. The study also attempts to adopt a turbocharger for higher power output. Preliminary results suggest a possibility of about 30 per cent increase in the output.

Keywords: producer gas, gas engines, gasifier, natural gas

1 INTRODUCTION

Increasing availability of gaseous fuel and demand for its use in power generation has led to the proliferation of gas engines. Initially, gas engines were adapted from a diesel engine frame using an appropriate ignition system, a gas governor, and a carburetion system along with suitable adjustment in compression ratio (CR) depending upon the type of gaseous fuel.

Researchers have been addressing the effects of various parameters on engine performance, which also depends on the fuel properties. The effects of engine speed and load, equivalence ratio, ignition timing, CR, and coolant temperature on the heat flux have been addressed for well-known fuels like gasoline and natural gas. Well-established techniques for measuring optimal ignition timing like using one or a combination of brake torque, specific fuel consumption, and brake thermal efficiency have been used as indicators. Brake thermal efficiency peaks at maximum brake torque while specific fuel consumption at mean brake torque [**1–5**] is minimum. Researchers have come up with different parameters that can be derived from evolving conditions within the cylinder and have identified them as combustion descriptors since they include a compact description of the combustion phasing [**6**]. The effect of ignition timing on the thermal loading parameters including the exhaust temperature has been addressed [**7**, **8**]. In some cases, changes in the combustion chamber design are desired from the point of better mixing of fuel and air for ensuring complete combustion. However, the details of specific modifications would depend on the type of gaseous fuel [**9–11**]. With these changes in place, the differentiating factors between liquid and gas-fuelled engines for the same volumetric capacity would be the maximum power delivered by the engine, efficiency, and the

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exhaust emissions. Recently, gas engines have gained popularity towards using piped natural gas, compressed natural gas, and biomass-derived gaseous fuels, like producer gas, digester gas, landfill gas, resulting in the advent of fuel specific gas engines.

The use of producer gas as an engine fuel for power generation has been explored less, except during the Second World War [**12**, **13**], compared with biogas over a range of engine capacity. Small power engines around 50 kW for distributed power generation are an important facet for many developing countries. These engines are capable of meeting the unmet electrical energy demand of the rural population in an island mode using biomass-derived fuel. It must be emphasized that producer gas engines are not commercially available at these capacity ranges from any standard engine manufacturer in India.

It is important to stress that the properties of producer gas are different compared with those of other fuels. Table 1 provides the comparison of properties with natural gas. It is clear that producer gas has different stoichiometric ratio, energy density, flame speed, adiabatic flame temperature, and the reactant-to-product mole ratio compared with natural gas. These parameters exert an influence on the overall performance of the engine including the maximum power output of an engine [**14**, **15**]. Laminar flame speed is an important parameter that determines the flame propagation inside the cylinder for the gaseous fuel, and it is higher for producer gas compared with natural gas due to the presence of hydrogen. Further, these parameters are important in identifying the optimum settings like the ignition timing and 'air-to-fuel ratio'.

From the ideal relationships [**1**] on engine performance, one has

$$\text{Power} = \frac{\eta_f \eta_v N V_d Q_{hv} \rho_a \lambda}{2} \tag{1}$$

$$\text{Mean effective pressure} = \eta_f \eta_v Q_{hv} \rho_a \lambda \tag{2}$$

$$\text{Otto cycle efficiency} = 1 - \frac{1}{r_c^{\gamma-1}} \tag{3}$$

where $\eta_f, \eta_v, V_d, Q_{hv}, \rho_a$, and λ are the thermal conversion efficiency, volumetric efficiency, displacement volume, lower heating value of the fuel, air density, and fuel-to-air ratio, respectively; N is the engine speed (r/min), r_c the CR, and γ the ratio of specific heats.

From the above expressions (1) to (3), power output is related to the CR through the mean effective pressure and the cycle efficiency. Heywood [**1**] indicates that few studies have focused on the effect of CR on engine performance and efficiency over a wide range of CRs in the case of spark-ignited engines. The basic limitation in using higher CR for spark-ignited engines arises out of the properties of the fuel used. The range of operating CR found in the literature is 8–14. For spark-ignited engines with CR less than 12, for a unit change of CR, the output changes by about 3 per cent [**1**].

Ahrenfeldt et al. [**16**] have presented the results for a small capacity engine with and without supercharging and have shown an increase in power output and efficiency with supercharging. They compare the performance of two engines of similar engine capacities and different CRs and suggest that an increase from 8.9 kW (9.5:1) to about 10.1 kW (18.5:1) is possible. It is not evident from the results that this is the maximum achievable and there is no mention of any knocking based on in-cylinder pressure measurements. A simple analysis by Dasappa [**15**] suggests that increasing CR by 1 point increases the output by about 3 per cent, which does not appear to have been the case with the work by Ahrenfeldt et al. [**16**]. Furthermore, based on the available information from Heywood [**1**] for a fully open throttle, the change in efficiency is about 3 per cent per unit change in CR in the range of $r_c \leqslant 12$ and is about 1.8–2.4 in the range of CR 12–17. This has been validated by Dasappa [**15**], who has shown that there is about 1.5 per cent increase in efficiency for every point change in CR. Tinaut et al. [**17**] have carried out analysis to predict the engine performance using Engine Fuel Quality (EFQ), which is the combined effect of stoichiometric air–fuel ratio and stoichiometric mixture heating value, both depending on the producer gas composition. The estimation of engine power made using the EFQ parameter indicates that power at full load is reduced to about two-thirds of the maximum obtained with a conventional liquid fuel. They use some of the results of the work done by the present authors for the analysis.

Table 1 Properties of natural gas and producer gas

Fuel + air	Fuel LCV (MJ/kg)	Air/fuel at ($\Phi = 1$)	Mixture (MJ/kg)	Φ, limit		S_L (limit; cm/s)		S_L, $\Phi = 1$ (cm/s)	Peak flame temperature (K)	Product/reactant mole ratio
				Lean	Rich	Lean	Rich			
Natural gas	50.2	17.2	2.76	0.54	1.69	2.5	14	35	2210	1.00
Producer gas*	5.00	1.35	2.12	0.47	1.60	10.3	12	50	1800	0.87

Note: *Producer gas composition: CO, 19%; H_2, 20%; CH_4, 1.5%; CO_2, 12%; N_2, rest.

This article presents the results on the adaptation of a six-cylinder natural gas engine of about 55 kW capacity using producer gas as the fuel [**18**]. Based on the detailed performance evaluation, necessary changes for efficient operation are highlighted for the changed fuel. This article also discusses the preliminary efforts towards improving the performance using a turbocharger. It is also important to state that commercial availability of producer gas engines is limited and modifications to diesel engine are being pursued by researchers. With this background, the current efforts are to adapt a gas engine designed to operate on natural gas for using producer gas a fuel, which will help in further research.

The specifications of the engine and the various methods used for the performance are presented in Section 2 on 'Materials and methods'. Section 3 is devoted to the testing and performance evaluation of the natural gas engine using producer gas and identifying the areas of further testing. Section 4 provides the results of the performance on longer duration testing. Section 5 presents the results from using a turbocharger to increase the power and Section 6 provides a brief outcome of this research as conclusions.

2 MATERIALS AND METHODS

The 6B series natural gas engine manufactured by Cummins India Limited was chosen for the testing with producer gas. The engine is a six-cylinder, naturally aspirated, water-cooled engine coupled to a 50-kVA alternator to operate on producer gas. The variable speed natural gas engine is rated for a maximum shaft output of 55 kW at a constant speed of 1500 r/min [**16**]. The detailed specification of the engine is shown in Table 2.

For constant supply of producer gas fuel, a biomass-fuelled gasifier was used. The gasifier used is an open-top down-draft gasification system with necessary cooling and cleaning arrangements [**9**, **19**]. The engine generator system was connected to an electrical load bank for loading the engine generator system.

The biomass gasification technology package consists of a reactor, a gas cooling, and a cleaning system. A typical gasifier system configuration is as shown in Fig. 1. The novel open-top down-draft reactor design is a ceramic-lined cylindrical vessel with a bottom screw for ash extraction. In brief, the reactor has air nozzles and an open top for air to be drawn into the system, resulting in dual air entry into the reactor. The dual air entry, i.e. air drawn from the open top of the reactor and the nozzles permits establishment of front-moving propagation towards the top of the reactor. This establishes a large thermal bed of high temperature (~700 K) inside the reactor, thus improving on the residence time [**19**]. The details of the gasification technology are discussed in the work by Dasappa et al [**19**, **20**]. A screw-based ash extraction system would allow for the extraction of the residue at a predetermined rate, depending on the ash content in the biomass

Table 2 Engine configuration details [**18**]

Engine model	Cummins India Limited, 6B series
Bore × stroke	102 mm× 120 mm
Number of cylinders	6 mm
Displacement	5.9 L
CR	10.5:1
Engine shaft output	55 kW
Combustion chamber	Flat cylinder head and shallow bowl-in piston

Note: CR: compression ratio.

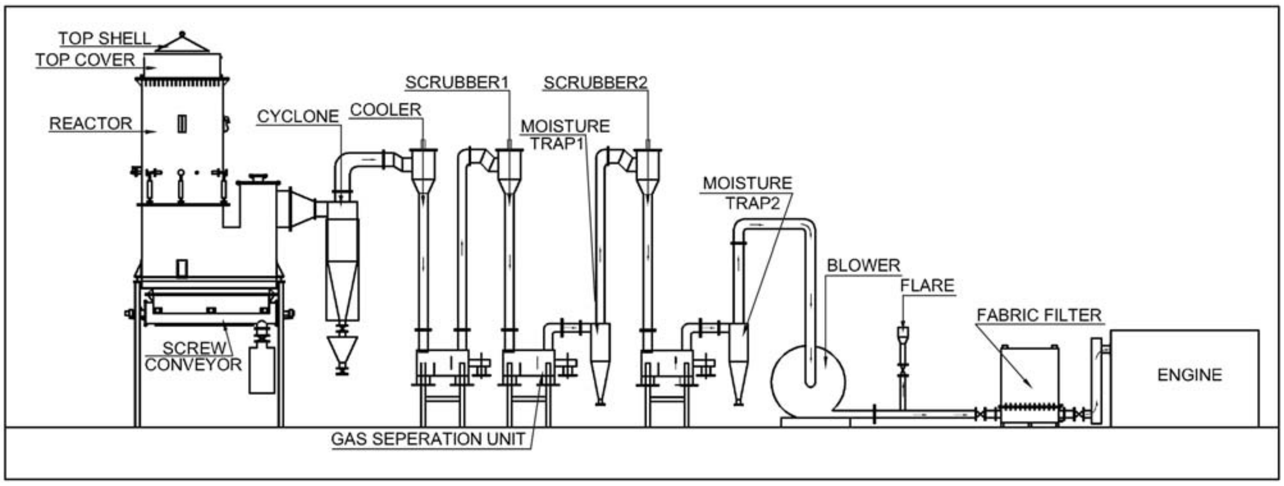

Fig. 1 Schematic of the open-top down-draft gasification system

feedstock. The gas conditioning system involves cyclone, scrubbers, and fabric filter. The gas is dehumidified using the principle of condensate nuclei to reduce the moisture and fine contaminants present in the gas. Further, a blower is employed to provide necessary draft for the gasifier operation and also to deliver producer gas at the required pressure for the satisfactory operation of the engine.

Biomass consumption rate was measured using topping-up method and ash/char extraction rate using the screw conveyor at regular intervals depending upon the load. Pressure drops across various system elements like the reactor outlet, cyclone outlet, scrubbers, and gas temperature at the reactor exit were recorded at periodic time intervals. Similarly, gas flowrate measurement using a venturimeter and gas composition measurement using a SICK Maihaik online gas analyzer were done for monitoring the gas quantity and quality at regular time intervals. The engine-related parameters like load, ignition timing, frequency, and exhaust composition were also monitored using a Quintox exhaust gas analyzer. Governor response to load changes was also measured.

3 EXPERIMENTS WITH PRODUCER GAS

3.1 Gasifier tests

In order to ensure consistency in gas quality, the 60 kg/h gasifier system available in the laboratory was tested independently. The producer gas was evaluated for gas quality (tar and particulate) and gas composition. A typical gas composition trace is presented in Fig. 2. The gas composition was measured over a period of about 5 h during one of the test operations. Gas composition was found to be CO and H_2 in the range of 19 ± 1 per cent, CH_4 1.8 ± 0.4 per cent, CO_2 9 ± 1 per cent, and the rest N_2, amounting to a gas calorific value of about 4.7 ± 0.2 MJ/kg. 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 75–80 per cent. Tar and particulates were measured using wet method [**21**] with anisole as the solvent. Test results indicated that the tar and particulates in the gas were found to be 10 ± 2 mg/m^3 and 15 ± 4 mg/m^3, respectively. Based on the available literature and the authors own experience, it is recommended that the contaminants, tar, and particulates be less than 20 mg/m^3 each to eliminate any fouling of engine components like the intake manifold, throttle valve, and intake valves.

It is evident from the gas analysis that the composition is nearly constant and has less than 5 per cent variation in the calorific value. These results establish that the gasifier generates nearly constant gas quality for the evaluation of the engine performance.

3.2 Establishing engine peak power output

The engine under test being designed to operate on natural gas, whose fuel properties are different compared with producer gas as indicated in Table 1, required study towards establishing some of the critical parameters that determine the peak power output of the engine. Thus, the initial efforts were focused in establishing the peak power output from the natural gas engine using the producer gas as the fuel. A producer gas carburetor system was integrated with the gas engine to ensure optimal air-to-fuel (A/F) mixture for combustion over a range of loads. Based on the properties of producer gas, especially the laminar flame speeds, the combustion process within the cylinder is shown to result in different combustion phasing [**14**]. Ensuring near-constant calorific value and A/F ratio, ignition timing was varied to arrive at the peak power output from the engine. The power output values at varying ignition timing are presented in Table 3. The ignition timing varied between 15° and 28° Before Top Dead Centre (BTDC). It was found that the maximum power realized with the engine was 27.2 kWe (kW electrical, the generator output) at 20° BTDC.

On establishing the ignition timing for peak power, further testing was carried out to evaluate the performance of the engine. In order to achieve

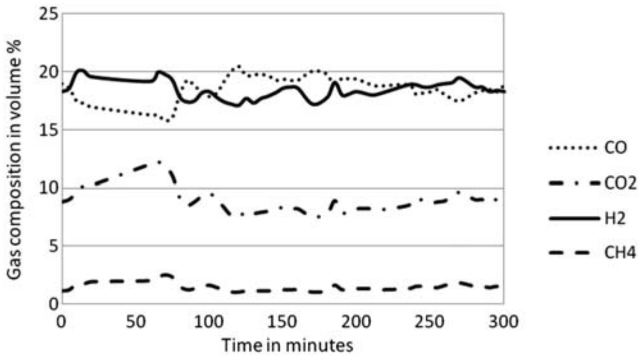

Fig. 2 Gas composition with time

Table 3 Power output at varying ignition timing

Ignition timing (°), BTDC	Power output (kWe)	Remarks
15	25.4	
20	27.2	Optimum
22	25.5	Original setting for natural gas
28	25.7	

steady operating conditions, the engine was operated on 24 h cycles. During these tests, biomass consumption, gas composition, gas flowrate, pressure drops on the gasification system and the current, voltage, air flowrate, engine exhaust temperature, and flue gas composition on the engine generator set were monitored.

From the biomass consumption data at the engine peak load condition, the specific biomass consumption (SBC) was found to be in the range 1.45 ± 0.05 kg/kWh, which translates to specific energy consumption (SEC) at 22 MJ/kWh and biomass calorific value of at 15 MJ/kg with 12 per cent moisture content. This value of SEC corresponds to an overall efficiency (ratio of the output electrical energy to the energy content in the biomass) of 16 ± 1 per cent. Overall efficiency is the combination of gasification and engine efficiency. With the measured cold gas efficiency (from biomass to gas) at 77 per cent, the engine efficiency (from gas to electricity) amounts to 21 ± 1.5 per cent. It is reported that the SEC for this engine using natural gas as the fuel is 12.7 MJ/kWh, amounting to an overall efficiency of about 29 per cent [**16**]. The extent of loss of efficiency with producer gas is about 8 per cent points compared with the natural gas operation. This accounts for about 28 per cent lower efficiency, a serious departure that calls for further research.

3.3 Reasons for lower efficiency

Investigations into the reasons for lower efficiency revealed that parameters with respect to the combustion process were found to be acceptable. The engine exhaust composition in terms of CO was well below 0.2 per cent and the oxygen in the exhaust was about 3 per cent, suggesting that oxygen is available for combustion. However, the exhaust temperature was higher by about 80 K compared to the baseline data with natural gas, consistent with the results from earlier work with a producer gas engine [**9**].

Based on the energy balance, it was difficult to explain the extent of the loss in efficiency. In arriving at the exhaust gas losses, both chemical energy due to unburnt CO and sensible heat were considered. Failing to arrive at a consensus on the loss in efficiency, systematic analysis of the energy flows into various subsystems revealed that the fraction of the engine shaft power consumed by the radiator fan was high. The 6B series engine under consideration had a radiator designed for natural gas engine shaft output for about 55 kW. This factory-assembled engine had a higher capacity radiator. Based on engine-related data, it became evident that the cooling system that was employed (radiator + fan) was overrated for the de-rated power output with producer gas. This aspect was further examined by making detailed measurements on the engine cooling system, presented in Table 4.

With the standard factory set configuration of fan and radiator for natural gas operation, the water inlet and outlet temperatures in the radiator at 27 kWe were 351 and 323 K, respectively, amounting to a temperature difference of about 28 K. During steady-state operation of the engine, the maximum temperature of the coolant was about 333 K, which is much lower than the recommended temperature 353 K. However, literature recommends setting the

Table 4 Performance with factory set and new fan + radiator

Load (kW)	Exhaust temperature (K)	Lubricating oil pressure (kg/cm^2)	Water temperature, radiator inlet (K)	Water temperature, radiator outlet (K)
Factory set configuration as in natural gas operation				
5.6	679	2.7	350	322
9.9	695	2.7	351	322
12.8	710	2.8	351	321
16.0	722	2.8	351	322
17.9	732	2.8	352	323
25.5	753	2.8	351	323
26.7	759	2.9	352	323
After changing the fan + radiator				
3.4	659	3	350	350
4.5	659	3	351	350
12.7	688	3	352	348
14.9	703	2.9	352	349
17.0	704	2.9	352	349
20.4	703	2.8	352	349
23.5	737	2.8	353	347
24.6	749	2.9	352	347
25.5	752	2.8	352	348
26.5	761	2.8	353	348
27.7	765	2.8	354	348

coolant temperature as high in order to reduce heat loss in the engine [**1**], thereby improving the engine efficiency. Based on this analysis, it was evident that reducing the auxiliary power consumed by the fan would result in an improvement of the efficiency.

From the above analysis, a lower capacity radiator and fan system was mounted on the engine. The performance evaluation of the total package was repeated similar to the earlier one. As indicated in Table 4, with the new radiator and the fan system, the exit coolant temperature increased by 30 K, with the radiator inlet temperature at 348 K. With this new arrangement, the peak power output increased to about 30 kWe, indicating an increase in power output by about 10 per cent. SBC was in the range 1.1–1.2 kg/kWh, amounting to SEC of about 17.5 MJ/kWh. This value of SEC amounts to an engine efficiency of 27 per cent (gas to electricity). The maximum temperature of the coolant was within the safe limit, 353 K. Thus, optimizing the size of the radiator unit improved the power output from engine by about 10 per cent and reduced the specific fuel consumption by about 15 per cent. The reported engine efficiency by Ahrenfeldt et al. [**16**] is 30 per cent with a producer gas having a calorific value of 5.6 MJ/Nm^3.

Unlike fossil fuel, producer gas is generated locally to fuel an engine and it is important to establish the quality of gas, a critical parameter while using producer gas, with respect to the amount of tar and particulate that could affect the engine operation. Apart from the gas quality measurements to ensure the quantitative estimation of the contaminants, as indicated in Section 2, the engine intake components like the manifold, throttle valve, and the carburetor were examined prior to and after the test. During this period, the engine had consumed over 9000 m^3 of gas. The amount of deposit was too small to make any kind of meaningful gravimetric measurements. Figures 5 and 6 show photographs of the throttle valve and intake manifold before and after the trials. The evidence in the photographs testifies the quality of producer gas as acceptable without any fouling tendency, an important requirement for engine operation.

4 FURTHER OPERATIONS ON THE PRODUCER GAS ENGINE

Having established the component level integration of the producer gas engine with optimal ignition, optimized capacity radiator with fan, and also the quality of producer gas, additional tests as discussed below were carried out.

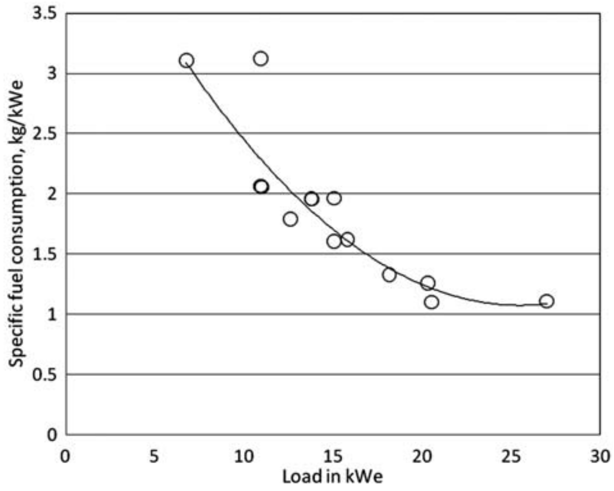

Fig. 3 Variation of SFC with load

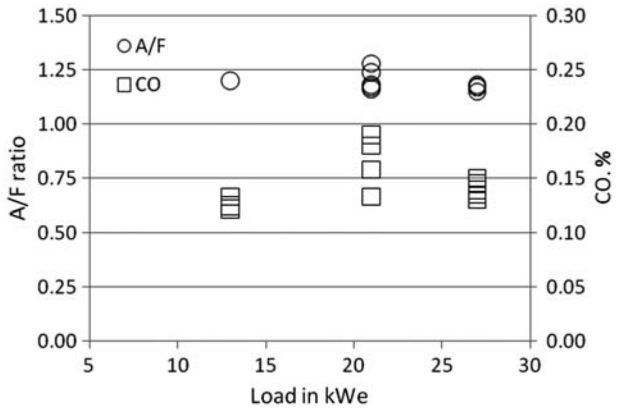

Fig. 4 Variation of air-to-fuel ratio in the mixture and CO in the exhaust with load

4.1 Long duration steady-state tests

The engine operations were found to be smooth at part as well as full load. During the course of these trials, relevant information of the gasifier and gas engine were continuously monitored at a time interval of 30 min. On the gasifier island, the biomass consumption was recorded in order to establish specific fuel consumption. Similarly on the gas engine side, exhaust emissions like CO and NOx were measured, apart from the electrical load and the generator frequency.

In order to establish operational data under varying load conditions, tests were carried out on the engine at loads of 10 per cent, 25 per cent, 50 per cent, 75 per cent, and 100 per cent, which correspond to 2.7, 6.8, 13.5, 20.1, and 27.7 kWe, respectively. The engine was tested at 27.7 kWe for over 100 h.

Figure 3 shows the variation of SBC with load measured during a 400-h operation. At 2.2 kW, the fuel consumption was over 3 kg/kWh reducing to about 2 kg/kWh at 50 per cent load. At 27 kWe, SBC

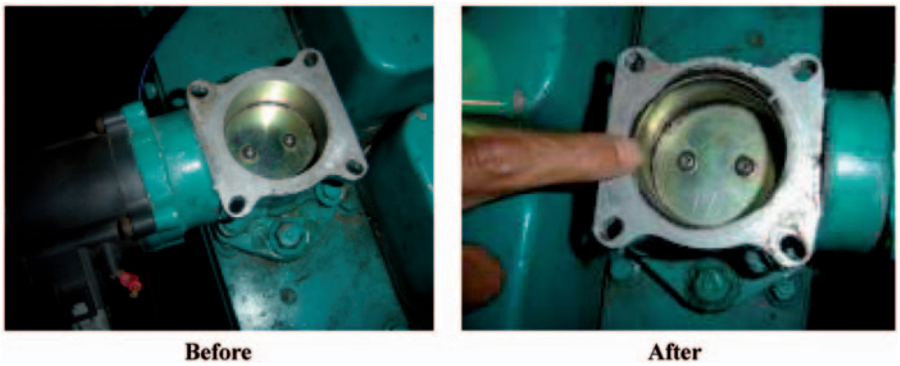

Fig. 5 Throttle valve before and after 100 hours

was about 1.1 kg/kWh. This value amounts to biomass-to-electricity efficiency of about 22 per cent and the gas-to-electricity of 28 per cent.

4.2 Air-to-fuel ratio measurement

It is important that the engine responds to varying load conditions, a requirement essential in a distributed power generation package. The patented producer gas carburetor used in conjunction with a zero pressure regulator ensures nearly constant air-to-fuel ratio (A/F) against load variation. Based on the air and gas flow measurements, the A/F was obtained for varying load conditions. Table 5 presents the performance of the carburetor. Oxygen in the mixture was also measured to confirm the A/F. From the results, it is clear that over the range of gas flowrates, the A/F has been maintained in the range of 1.25 ± 0.05, which provides an excellent control on the mixture.

Figure 4 presents the results on the A/F ratio for varying engine load and CO concentration in the engine exhaust gas. Air and gas flowrates were independently measured at each load to arrive at the A/F ratio. The A/F ratio of the mixture was found to be between 1.3 and 1.15 at varying load condition, suggesting that the carburetion system performs satisfactorily in maintaining the A/F in a certain bandwidth within the flammability limits as indicated in Table 1 for engine operations. The CO level in the engine exhaust varies from 0.12 per cent to 0.2 per cent (0.64–0.95 g/MJ) for a varying A/F ratio at a particular load.

The exhaust emission at various loads is shown in Table 6 in accordance with ISO 8178 D2, 5-Mode Test Cycle for constant speed generator set with intermittent load. The weightage given for 10 per cent, 25 per cent, 50 per cent, 75 per cent, and 100 per cent loads were 10 per cent, 30 per cent, 30 per cent, 25 per cent, and 5 per cent, respectively, as per the ISO 8178 D2. The compiled data sets are

Table 5 Measurements on gas, air, and mixture composition

Gas flowrate ($\times 10^3$ kg/s)	Air flowrate ($\times 10^3$ kg/s)	A/F ratio	Oxygen in the mixture (%)
12.7	17.1	1.35	13
11.9	16.3	1.37	14
21.7	28.3	1.30	11
21.5	27.0	1.26	11
21.0	26.7	1.27	11
25.7	34.0	1.32	12
26.6	34.7	1.30	11
27.5	35.1	1.28	11

Fig. 6 Intake manifold at the completion of 100 hours

shown in Table 6 and they suggest that the CO and NOx levels were well within the Central Pollution Control Board norms of 0.97 and 1.67 g/MJ, respectively. The SO_2 in the exhaust could be contributed partly by the engine lube oil and partly from biomass. This aspect needs further investigations.

Table 7 compares the overall engine performance of the producer gas with that of the natural gas. The engine was rated for 55 kW shaft output in natural gas engine with a brake mean effective pressure (BMEP) of 672.3 kPa. With the producer gas, the engine shaft power output was 30 kW after accounting for the alternator efficiency, the BMEP is 408.3 kPa, which is about 40 per cent lower than the natural gas engine. Further investigations into the engine in-cylinder processes are essential to improve the engine output and thus the BMEP.

Table 6 Emission results from the engine

Load (kWe)	Load (%)	O_2 (%)	CO (g/MJ)	NOx (g/MJ)	SO_2 (g/MJ)
2.20	8	2.8	0.93	0.00	0.03
6.75	25	3.1	0.98	0.01	0.04
13.7	51	2.7	0.94	0.01	0.02
20.4	76	3.0	0.72	0.02	0.02
27.0	100		0.92	0.03	0.02
ISO 8178 D2, 5-Mode			0.94	0.02	0.02

Table 7 Comparison of engine performance with producer gas and natural gas

Engine parameter	Natural gas	Producer gas
Displacement (L)	5.9	5.9
Shaft power (kW)	55	30
BMEP (kPa)	672.3	375.6

Note: BMEP: brake mean effective pressure.

Table 8 Data on variation of frequency with load

Initial condition during loading and load throw-off		Final condition after loading and load throw-off		
Load (kWe)	Frequency (Hz)	Load (kWe)	Frequency (Hz)	Recovery time (s)
0	52.09	6.9	52.09	1
6.9	52.09	0	52.04	1
0	52.09	13.8	52.09	1.4
13.8	52.09	0	51.99	2
0	52.09	21.5	52.06	3
21.5	52.09	0	52.06	3.5
0	52.06	25	52.06	6
25	52.06	0	52.06	3.5
0	52.06	27	52.00	6
27.8	52.06	0	52.10	3.5

4.3 Governor response

Experiments were conducted to check the response of the governor to the load changes. The data on the frequency variation were acquired on to a computer using a data acquisition using IoTech DAQ2000. The details of the measurements made are depicted in Table 8. Critical experiments were conducted with 100 per cent loading and load throw-off conditions. The performance of the engine was been found to be satisfactory; the engine was able to equilibrate in about 6 s for 100 per cent loading and similarly for load throw-off it was about 3.5 s. The no-load generator frequency was set at 52 Hz.

Three cases (6.9, 21.5 and 27 kWe) have been depicted in Table 8, from no load to about 25 per cent, 75 per cent, and 100 per cent of the rated load and subsequent load throw-off. It is important to notice that with increasing load, the governor response becomes sluggish towards frequency recovery. Frequency recovery time is defined as the time interval between the departure from the steady-state frequency after a sudden specified load change and return to the specified steady-state frequency. From Table 8, it is evident that the governor response is slightly better for load throw-off condition in comparison to block loading. Thus, the governor response time for frequency recovery increases with loading and virtually doubles when the load increases from 20 to 27 kWe capacity. This aspect needs further investigation by mapping the transient response of each of the elements in the gas circuit to establish the time constants and to address the issues.

Using the ISO 8528-5:1993(E) code, the governor response is classified as G1, G2, and G3 (s), based on the engine frequency/voltage recovery time. ISO 8528-5:1993(E) code identifies the following ranges

Table 9 Lubricating oil analysis with time

Parameter	Reference range for fresh oil	ASTM standards		Sampling time (h) 200	300	400	500
Flash point, COC (°C)	230	Caution limit	Indication for end of life limit	220	220	220	220
Insoluble pentane (wt.%)	0.04			0.14	0.07	0.26	0.09
Insoluble toluene (wt.%)	0.02			0.06	0.05	0.17	0.07
Kinematic viscosity at 40°C (cSt)	113.9			103.01	104.56	99.15	99.15
Kinematic viscosity at 100°C (cSt)	14.75	16.96	17.46	12.98	13.38	12.54	12.65
Nitration Abs (cm^{-1})	5	10	15	7	18	39	49
Oxidation Abs (cm^{-1})	5	15	20	12	20	43	41
TAN (mg KOH/g)	1.06	3.06	3.56	2.1	2.07	2.96	2.47
TBN (mg KOH/g)	5.68	2.2	2.8	4.4	3.9	4.1	3.8
Viscosity index	136			120	126	121	122
Water (wt.%)	0	0.2	0.2	0	0	0	0

$G1 \leqslant 10$, $G2 \leqslant 5$, and $G3 \leqslant 3$. $G3 \leqslant 3$ means that the governor is classified as G3 if the recovery time is within 3 s from the load variation. Transient analysis was not carried out on the system.

From the above set of experiments, it is clear that power generation system using a gasifier-coupled gas engine can be used for varying load applications, and limit the block loading to less than 30 per cent of the engine capacity for better engine response. The governor performance is one of the parameters used for classifying the engine on the frequency response time. It is found to be good up to 75 per cent of the load, i.e. in the range up to 20 kW as G3 class and G2 beyond.

4.4 Lube oil sample analysis

Unlike other gaseous fossil fuels processed in a centralized facility, producer gas is generated *in situ* to fuel an engine. The quality of producer gas used reflects on the engine lubricating oil contamination. During the performance tests, the engine lube oil samples were drawn at end of 200, 300, 400, and 500 h and samples were analysed by Cummins India Limited. The results are shown in Table 9.

Table 9 also provides the reference range for various parameters. Based on the lubricating oil analysis, it is clear that the quality of the oil is satisfactory at the end of 500 h. Total Base Number (TBN) determines how effectively the acid formation during the combustion process is controlled. Higher the TBN, the more effective it is in suspending wear-causing contaminants and reducing the corrosive effects of acids over an extended period. Thus, TBN, an indicator of the alkaline nature of the lube oil, which gets altered due the presence of phenolic compounds in the tar, is found to be well within the limits. Viscosity index is another parameter that reflects the influence of particulate matter in the producer gas. With higher particulate content in the lubricating oil, the viscosity tends to increase. The data in Table 9 suggest that the viscosity index is also found to be within the limits. It must be mentioned that the nitration and oxidation numbers are indicators for oil degradation; the changes have not resulted in the increase of the viscosity of the oil. This aspect needs further study. The above analysis ensures that the producer gas is of requisite quality for engine operation.

5 PRELIMINARY EXPERIMENTS WITH TURBOCHARGER

Having established the peak power from the naturally aspirated producer gas, the authors addressed the issue on improving the power output of the gas engine using a turbocharger manufactured by Holset. The compressor is rated for a mass flow of about 275 kg/h with a pressure ratio of 1:1.35.

Adding the turbocharger required introduction of the new engine elements like after-cooler to reduce the air–gas mixture temperature after the compression, a separate heat exchanger for oil, coolant tank, and a pump to cool the compressed gas–air mixture. The gas and air circuit was similar to the naturally aspirated test conditions. In this configuration, the mixture of gas and air after the carburetor is drawn through the turbo-compressor before being taken into the engine intake manifold. The temperature of the gas–air mixutre at the intake manifold was in the range 318 ± 2 K.

5.1 Performance with turbocharger

Preliminary testing of the engine with turbocharger revealed that total mass flow into the engine increased by about 30 per cent from 190 to about 255 kg/h. Under these conditions, the boost pressure ratio measured was about 1.2. The increased mass flow enhanced the electrical output to 36 kWe, an increase of about 9 kWe, amounting to about 30 per cent increase in load compared with the naturally aspirated operation.

Table 10 summarizes the results for BMEP in both naturally aspirated and turbocharged modes. The natural gas engine shaft output in the turbocharged mode is rated at 77 kW. The initial evaluation of the engine with the turbocharger using producer gas as the fuel resulted in an engine shaft output of 40 kW. The BMEP achieved is about 50 per cent of the natural gas operation and there is an improvement in BMEP from 408 kPa in naturally aspirated mode to about 530 kPa in turbocharged mode. There is a significant scope for improving the BMEP by suitably choosing the turbine–compressor system.

Table 11 provides the measurements on the engine exhaust gas composition. CO and NOx levels are higher in the turbocharger mode compared with the naturally aspirated engine mode. This may be related to the operation of the engine close to stoichiometric

Table 10 Comparison of BMEP values of producer gas with those of natural gas

	Naturally aspirated		Turbocharger	
	Natural gas	Producer gas	Natural gas	Producer gas
Shaft power (kW)	55	30	77	40
BMEP (kPa)	672.3	408.3	1053.6	530.6

Note: BMEP: brake mean effective pressure.

Table 11 Comparison of the exhaust emissions in naturally aspirated and turbocharged operations

Species	Naturally aspirated	Turbocharged
Load (kWe)	27.7	36
CO (%)	0.817	0.192
NOx (ppm)	290	133
SO_2 (ppm)	28	50
CO_2 (%)	17.8	16.7
O_2 (%)	0.7	2.9

condition, inferred by the oxygen concentration in the exhaust. These measurements are done without any catalytic converter in the exhaust line.

It is also important to state that further investigations into the engine operation are essential. These would focus on generating in-cylinder process performance data along with the cylinder pressure–time variation, establishing heat and mass balance, both in naturally aspirated and turbocharged modes of operation. The BMEP is lower by about 50 per cent compared with natural gas. There is scope for improving the performance of the engine and address issues related to the emissions from the engine by carrying out in-cylinder measurements.

6 CONCLUSIONS

Systematic evaluation of a natural gas engine for producer gas operation is presented. Optimum ignition timing values for peak power, optimal air-to-fuel ratio, and the gas governing for varying load application have been achieved for producer gas operation. SBC of 1.2 ± 0.1 kg/kWh amounting to an overall efficiency of 22 per cent – biomass to electricity has been realized. The governor response is found to be good up to 75 per cent of the load, i.e. in the range up to 20 kW as G3 class and G2 beyond. Producer gas quality was found to be acceptable based on the component level evaluation and lubricating oil analysis. With the engine shaft power output at 30 kW, the BMEP is 408.3 kPa, which is about 40 per cent lower than the natural gas engine. Initial studies indicate that the power output can be enhanced using a turbocharger with after-cooler. It must be stated that the use of a producer gas engine for commercial operation has been established using a natural gas engine.

FUNDING

This work was supported by Ministry of New and Renewable Energy, Government of India, New Delhi.

ACKNOWLEDGEMENT

The authors thank Cummins India Limited for the support provided.

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