Development of producer gas engines

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Abstract: This paper summarizes the findings involved in the development of producer gas fuelled reciprocating engines over a time frame of six years. The high octane rating, ultra clean, and low-energy density producer gas derived from biomass has been examined. Development efforts are aimed at a fundamental level, wherein the parametric effects of the compression ratio and ignition timing on the power output are studied. These findings are subsequently applied in the adaptation of commercially available gas engines at two different power levels and make. Design of a producer gas carburettor also formed a part of this developmental activity. The successful operations with producer gas fuel have opened possibilities for adapting a commercially available gas engine for large-scale power generation application, albeit with a loss of power to an extent of 20–30 per cent. This loss in power is compensated to a much larger extent by the way toxic emissions are reduced; these technologies generate smaller amounts of toxic gases (low NO_x and almost zero SO_x), being zero for greenhouse gas (GHG).

Keywords: gas engine, producer gas, biomass, compression ratio, ignition timing

1 INTRODUCTION

In the recent times, gaseous fuels are gaining prominence as cleaner fuels for power generation via the internal combustion engine route; the power generation package includes both reciprocating engines and gas turbine machinery. Complete combustion with minimal emission is the key feature of gaseous fuels and this feature is currently being exploited the world over for power generation purposes. Among the clean sources of fuel for power generation, natural gas has been exploited largely due to significant availability in specific locations. Similarly, there is also an impetus on using gas generated from industrial and municipal wastes, namely diluted natural gas—biogas and landfill gas. As distinct from gas generation from biological/organic wastes by the biological conversion process, which is limited to non-lignaceous matter, the thermochemical conversion route (also termed gasification) can process any solid organic matter. The range of biomass includes agro-residues such as rice husk, sugarcane trash, and bagasse in compact or briquetted form. The resultant gas, known as 'producer gas' (PG), can be used for fuelling a compression ignition (CI) engine in the dual-fuel mode or a spark ignition (SI) engine in the gas-alone mode. Harnessing of energy from biomass via the gasification route is not only proving to be economical but also environmentally benign [**1**]. In fact, renewable energy is gaining popularity in Europe and the West, referred to commonly as the 'green energy' and its harnessing is encouraged through attractive incentives on the tariff by governments.

The technology of biomass gasification has existed for more than seventy years. Some of the work done during World War II was well documented by the Solar Energy Research Institute (SERI) [**2**]. Subsequent to World War II, the technology did not gain popularity on two counts, the first reason being the unrestricted availability of petroleum fuels the world over at a low cost and the other reason being technological problems relating to the presence of a high level of tar content in the product gas, which posed a threat to engine operations. Though there has been a sporadic interest in biomass gasifiers whenever there is an oil crisis, sustained global interest was developed only in recent times for reasons like greenhouse gas (GHG) emission reduction and carbon trading through clean development mechanisms.

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In addition, a steep rise in oil prices has had a severe impact on the industrial economy, which has forced many oil-importing countries to reconsider gasification technology and initiate improvements in them. The technology of an open-top, twin air entry, re-burn gasifier developed at the Combustion, Gasification and Propulsion Laboratory (CGPL) of the Indian Institute of Science (IISc) is unique in terms of generating superior quality producer gas [**3**].

The work reported in this paper is the cumulative effort of six years in realizing a producer gas engine. During this period, engines with a power level between 20 and 200 kW have been evaluated using producer gas fuel. This development work involving systematic and scientific investigation was necessary in order to erase some of the misconceptions associated with the low-energy density and potential gaseous fuel.

2 BACKGROUND

This development work was initiated against the backdrop of limited information and modest work reported in the field of producer gas engines. A literature survey in the field of producer gas based engines reveals that only modest research work has been accomplished since the inception of biomass/charcoal gasification systems. This could be attributed to two reasons, namely the non-availability of a standard gasification system that could generate consistent quality producer gas and misconceptions about producer gas fuel. The misconceptions are essentially related to the compression ratio limitation due to knock and de-rating. The knock tendency with producer gas can be expected to be better on account of the large fraction of inert gas it contains as compared to natural gas (NG). However, there has been no research octane rating test conducted on producer gas fuel and, moreover, it is not clear whether any established test procedure exists for producer gas, such as the methane number test for natural gas and biogas. One crude method of assessment is to test the fuel gas in standard engines and place them accordingly in the octane rating table. De-rating using producer gas could be expected on account of a reduction in the mixture energy density and the product–reactant mole ratio. These issues are addressed in the next section, which discusses the properties of producer gas vis-à-vis other gaseous fuels. The literature survey addresses some of the research activities conducted in Europe, America, and the Indian subcontinent.

It is reported that Europe exploited gasification technology the most during the petroleum oil crisis of World War II. Among the European nations, Sweden accounts for a large amount of work in the area of wood and charcoal gasification. The National Swedish Testing Institute of Agricultural Machinery [**4**] has reported extensive work on the design and development of closed-top charcoal and wood gasifiers for use in reciprocating engines. These reciprocating engines were mostly diesel engines mounted on trucks and tractors for operation in the dual-fuel mode. Many finer aspects relating to dual-fuel operation have been extensively reported, with cumulative operational experience exceeding a few thousand hours. However, whatever work was conducted on the producer gas alone, operation is either proprietary to the engine manufacturers or is not adequately reported in literature in the public domain. These engines were, however, in the compression ratio (CR) range of 10, either adapted from petrol engines or modified diesel engines. In recent times, Martin and Wauters [**5**] have reported work using charcoal gas and biomass based producer gas on an SI engine with a de-rating of 50 and 40 per cent respectively at a CR of 7. However, the same authors state a 20 per cent de-rating when worked with producer gas at a CR of 11. They indicate an upper limit of the CR of 14 and 11 for charcoal and biomass based producer gas respectively. However, there is no presentation of experimental evidence in favour of these results.

The American subcontinent also claims experimental work relating to producer gas engines. Tatom *et al.* [**6**] have reported working on a gasoline truck engine with a simulated pyrolysis gas at a de-rating of 60–65 per cent. The authors have also identified the optimum ignition timings as a function of speed. Parke *et al.* [**7**] and Parke and Clark [**8**] have worked on both naturally aspirated and super-charged gas engines. The authors state a de-rating of 34 per cent compared to gasoline operation and a lesser de-rating in the supercharged mode. The authors discuss aspects relating to the fuel–air mixture ratio, flame speed, and its relation to the ignition timing for producer gas operation. They have also identified the best possible mixture for maximum power and efficiency along with ignition timing at various speeds.

In the Indian subcontinent, work in the area of the producer gas engine has been reported by the Indian Institute of Technology, Mumbai. Shashikantha *et al.* [**9**], Shashikantha and Parikh [**10**], and Parikh *et al.* [**11**] have reported work on a gas engine converted from a naturally aspirated diesel engine at a CR of 11.5. The reason for limiting the CR is cited to be the knocking tendency; however, no experimental evidence is provided in support of this. The work is reported on a gas engine converted from a diesel engine with a modified combustion chamber. The modified combustion chamber of Hesselman (shallow W) shape is stated to enhance the in-cylinder turbulence by suppression of swirl and promotion of the squish effect. With the above modification, a power output of 16 kW is reported in the gas mode against a rated output of 17 kW in the diesel mode. The maximum thermal efficiency is stated at 32 per cent, which is close to the results in the CI (with diesel) mode at an output of 15 kW. It is quite surprising to note that the conversion efficiencies are the same, although the CRs are widely different. The authors also state an optimum ignition timing of 35° BTC (before top centre) compared to 22° BTC for natural gas on the same engine. With the producer gas stated to contain about 24.1% H_2, 21.5% CO, and 2.1% CH_4, the burning velocities ought to be higher than those of natural gas. This therefore requires the ignition timing to be located close to TC (top centre) as against what has been stated.

The only earlier experimental work in the higher CR range is reported by Ramachandra [**12**] on a single-cylinder diesel engine (CR = 16.5) coupled to a water pump. A power de-rating of 20 per cent was reported at an overall efficiency of 19 per cent without any signs of detonation. This work does not report detailed measurements such as the gas composition, pressure–crank angle diagram, and emissions, which are essential for systematic investigation and scientific understanding.

If the findings of earlier studies could be summarized, it becomes evident that no systematic investigation has been attempted so far in identifying whether the limitation of knock exists with producer gas operation at a CR comparable with diesel engine operation. This topic is worth analysing since producer gas contains a large fraction of inert substances (>50 per cent), and with laminar burning velocity being high (due to the presence of H_2), smooth operation at a higher CR does not seem impossible. These aspects are very vital in establishing the fact that close to comparable power (with a lesser extent of de-rating of 15–20 per cent) could be achieved with producer gas by operating engines at a higher CR.

3 PRODUCER GAS FUEL

Producer gas derived from biomass typically contains 18–20 per cent each of H_2 and CO, 2 per cent of CH_4, and the rest inert gases such as CO_2 and N_2. The lower calorific value varies between 4.5 and 4.9 MJ/kg, with the stoichiometric air–fuel ratio being 1.25 ± 0.05 on a mass basis. Some of the fundamental data relating to producer gas are compared with pure gases in Table 1 The comparison of producer gas with methane is more vital with regard to the internal combustion engine operation. This is because most of the engines operating on gaseous fuels are either close to pure methane (natural gas) or diluted methane (biogas, landfill gas). The fuel–air equivalence ratio Φ (actual fuel–air ratio)/(stoichiometric fuel–air ratio), at the flammability limits [**15**] compares closely for both the gases, but the laminar burning velocity for producer gas at the lean limits is much higher. The laminar burning velocity for producer gas (at 0.1 MPa and 300 K) is about 0.5 m/s [**15**], which is about 30 per cent higher than methane. It is argued that this feature demands lower advancement in the ignition timing, which needs consideration when arriving at the optimum ignition timing for the producer gas fuel.

Table 1 Properties of producer gas (PG) compared with pure combustible gases. (From references [**13**] to [**15**])

Fuel + Air	Fuel LCV (MJ/kg) [MJ/N m^3]	Air–fuel ratio at $\Phi = 1$ (mass) [mole]	Mixture (MJ/kg) [MJ/N m^3]	Φ (limit) Lean	Φ (limit) Rich	S_L (limit) (cm/s) Lean	S_L (limit) (cm/s) Rich	S_L at $\Phi = 1$ (cm/s)	Peak flame temperature (K)	Product–reactant mole ratio
H_2	121 [10.8]	34.4 [2.38]	3.41 [3.2]	0.01	7.17	65	75	270	2400	0.67
CO	10.2 [12.7]	2.46 [2.38]	2.92 [3.8]	0.34	6.80	12	23	45	2400	0.67
CH_4	50.2 [35.8]	17.2 [9.52]	2.76 [3.4]	0.54	1.69	2.5	14	35	2210	1.00
C_3H_8	46.5 [91.3]	15.6 [23.8]	2.80 [3.7]	0.52	2.26	—	—	44	2250	1.17
C_4H_{10}	45.5 [117.7]	15.4 [30.9]	2.77 [3.7]	0.59	2.63	—	—	44	2250	1.20
PG	5.0 [5.6]	1.35 [1.12]	2.12 [2.6}	0.47*	1.60†	10.3	12	50‡	1800§	0.87

PG: H_2, 20%; CO, 20%; CH_4, 2%.
* ±0.01.
† ±0.05.
‡ ±5.0.
§ ±50.

Like any other gaseous fuel, producer gas can be used for internal combustion engine operation provided the gas is sufficiently clean that contaminant does not accumulate in the intermediary passages to the engine cylinder. However, this fuel has largely been left unexploited due to additional perceptions, namely (a) an auto-ignition tendency at a higher CR and (b) a large de-rating in power due to the energy density being low. However, these perceptions need re-examination and clarification. The arguments against the classical view in favour of better knock resistivity are as follows. Firstly, the fact that the laminar burning velocity is high due to the presence of hydrogen (more so, with the gasifier system adopted in this work) might reduce the tendency for knock. Secondly, the presence of inert gases in the raw gas (CO_2 and N_2) might suppress the pre-flame reactions that are responsible for knocking on account of increased dilution. Also, as the maximum flame temperature attainable with the producer gas is lower compared to conventional fuels like methane, better knock resisitivity could be expected. An examination of the literature shows that producer gas has not been subjected to study on knock behaviour.

Further, there is a general perception that as producer gas is a low-density energy fuel, the extent of de-rating in power would be large when compared to high-energy density fuels like natural gas and liquefied petroleum gas. This could be misleading because what needs to be accounted for by way of comparison is the mixture energy density [**16**] and not the fuel energy density per se. Compared with CH_4, the mixture energy density for producer gas is lower by 23 per cent, as reflected in Table 1. The product–reactant mole ratio for producer gas is less than one. These two parameters could contribute to de-rating of engine output. However, it might be possible to reduce de-rating by working with engines of higher CR, perhaps higher than what has been examined by Das and Watson [**17**] using natural gas ($CR = 15.8$).

4 PRODUCER GAS CARBURETTOR

Designing a gas carburettor for producer gas fuel assumed major proportions as there were no carburettors available for such low-energy density gaseous fuels. The carburettors available for other gaseous fuels, namely natural gas, biogas, and landfill gas, are unsuitable due to widely different stoichiometric air–fuel requirements. The stoichiometric air–fuel ratio varies between 10 and 6 (on a volume basis) for fuels such as natural gas and biogas/landfill gas based on the methane content in the gas. However, the stoichiometric air–fuel ratio for producer gas is about 1.2–1.4 (on a volume basis) based on the constituents of the gas. The envisaged features in the gas carburettor are:

(a) the ability to maintain the required air–fuel ratio (1.2–1.5 : 1) with load or throttle variations;
(b) smooth operation with minimal pressure loss;
(c) shut-off of the fuel in the case of engine tripping or shut-down;
(d) on-line provision for air/fuel tuning during testing.

These features were incorporated in the development of a gas carburettor and are shown in Fig. 1. The carburettor is simple in design and does not have moving components. It has a separate port for air and fuel, where the individual ports could be modified or tuned to achieve the required air–fuel ratio. The carburettor is designed to operate in conjunction with the zero-pressure regulator. The combined pressure regulator and gas carburettor was located between the gasifier and the engine intake system, as shown in Fig. 1. The zero-pressure regulator ensures a gas pressure (downstream of the pressure regulator) identical to that of air pressure, which is achieved by connecting the air pressure line (downstream of the air filter) to the upper chamber of the regulator. This arrangement ensures that the regulator maintains the gas pressure close to that of air pressure ($\sim$a few

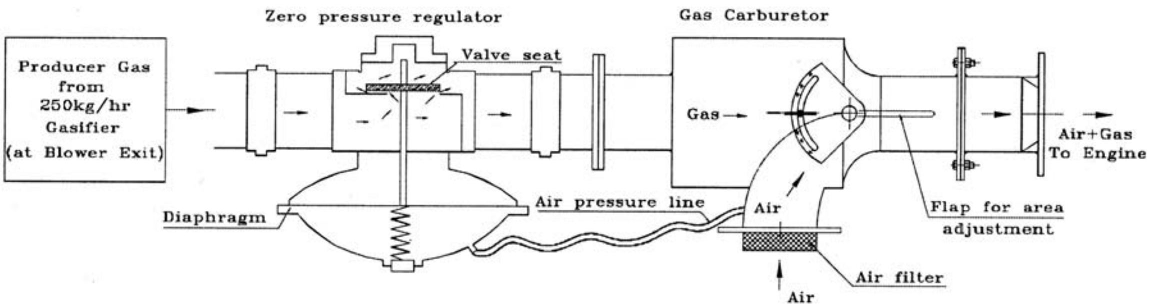

Fig. 1 Schematic of the producer gas carburettor with a zero-pressure regulator in the gas–air line circuit

mm below atmospheric pressure), thereby ensuring that the air–fuel ratio is set irrespective of the total mixture flowrate. Homogeneity of the fuel and air mixture entering the engine was effected by having sufficiently long interconnecting ducting along with a few bends (with a diameter sufficiently large to keep pressure losses to a minimum) between the gas carburettor and the turbocharger or the intake manifold.

Flow tests performed with the zero-pressure regulator and the gas carburettor showed reasonable functioning in terms of air–fuel ratio (A/F) control against total or mixture flow rate variations, as shown in Fig. 2. A flow test was conducted using a blower to simulate the engine suction. This simulation should hold good for a multicylinder engine where the pulsating flow is evened out, but possibly does not apply to a single-cylinder engine. The air and fuel flowrates were individually measured over a range of engine operating conditions. However, the homogeneity of the air and fuel mixture is not addressed here but is expected to be taken up as a separate study owing to its importance, particularly for naturally aspirated engines. The two cases shown in Fig. 2 correspond to area ratios for the air and fuel entry. These cases are possibly the extreme limits and the required operation point for the engine operation could lie in between them. The air–fuel ratio was reasonably constant beyond a specified mixture flowrate, with a relatively rich mixture at low mixture flowrates. This characteristic is desirable from the viewpoint of engine operation—a rich mixture for engine start-up and no-load operations and a relatively leaner mixture during part-load operation. However, for peak load operation stoichiometry or a rich mixture is desired, calling for an adjustment of the carburettor flap. Considering gas engine operation at the field level, the carburettor is designed in such a manner that in the event of load throw-off the flap of the carburettor could move to the full air flow (by motorizing) condition thus ensuring safety of the engine.

5 THE EXPERIMENTS

Three engines of the configuration given in Table 2 were tested. These are basically an engine–alternator set meant for power generation applications. One of these is a diesel engine (E1), which was converted to a spark ignition engine at the laboratory. The details of conversion are dealt with in earlier work [**18**, **19**]. Engine E1 was subjected to systematic investigations, wherein the engine was tested at varying CRs of 17, 14.0, 13.5, and 11.5. However, the engines E2 and E3 were tested at the fixed CR mentioned in Table 2. These two engines are essentially factory-converted spark ignition engines from the diesel engine frame mode to operate on gaseous fuels. Load tests were conducted on the engine in order to determine the maximum power delivered. Therefore, the air–fuel ratio and the ignition timing were tuned in order to derive maximum output.

The gas engine was connected to the biomass gasification plant, whose system elements are shown in Fig. 3. Experiments were initiated on the engines

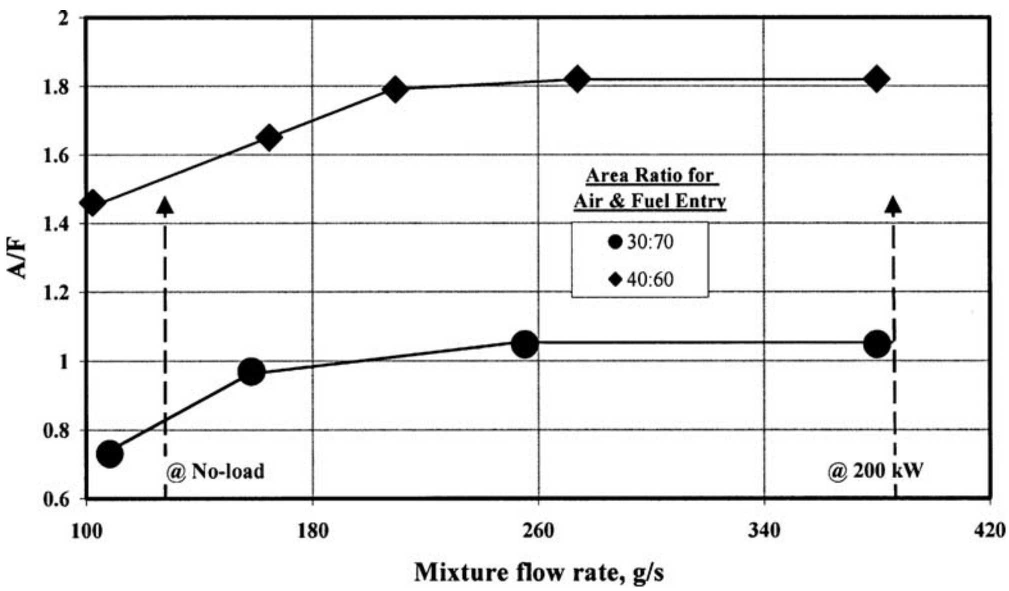

Fig. 2 Flow tests with the gas carburettor at varying area ratios for air and fuel entry

Table 2 Engine specifications

Parameter	Engine 1 (E1)	Engine 2 (E2)	Engine 3 (E3)
Make and model	Kirloskar, RB-33	Greaves, TBDV12	Cummins, G743G
Engine type	In-line, 3-cylinder, naturally aspirated diesel engine	'V' configuration, 12-cylinder, turbocharged with after-cooler gas engine	In-line, 6-cylinder,4-stroke, naturally aspirated gas engine
Rated output—1500 r/min at sea level	28 kW, with diesel fuel	290–310 kW (estimated) using diesel	101 kW, with natural gas
Net output*—at Bangalore, approximately 1000 m above sea level	24 kW, with diesel fuel	240–258 kW, with diluted natural gas	84 kW, with natural gas
Bore × stroke (mm)	110 × 116	128 × 140	130 × 152
Total displacement (L)	3.3	21.6	12.1
Specific power (kW/L)	8.5 (with diesel)	13.4–14.4 (with diesel)	7.0 (with NG)
Compression ratio (CR)	17	12	10
Bumping clearance (mm)	1.5	1.6	11
Combustion chamber	Flat cylinder head and hemispherical bowl-in-piston type	Flat cylinder head and cylindrical bowl-in-piston type	Flat cylinder head and shallow bowl-in-piston type
Squish area (%)	70	68	35
Spark plug type and location—gas mode	Cold, offset from the axis of cylinder by 8 mm	Cold, offset, located in the vertical plane close to the outer edge of the bowl	Central
Conversion/modification, if any	Converted to SI engine	Producer gas carburettor adapted	Producer gas carburettor adapted

*Net output after deducting power drawn by engine accessory drives.

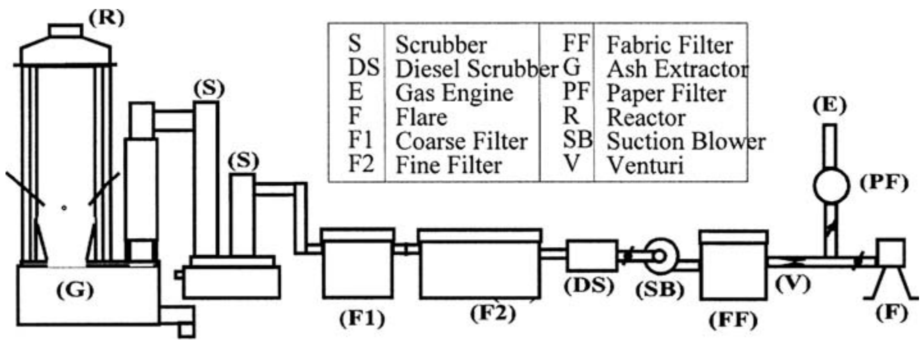

Fig. 3 Schematic of the open-top re-burn gasifier connected to the gas engine

only after the gasifier system stabilized, i.e. attained steady state operation in terms of generation of consistent quality gas. A typical timescale for attaining a steady state of operation from the cold start was 2–3 hours. During this period the gas was flared in a burner. The gas composition was determined using on-line gas analysers, pre-calibrated using a known producer gas mixture. The calibrations of these analysers were checked at random time intervals so as to minimize errors in long-duration operation. Typically the gas composition at the time of start of the engine test was $19 \pm 1\%$ H_2, $19 \pm 1\%$ CO, 2% CH_4, $12 \pm 1\%$ CO_2, $2 \pm 0.5\%$ H_2O, and the rest N_2. The mean calorific value of the gas varied around 4.5 + 0.2 MJ/kg. The contaminant levels (particulate and tar) in the gas at the entry to the engine were of the order of 60 mg/N m^3 (ppm) in the case of the tests with engine E1, but in the tests with engines E2 and E3, it was as low as 2 and 0.02 ppm respectively. This low level of contaminant was possible by employing a much superior gas scrubbing system during the experiments on engines E2 and E3. The feedstock used for gasification is Causurina species wood and coconut shells with a moisture content between 12 and 15 per cent on a dry basis (sun-dried wood).

All the tests on the engine were conducted around a constant speed of 1500 ± 50 r/min. The throttling for speed control and air and fuel proportioning was achieved using manually operated valves in the case of engine E1. However, in the case of engines E2 and E3, it was achieved using the gas carburettor and electronic/hydraulic governor respectively. Engine E1 was tested at varying CRs of 17, 14, 13.5, and 11.5; however, engines E2 and E3 were tested at CR = 12 and 10 respectively. The engines were tested at different ignition timing settings to determine the optimum ignition timing, referred to as the MBT (minimum advance for best torque), at different CRs. Measurements were made with respect to the power output (voltage and current), air and input fuel gas flow and exhaust emissions (CO and NO) [**18**, **19**]. A particulate measurement was not envisaged since the input feed was gas with low particulate matter. For instance, in the case of engine E1, particulate averaged about 60 mg/N m^3, which would amount to less than 25 mg/N m^3 (some particulate matter would burn) in the exhaust (with an air–fuel gas ratio of approximately 1.3). The in-cylinder pressure data with a resolution of 1° crank angle (CA) was acquired on a computer for engines E1 and E2.

6 PERFORMANCE

The performance of the three engines in terms of power output, energy balance, and emissions are discussed in the following sections in a sequence. The findings obtained from the investigations [**18**, **19**] of engine E1 were used in the adaptation and testing of engines E2 and E3. Further, engine E1 was subjected to reliability tests by continuously operating it for 100 hours; similarly, engine E3 was subjected to two trials of 24 hours duration (non-stop) each and the performance was assessed. The engine operations were found to be satisfactory. Examination of the interior components of the engine, the piston and cylinder, revealed that combustion was complete, with carbon deposits much lower than for diesel fuel operations for a comparable duration run.

6.1 Power output

Summarizing the performance of the engines, pressure–crank angle (p–θ) data were acquired on engines E1 and E2 in order to establish whether knock occurs with producer gas operation at varying CRs. Apart from this it was also used for identifying MBT. The outcome of these tests was that engine E1 worked smoothly without any sign of knock at CR = 17 and similarly for engine E2 at CR = 12 in the turbocharger mode. There was no sign of audible knock during the entire load range. Moreover, the absence of knock is clear from the p–θ data, which do not show any pressure oscillations (based on a number of individual cycles), either at part load or at full load (wide-open throttle) conditions. The p–θ diagrams for engine E1 at various CR around the optimum ignition timing are shown in Fig. 4; similarly, the data at a fixed CR = 12 for engine E2 are shown in Fig. 5.

The network delivered over a complete cycle can be found by integrating the pressure–volume (p–v) data over the four processes. This also helped in identifying the optimum ignition timing for a given CR, commonly referred to as MBT. It is well identified in the literature [**20**, **21**] that MBT corresponds to a value wherein the peak cylinder pressure should occur at 16–17° ATC (after top centre). The net indicated mean effective pressure (i.m.e.p.) obtained from the integrated p–v data is a measure of effectiveness with which an engine of a given volumetric displacement converts the input energy into useful work. The i.m.e.p obtained from ensemble-averaged p–v data ($\sim$30 cycles) in the case of engine E1 at varying CR as a function of ignition timing (IGN) is shown in Fig. 6. At CR = 17, the maximum i.m.e.p. recorded is 5.98 bar, corresponding to an ignition timing of 6° CA, which is declined to 4.85 bar with an ignition timing of 15° CA at CR = 11.5. These values are obtained at $\Phi = 1.08 \pm 0.2$ and fall within the anticipated value of $\Phi = 1.0$–1.1 [**20**]. It is also evident from the plot that variations in the i.m.e.p. values are modest between ignition timings of 6 and 12° CA corresponding to CR = 17.

The coefficient of variation (COV) of the i.m.e.p. at all CRs and ignition settings occurred well within 3–3.5 per cent, implying low cycle-to-cycle variations. The reason for low cyclic variation is the faster rate of combustion [**19**, **22**] occurring inside the engine cylinder. The faster rate of combustion is attributed to higher flame speeds due to the presence of hydrogen in the gas and also to the combustion chamber design. Exploring further the p–θ data, the peak pressure and the point of occurrence at ignition timings close to MBT are listed in Table 3. The measurements made on the engines are accurate to within −1.0° CA [due to possible lag in the signal and error in TC (top centre) identification, as mentioned earlier]. In the case of engine E1, it is evident from the data that peak pressure seemed to occur between 17 and 19° ATC at all CRs. In the case of CR = 13.5, the peak pressure seemed to occur at the optimum value (17° ATC) identified in the literature [**20**, **21**]. In the case of CR = 11.5, the peak pressure occurred at 17 and 12° ATC for an ignition timing of 15 and 17° BTC respectively. The difference in the i.m.e.p. between

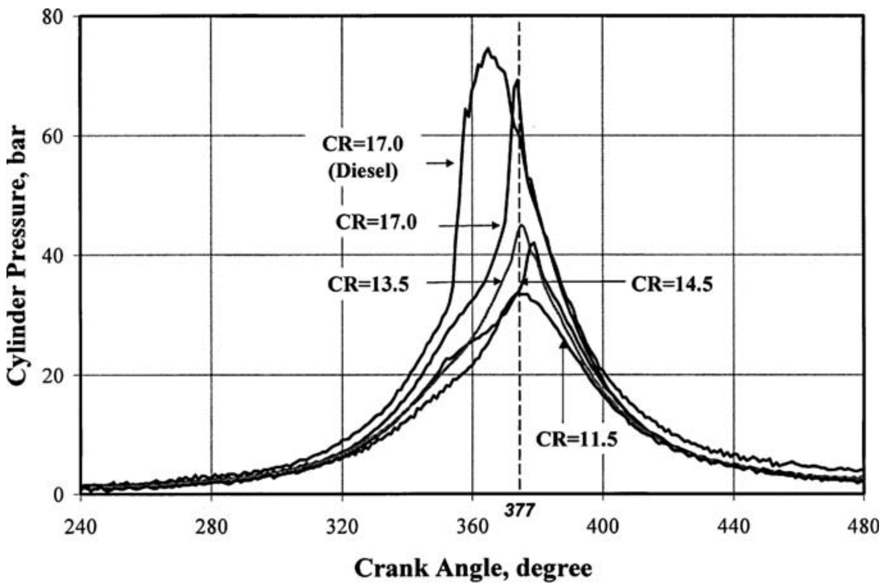

Fig. 4 Comparison of p–θ curves at different CRs. The ignition timing is at MBT or close to MBT (within MBT $+2^\circ$ CA). The p–θ curves correspond to ignition settings of 10, 10, 14 and 15° BTC for CRs of 17, 14.5, 13.5, and 11.5 respectively. Operation is in the diesel mode at 90 per cent of rated load (at an optimum injection timing of 34° BTC). All are ensemble-averaged data over 30 consecutive cycles. The net brake power and Φ for these cycles are shown in Table 5

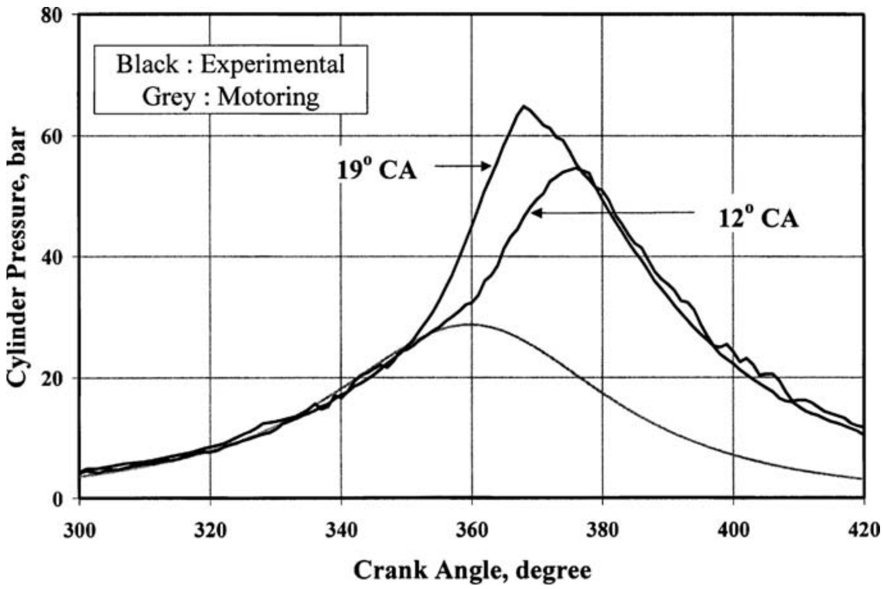

Fig. 5 A p–θ recording at varying ignition advances at 80 per cent of the achieved maximum output with producer gas. The ensemble-averaged data are over 30 consecutive cycles

the two ignition timings was found to be 3 per cent. However, for CRs of 17 and 14.5, the ignition timing identified in Table 3 seemed to marginally deviate from the optimum value. The variation of i.m.e.p. within this close range would be marginal as it is acknowledged that the relative torque delivered has a flatter characteristic around MBT [**20**]. Similarly, the MBT for engine E2 is found to be at 12° BTC, as shown in Table 3. The understanding obtained on MBT from engines E1 and E2 was used in identifying the MBT for engine E3, since p–θ data were not acquired for this particular engine.

The net brake output at varying ignition timing for four different CRs on engine E1 is shown in Table 4. It is evident from the data that ignition timing had to be retarded with the increase in CR in order to

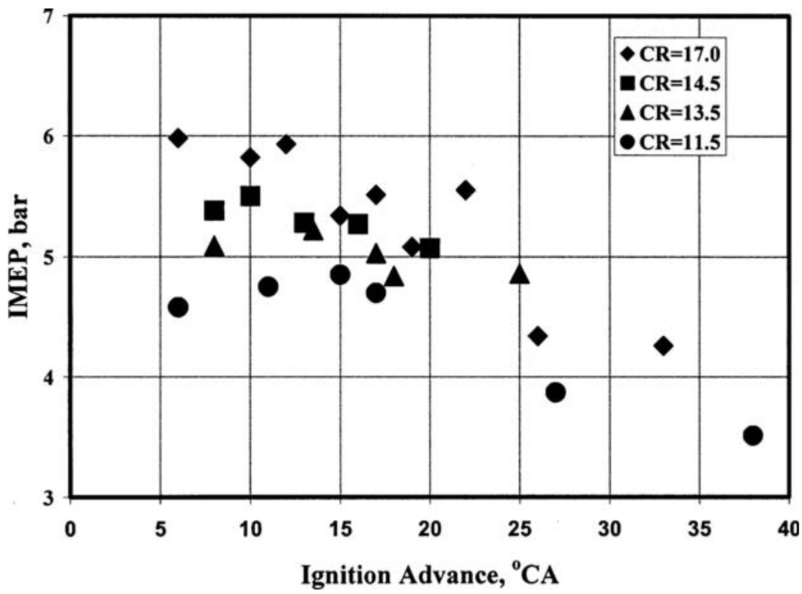

Fig. 6 Variation of i.m.e.p. (net) for engine E1 with ignition advance at various CRs

Table 3 Cylinder peak pressures and their occurrence close to MBT

Engine	CR	Ignition advance (°C)	Peak pressure (bar)	Occurrence (°ATC)
E1	17.0	6	55	20
	14.5	10	43.3	19
	13.5	14	45.0	17
	11.5	15, 17	33, 38	17, 12
E2	12.0	12	55	16

obtain higher output. This is because the thermodynamic conditions in terms of pressure and temperature are more severe at higher CRs and thereof the combustion is faster, thus calling for the optimum ignition timing to be located close to TC. The maximum output was recorded at an ignition advance of 6° BTC at CR = 17 and increased to about 15–17° BTC at CR = 11.5. At intermediate CRs of 14.5 and 13.5 the ignition advance was 10 and 14° BTC respectively. The fuel–air equivalence ratio was about 1.06 ± 0.5 in most of the cases, with efficiencies η of 30.7 and 27.5 per cent corresponding to maximum output at higher and lower CRs respectively. An isolated case of efficiency at 31 per cent was seen to correspond to an ignition setting of 12° CA, probably due to relatively leaner operation. In the data presented, the air–fuel ratio was tuned from the viewpoint of deriving maximum output and therefore the efficiency figures are necessarily not the maximum that can be obtained.

The result of the net brake output with three engines at MBT is shown in Table 5. At CR = 17, the engine delivered a maximum net brake output

Table 4 Maximum net engine output on engine E1 as a function of ignition timing and CR

IGN (BTC)	Φ	BP_{net} (kW)*	η: gas-to-shaft (%)	IGN (BTC)	Φ	BP_{net} (kW)*	η: gas-to-shaft (%)
CR = 17.0				CR = 14.5			
06	1.10	20.0	30.8	08	1.20	18.6	25.0
12	1.00	19.8	31.0	10	1.10	18.8	29.0
17	1.09	18.4	29.0	16	1.11	17.9	27.5
22	1.03	17.9	28.0	20	1.11	17.7	27.2
26	1.10	16.2	25.3				
33	1.25	14.0	19.0				
CR = 13.5				CR = 11.5			
08	1.05	18.2	28.6	06	1.07	17.0	27.0
14	1.06	18.6	29.0	15, 17	1.07	17.6	27.5
18	1.07	17.0	27.8	27	1.09	15.6	25.5
25	1.06	17.0	28.0	38	1.07	13.3	20.0

*Excluding radiator fan power.

Table 5 Maximum net engine output on different engines

Engine	CR	IGN (BTC)	Φ	Net electrical power (kW_e)	Net brake power (BP_{Net}) (kW)	Mixture energy density (MJ/kg)	η: gas-to-shaft (%)
E1	17.0	06	1.10	17.5	20.0	2.20	30.7
	14.5	10	1.10	16.4	18.8	2.20	29.0
	13.5	14	1.06	16.2	18.6	2.10	29.3
	11.5	15, 17	1.07	15.3	17.6	2.20	27.5
E2	12	12,14	0.94	165	182	1.90	28.3
E3	10	22,24	1.01	55	60	2.15	27.4

of 20 kW (17.5 kW_e) at an efficiency of 30.7 per cent compared to 24 kW (21 kW_e) brake output at 33 per cent efficiency with diesel (in the compression ignition mode). The efficiency calculation is based on the ratio of net brake output to energy content of the air and gas mixture. The useful output and efficiency decreased by lowering the CR. A maximum net brake output of 17.6 kW (15.3 kW_e) at an efficiency of 27.5 per cent was obtained at CR = 11.5. The power output at intermediate CRs of 14.5 and 13.5 were 18.8 and 18.6 kW respectively, with efficiencies of around 29 per cent. The efficiency at CR = 13.5 was comparable to that at 14.5, probably due to relatively leaner operation. The extent of de-rating in brake power was about 16.7 per cent at CR = 17 and increased to as high as 26 per cent at CR = 11.5 compared with baseline operations in the diesel mode.

In the case of engine E2 at a fixed CR of 12 a maximum net brake output of 182 kW (excluding 12 kW consumed by the radiator fan) was recorded with an ignition advance between 12 and 14° CA at $\Phi = 0.94$, against an MBT of 28° CA with diluted natural gas [**19**]. The value of Φ was lower in the current case because of limitations coming from the gasification system. In fact, the gas composition in terms of combustibles deteriorated with an increased supply of the gas to the engine. This therefore limited the input energy to the engine. The maximum net brake output was obtained at an ignition advance between 12 and 14° CA with gas-to-shaft efficiency being 28.3 per cent. The point to be noted here is the optimum timing; the maximum power output is obtained at slightly retarded ignition timing as compared to engine E1 at a comparable CR. This could probably be due to faster combustion due to higher turbulence (the mean speed of the piston is 7.0 m/s against 5.8 m/s in engine E1) and higher cylinder pressure and temperature due to turbocharging. In the case of engine E3, a maximum output of 60 kW was obtained at an ignition advance of 22–24° BTC against an MBT of 35° CA with pure natural gas [**23**]. The advancement in ignition timing as compared to engines E1 and E2 could mostly be related to the combustion chamber design and to some extent due to reduction in the CR.

6.2 Energy balance

Figure 7 represents the overall energy balance at CR = 17. The energy balance is based on gross brake power output. The gross brake output is the sum of net shaft output and power consumed by engine accessories (water pump/fan, dynamo, and fuel injection pump (FIP) = 1.4 kW). The energy balance in the gas mode corresponding to maximum brake output (at 6° CA) showed a useful output (gross brake power) of 32.9 per cent, about 30 per cent is lost through exhaust (sensible and chemical enthalpy of CO), and the remaining 37 per cent is lost to cooling water (inclusive of frictional and radiative losses). Figure 7 also compares the energy balance in the gas and diesel modes (at a rated output of 24 kW) at CR = 17. The energy loss to the coolant and miscellaneous is about 37 per cent compared to 30 per cent in diesel, and the energy loss through the exhaust is lower by about 5 per cent in gas mode. Overall the brake thermal efficiency is lower by about 1.5 per cent in gas. The energy balance as a function of the CR is shown in Fig. 8. There is an increase in energy loss through exhaust with the reduction in the CR, whereas the loss through the coolant is higher at a higher CR. The increased amount of heat loss to the cooling water in gas operations is attributed to engine combustion chamber design. Heywood [**20**] indicates that engine geometries such as the bowl-in-piston would experience 10 per cent higher heat transfer. The heat transfer to the coolant in the current case falls well within this range (7–10 per cent). The influence of engine geometry on heat loss could be more in the gas mode compared to diesel because of the basic difference in the nature of combustion. In the case of the compression ignition engine, combustion is heterogeneous and essentially occurs at

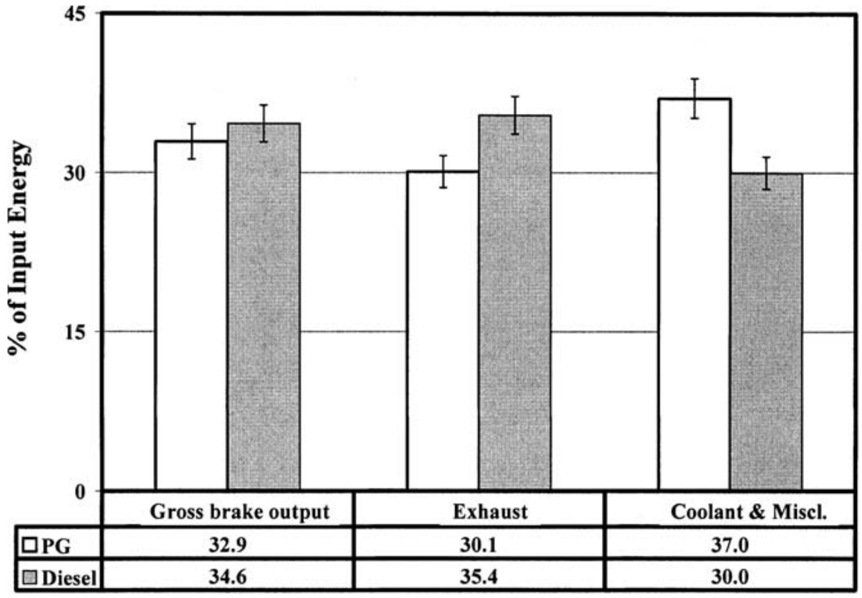

Fig. 7 Energy balance comparison for engine E1 in the diesel and producer gas modes at maximum brake output. The marker refers to the error band

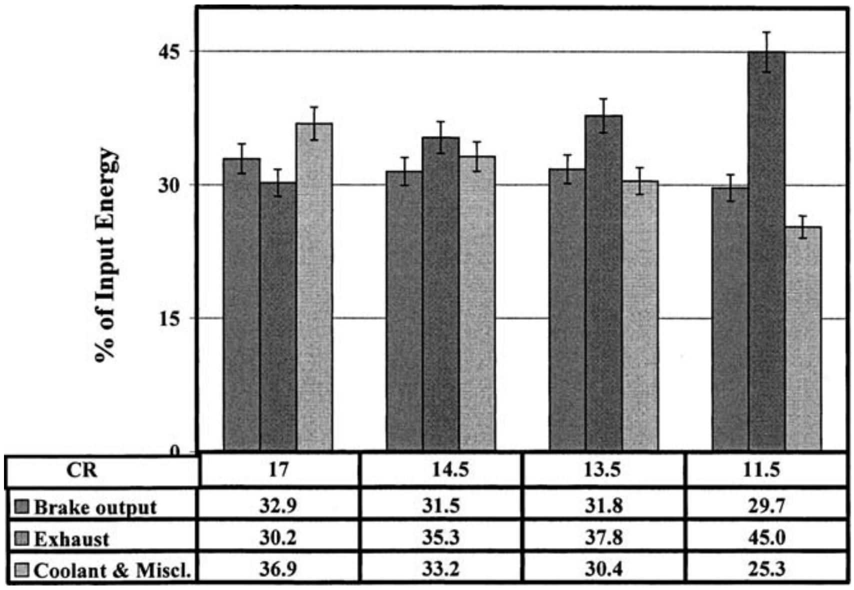

Fig. 8 Comparison of energy balance for engine E1 at various CRs with the producer gas corresponding to the maximum brake output. The marker refers to the error band

multiple ignition sites in a diffusion mode. Therefore, there is no definite flame front propagation and combustion does not occur close to the wall, unlike that in a spark ignition engine. This is one of the prime reasons for increased heat loss to the coolant in the spark ignition engine, which is the case in the current study in the gas mode. This increased heat loss to the coolant in the gas mode was leading to overheating of the engine within about 30 minutes of operation, but was overcome subsequently by increasing the cooling fan capacity by about 0.5 kW (incremental power measured by external motoring), thereby reducing the net peak output by the same value.

The energy balance of engines E2 and E3 at respective MBTs was more or less alike. The engine E2 showed that 30.2 per cent was realized as useful output and 28 and 42 per cent were lost to exhaust and coolant respectively. Moreover, there was no problem of coolant water becoming overheated because the basic cooling system adopted for these engines is identical to that of the diesel engine frame (at the same swept volume) and therefore designed to handle larger thermal dissipation.

6.3 Emissions

The emissions with producer gas operation on engines E1 and E2 are compared against existing emission standards of various countries in Table 6. The standard given for Indian conditions corresponds to that of the diesel-powered vehicle (Euro I) for gross vehicle weight greater than 3.5 tons (http://terin.org/urban/standard.htm). As stated earlier, there are no standards existing for a stationary engine (<2 MW). A suggestion made by the Indian Diesel Engine Manufacturers Association (http://www.kirloskar.com/html/sw/emissions) is pending waiting for approval from the Central Pollution Control Board (CPCB). These are the figures given in parentheses in Table 6 under the India column. The emissions with producer gas operation correspond to those measured under steady state conditions, using pre-calibrated instruments. However, the standards of various countries correspond to a specific procedure (the steady state test cycle) meant for commercial engines. Therefore, the exact procedure might not have been followed in the current study, but measurements were made under steady state conditions.

It can be seen that NO emission with producer gas is lower than all the existing norms. The prime reason for this to be lower is due to lower peak cylinder temperatures and also lower residence times in the combustion chamber as the MBT is located close to TC. The CO results with engine E2 are encouraging; however, there are large deviations with respect to engine E1 results. Therefore, treatment of exhaust in terms of CO is mandatory from the viewpoint of deriving maximum output ($\Phi > 1.0$). This could be true even with respect to HC emissions. However, Particular Matter (PM) is expected to be low, even though measurements were not done because the input feed is gas with PM matter less than 2 mg/N m^3 (with engine E2 experiments), which amounts to <0.5 mg/MJ. In the case of engine E1 experiments PM is estimated to be less than 14 mg/MJ, with input gas containing PM to the extent of 60 mg/N m^3. In the case of engine E3, the particulate content should be much lower as the input feed was found to contain PM of the order of 0.02 mg/N m^3. The emission recorded on engine E3 during the 24 hour long duration operation is shown in Fig. 9, at a constant load of 52–54 kW$_e$. During the initial few hours of operation CO was found to be higher and is related to the tuning of the carburettor in arriving at the correct air–fuel ratio. Subsequently, however, the values become much lower than most of the existing emission norms, which implies environmental friendly operation with producer gas fuel.

Table 6 Comparison of emissions (g/MJ) with producer gas operation against existing emission norms in various countries. (From http://app10.internet.gov./scripts/nea/cms/htdocs/article.asp)

	Country			
Parameter	USA	EU	Japan	India
CO	3.06	1.4–1.8	1.67	1.25 (3.9)
NO_x	2.56	2.56	2.6–3.06	2.22 (5.0)
HC	0.36	0.36	0.4–0.56	0.3 (0.98)
PM*	0.15	0.15–0.24	—	0.1–0.2 (<3.5 Bosch)
Engine E1 results between 6 and 20° CA for all CRs (minimum and maximum values) at $\Phi = 1.0$–1.2				
Parameter	17.0	14.5	13.5	11.5
CO	1.1–11.0	11.0–15.0	4.0–16.0	9.0–14.0
NO_x	0.03–0.28	0.02–0.22	0.03–0.20	0.05
PM			<0.014	
Engine E2 results between 12 and 24° CA for CR = 12.0 at $\Phi = 0.94$–0.97				
CO			0.58–1.2	
NO_x			0.32–0.7	
PM			<0.0005	
Engine E3 results between 22 and 24° CA for CR = 10.0 at $\Phi = 1.01$–1.03				
CO			0.4–1.8	
NO_x			0.2–0.7	
PM			$\ll 0.0005$	

*PM, particulate matter. This excludes engine oil, which is typically in the range of 0.003–0.1 g/MJ [**20**] in large diesel engines.

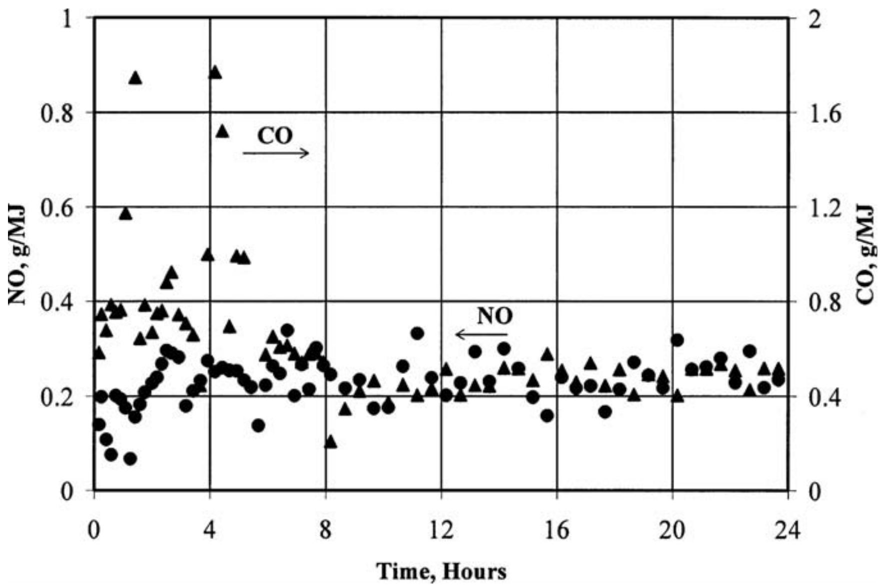

Fig. 9 Variation of emission in the case of engine E3 at 90 per cent load during 24 hours of continuous operation

7 OBSERVATIONS

Summarizing the overall development effort, performance of the engine at a higher CR is smooth and it has been established that operating engines using producer gas in the SI mode at a CR of up to 17 is feasible. This is obvious from the p–θ curve, which shows a smooth rise in pressure without any presure oscillations. A shorter duration of combustion has been observed with producer gas fuel, requiring retardation of the ignition timing to achieve MBT. These faster burning cycles are corroborated by low cyclic pressure fluctuations with a coefficient of variation of about 3 per cent. The faster burning process has been identified to be due to the higher flame speed of the fuel/air mixture, which is attributed to the hydrogen content in the gas. The MBT found from this study is much retarded when compared with those stated by earlier researchers. The MBT in the current case is in the range between 6 and 22° CA for a CR range between 17 and 10 against 30–45° CA (for a CR of 11.5 and below) stated by earlier researchers. This change in ignition advance in the present study can only be attributed to the improved producer gas composition. The hydrogen content in the present case is about 18–20 per cent against 11–12 per cent stated by Parke *et al.* [**7**] and Parke and Clark [**8**] and 10 per cent (theoretical) by Martin and Wauters [**5**]. However, it is difficult to comment on the retarded MBT of 35° CA stated by Shashikanta and Parikh [**10**] with a hydrogen content of 24 per cent. The MBT with producer gas is much retarded when compared to engines fuelled with natural gas; this is evident from the results on engines E2 and E3, which is due to a higher laminar flame speed with producer gas.

The de-rating in power output with producer gas fuel for the three engines is summarized in Table 7. This de-rating is due to the resultant effect of reduction in the mixture energy density and the product–reactant mole ratio as discussed in Table 1. In the case of engine E1 at CR = 17, the de-rating is about 16.7 per cent and increases to about 19 per cent in the event of enhancing the engine cooling system for practical field operations. The extent of de-rating is much lower when compared to any of the previous studies [**5**, **7**, **8**, **12**]. This value at CR = 17 matches a similar kind of de-rating reported for

Table 7 Summary of producer gas engines

Engine	CR	Net rating (kW)	Achieved output (kW)	De-rating (%)	Remarks
E1	17.0	24	20	16.7	
E1			19.5	19.0	Increased cooling capacity
E2	12.0	258 on diluted natural gas (75% CH_4)	182	30.0	
E2	12.0		202	22.0	With 10% increase in LCV
E3	10.0	84	60	28.5	

natural gas operation by Das and Watson [**17**]. In the case of engine E2, the de-rating is about 30 per cent compared with operations on an identical engine using diluted natural gas. The data with diluted natural gas (biogas) have been recorded on a field system comprising a Greaves engine (identical to engine E2) at UGAR Sugars Limited, Belgaum, Karnataka, India. This de-rating appears to be higher when compared with the results of engine E1 (26 per cent at CR = 11.5). However, as indicated in Table 5, the mixture density in the experiments on engine E2 was about 1.9 MJ/kg, which is about 10 per cent lower than what has been measured on a similar class of gasifier. If an increment of 10 per cent in the mixture density is considered (which is actually so with respect to the tests on E1—Table 2) the de-rating is reduced to about 22 per cent at the expected output of 202 kW with producer gas [with a lower calorific value (LCV) of approximately 4.8 MJ/kg]. In the case of engine E3, the de-rating is to the extent of 28.5 per cent, with the basic rating of the engine at 84 kW on pure natural gas. In reality, it is possible that the difference in de-rating for engines E2 and E3 is marginal, if true rating for the engines is considered using natural gas. This loss in power to some extent can be lowered by working at a higher CR in the case of a naturally aspirated engine and by addressing issues related to compressor–turbine matching in the case of a turbocharged engine. For instance, the incremental gain in power per unit CR is about 2.2–2.5 per cent [**18**, **20**]; therefore, adopting a CR of 15–17 as against CR = 10 implies a higher output of 13–18 per cent.

The emission in terms of NO is found to be much lower than the stipulated emission norms of the CPCB and the Swiss. However, the CO levels are found to be higher in the case of engine E1, but meet the emission norms in the case of engine E2 by using the indigenously built producer gas carburettor. These observations are consistent with the results of Giordano [**24**] on a producer gas engine powered with an IISc gasifier. Lastly, the information that is relevant to the biomass gasifier coupled to a gas engine is the specific fuel (biomass) consumption. The specific biomass works out to about 1.00–1.20 kg per kW h electric energy generated, which corresponds to an overall efficiency (biomass to electricity) of 24–20 per cent. Currently the tested systems are being monitored as field installations at industrial establishments.

Based on the above study, a short range of optimum ignition timing is identified as shown in Table 8 for producer gas fuel. This could nevertheless vary depending upon the actual fuel-gas composition,

Table 8 Optimum ignition timing (MBT) at varying CR

Range in CR	Ignition timing (° BTC)
17	6–10
14–15	10–12
12–13	12–14
11–12	15–17
10–11	22–24
8–9	26–28

combustion chamber design, and engine speed. However, these data can be used as initial indicators and further tuned for best performance based on the actual engine configuration and fuel-gas composition.

8 CONCLUSION

The developmental studies on the producer gas engine reveal that smooth operation is possible from the highest CR of 17, with varying de-rating identified at lower CRs. It is shown from the above sets of trials that it is possible to operate commercially available gas engines (meant for natural gas, etc.) on low-energy density producer gas by employing a suitably designed gas carburettor. This study therefore paves the way for the possibility of adapting a commercially available gas engine for a large-scale power generation application, albeit with a loss of power to an extent of 20–30 per cent. This loss in power is compensated to a much larger extent as these technologies generate smaller amounts of toxic gases (low NO_x and almost zero SO_x), being zero for greenhouse gas emissions.

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APPENDIX

Notation

ABC	after bottom centre
ATC	after top centre
b.s.f.c.	brake specific fuel consumption
BBC	before bottom centre
BC	bottom centre
BP	brake power
BTC	before top centre
CA	crank angle
CFR	cooperative fuel research
CGPL	Combustion, Gasification and Propulsion Laboratory
CI	compression ignition
COV	coefficient at variation
CPCB	Central Pollution Control Board
CR	compression ratio
GHG	greenhouse gas
i.m.e.p.	indicated mean effective pressure

IGN	ignition timing
IISc	Indian Institute of Science
INJ	injection timing
IP	indicated power
LCV	lower calorific value
MBT	minimum advance for best torque
NG	natural gas
p–v	pressure–volume
p–θ	pressure–crank angle
PG	producer gas
PM	particulate matter
RG	recycled gas
S_L	laminar burning velocity
SI	spark ignition
SERI	Solar Energy Research Institute
TC	top centre
Φ	fuel–air equivalence ratio