EXPERIMENTAL STUDIES ON MULTI-CYLINDER NATURAL GAS ENGINE FUELED WITH PRODUCER GAS

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ABSTRACT: Biomass gasification- a thermo-chemical conversion process, with near zero net carbon emission has a significant potential to displace fossil fuel use especially in various modes of power generation. The current study presents the results from investigations carried out on a six-cylinder natural gas engine rated at 55.0 kW fueled using producer gas as a fuel. The peak load on the engine was 29 kW at 24 Deg BTDC ignition setting. At 29 kW, the pressure trace yielded a peak pressure of 35 atmospheres and the indicated power was 42 kW. A specific biomass consumption of 1.35 kg/kWh of electricity was recorded with brake thermal efficiency of 23% and an overall biomass to electricity efficiency of 18%. Measured exhaust composition without a catalytic converter revealed 0.22 g/MJ NO_x, 0.017 g/MJ HC meeting the international specifications for stationary power generation systems, while CO was found to be 4.9 g/MJ which is higher than the norm.
Keywords: alternative fuel, biomass, gasification, internal combustion engine, thermo chemical conversion

1 INTRODUCTION

Over the past few decades, the world is witnessing a general upward long-term trend in terms of energy consumption and net CO_2 emission along with other pollutants. The demand for energy is primarily satiated by fossil fuels that remain the dominant source of energy.

A conservative estimate projects an annual increase of 1.5% in the world energy demand over the next few decades with coal seeing the biggest increase in demand followed by oil and gas. The primary driver for coal is to meet the projected electricity demand increase at a rate of 2.5% per annum thus requiring an addition of nearly 5000 GW globally over the next two decades [1].

Considering the various challenges and constraints pertaining to energy demand and consequential emissions influencing climate changes, technological solutions that cater to the energy demand while reducing emissions are desired. In this direction, gaseous fuels are gaining prominence for power generation using internal combustion engines primarily due to easy adaptablility and efficient combustion with minimal exhaust emissions [2]. Producer gas obtained from the thermo-chemical conversion of biomass is gaining importance in energy starved nations where it can meet national goals of providing electricity by grid linked independent power generation and decentralised power generation. The principal advantage of producer gas apart from being a gaseous fuel is that, it is also identified as nearly carbon neutral technology.

The present work focuses on evaluating the performance of a spark ignitied multi cylinder engine fired with producer gas under naturally aspirated conditions.

2 PRODUCER GAS AS A FUEL FOR SI ENGINES

The typical composition of producer gas as reported in literature is 18-20% each of H_2 and CO, 2% CH_4 and, rest inerts like CO_2 and N_2 [3-4]. The lower calorific value (LCV) varies between 4.5 – 4.9 MJ/kg, with stoichiometric air-to-fuel ratio being 1.35 + 0.05 on mass basis. Table I lists some of the important properties of producer gas in comparison with CH_4. Since a natural gas engine is being fired with producer gas, the properties of CH_4 are also listed alongside that of producer gas [3-5][6].

Table I: Properties of Producer Gas and Methane

Property ▼ \ Fuel ►	PG	CH_4
LCV (MJ/kg)	05.0	50.2
Stoichiometric A/F(kg/kg)	1.35	17.2
Mixture CV (MJ/kg)	2.12	2.75
Product/Reactant (Mole/Mole)	0.87	01.0
Stoic flame speed (m/s)	0.50	0.35

The data in Table I indicates that the calorific value of producer gas is almost 10 times lower than that of CH_4. However, in case of combustion systems the calorific value of the fuel cannot be looked at in isolation but the mixture calorific value needs to be accounted for.

On this basis, at stoichiometric conditions, producer gas – air mixture has a calorific value of 2.12 MJ/kg of mixture while Methane – air mixture has a lower calorific value of 2.75 MJ/kg. Thus, for the same quantity of mixture flow rate, the engine would suffer a minimum de-rating of 23% on account of lower calorific value only. A further look at the table also indicates that the product to reactant mole ratio is less than unity for producer gas leading to lower cylinder pressures which again causes some loss of power. Thus, on operating a natural gas engine using producer gas, de-rating occurs because of the fuel properties. This aspect has been brought out in greater details by Sridhar et al [2] and Dasappa [6].

Parameters that are to be optimized for operating the engine on producer gas owing to the higher flame speed as compared to natural gas (due to the presence of H_2) are the ignition timing for maximum brake torque (MBT) timing [7] and the carburetion system [2][6].

Producer gas is generated from an open top, twin air entry, re-burn gasifier developed at Combustion, Gasification and Propulsion Laboratory (CGPL) of Indian Institute of Science (IISc). The gasifier is unique in terms of generating superior quality producer gas well suited for engine operations. The details pertaining to the suitability of the mentioned gasification system for engine operation has been discussed in greater depth by Mukunda et al [8], Sharan et al [9] and Hasler[10].

3 EXPERIMENTAL SETUP AND PROCEDURE

Experiments were conducted on a six cylinder Cummins India Limited (CIL) make natural gas engine operated under naturally aspirated conditions. The specifications of the engine are as described in Table II.

Table II: Specifications

Make and model	CIL 6B59NA
Number of cylinders	6
Bore X Stroke	0.102 m X 0.120 m
Compression ratio	10.5
Ignition system	Spark Ignited
Engine type	Four Stroke
Aspiration	Natural
Rating	55 kW @ 1500 rpm
Alternator	3 Phase, 50 kVA

The engine, designed to operate on natural gas had its intake modified to accommodate the high fuel flow rates since air and fuel are supplied in near equal proportions when operating on producer gas [2]. The necessary modifications for the engine has been reported by Sridhar et al [2] and Dasappa et al [6].The engine, through the alternator is connected to an electric loading panel with resistive loading. The engine is operated from no load to full supported load with the full supported load being identified as the maximum load beyond which the alternator frequency drops off from 50 Hz. At each load, various parameters are measured as described below.

3.1 Pressure measurement

The in-cylinder pressure is measured using piezo-electric differential pressure transducer in an un-cooled spark plug of AVL make. The sensor through a charge amplifier is connected to an eight channel data acquisition unit having peak individual channel frequency of 800 MHz. The primary advantage of such a sensor is that no modifications to the cylinder head is necessary since the sensor is adapted into the spark plug itself as shown in figure 1.

Figure 1: Spark plug adapted pressure sensor

The pressure transducer is connected to a data acquisition unit through a charge amplifier. The data acquisition unit has capacity to acquire data at the rate of 800 MHz per channel thus allowing for recording any transient traces. Since the pressure sensor is a differential sensor, it can detect only the change in the pressure content and not the absolute value. Towards converting the differential pressure into absolute value, the thermodynamic zero line correction as suggested by Hohenberg [11] is adopted wherein the assumption of constant polytrophic coefficient is invoked for a certain range of compression crank angle.

3.2 Producer gas composition

The producer gas composition is continuously monitored for the entire duration of the experiment using SIC MAIHAK gas analyzer. The analyzer gives the gas composition on volume fraction basis and the composition in terms of CO, CO_2, H_2 and O_2 can be monitored. The gas composition is monitored online for the entire duration of testing.

3.3 Exhaust gas composition

The exhaust gas composition is measured using Quintox Flue Gas Analyzer. The analyzer gives the composition of various entities on dry basis in % volume / ppm levels.

3.4 Temperature and Flow

Temperature of mixture, flue gas, inlet and exit water temperature for the water jacket are all measured using a simple K type thermocouple. Air and gas flow measurement were carried out using calibrated venturimeter connected to differential pressure sensor.

Experimental investigation on the engine involved a preparatory step to operate the gasification system and allowing it to stabilize so as to deliver consistent quality gas. Typically, about an hour is the time required for the gasifier to attain steady state of condition from a fresh cold start. At any operating condition, the peak supported load was considered as that load beyond which the engine speed was dropping leading to a drop in the electricity frequency. The first sets of experiments were to determine the ignition timing that delivered best torque i.e. the maximum brake torque (MBT) timing. This was necessary since the properties of producer gas are different from that of natural gas for which the engine is tuned. Towards this, the engine was operated from no load to maximum supported load in steps of approximately 5kW for six different ignition timings 15, 18, 20, 24, 28 and 32 Deg BTDC. All subsequent experiments were carried out with ignition set at the MBT timing. Experiments were carried out from no load to full load at the MBT timing with and without the radiator whereby the radiator based cooling system was replaced by an external cooling system wherein coolant flow parameters like flow rate and inlet and exit temperatures could be measured. Using these information a complete energy balance for the engine was carried out. During each of the experiments, measurements were made with respect to the power output as indicated on the loading panel as well as voltage and line current, air and input fuel gas flow, exhaust emissions and temperature. Cooling water flow rate along with inlet and exit temperature were also recorded. On the emission front, particulate measurement was not envisaged because the feed itself was gas with low particulate matter. This aspect has been brought out in discussion by Sridhar et al [5].

4 RESULTS AND DISCUSSIONS

This section presents the results from the experiments and the analysis of the data. All the pressure-volume and pressure-crank angle and heat release-crank angle diagrams presented are ensemble average values of 250 consecutive cycles. Any deviation is explicitly mentioned in the relevant discussion.

4.1 Gas Composition

Experiments on the 6B series engine were conducted over a cumulative time of around 30 hours spread over 5 days. The average gas composition obtained in volume percent was CO 16.04%, H_2 16.92%, CO_2 10.50%, CH_4 1.22%, and balance N_2. The average lower calorific value of the gas for this period comes to 4.2 MJ/m3. As can be observed, the gas quality seems to have suffered with a reduction in CO and H_2 content to the tune of around 3%.

4.2 Peak Supported Load

The engine supported a peak load of 29 kW at 1500 rpm. This essentially amounts to about 37% de-rating as compared to operation on natural gas. The de-rating is slightly on the higher side on comparing with the literature reported values [2]. This is mainly due to slight deterioration in the quality of the gas and the corresponding drop in the calorific value to 4.2 MJ/m^3 as against 4.5 MJ/m^3.

4.3 Maximum brake torque timing

Any engine, under a set of operating conditions has a particular ignition timing at which maximum torque is delivered. This timing is known as the Maximum Brake Torque (MBT) timing. The original ignition timing set for the engine operation on natural gas was 22 Deg before the top dead center (BTDC) as reported by Dasappa et al [12]. Owing to the difference in properties of producer gas, the MBT timing had to be re-established.

MBT was established by adopting the spark sweep test which is a well established procedure for determining the MBT timing [7][13-15]. In the spark sweep test, the engine is operated at various spark timings within a particular range and the peak supported load at each of the loads is recorded. In the present work, spark sweep spanned a total range of 17 Deg from 15 Deg BTDC to 32 Deg BTDC.

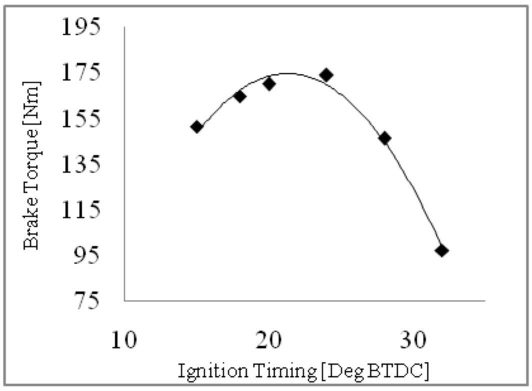

Figure 2. Brake torque variation with ignition timing

Amman at [16] and Hubbard et al [17] al have well established the fact that the brake torque peaks at the MBT timing and a plot of maximum brake torque plotted against ignition timing would give an indication of the MBT timing. This further confirmed by a latest work by Pipitone [18] who has reported on comparison of various MBT detecting techniques.

Figure 2 presents the variation of brake torque with ignition advance, bullets indicating experimental values and the line a representative cure fit. The trend is along the expected lines [7][16] with the brake torque initially increasing and subsequently decreasing on advancing the ignition angle. The top region where the brake torque peaks is quite flat and the top most point in this region can be identified as the ignition angle delivering maximum brake torque or MBT. Experiments have yielded 24 Deg BTDC as the ignition timing and call for advancing the ignition timing by 2 Deg from the original setting. A careful observation of the trend indicate that the point of inflection for the torque curve is between 20 and 24 Deg BTDC and as identified by Amman et al [16] and Heywood [7] advantage is drawn from this flat peak to set the ignition timing at slightly retarded position to avoid knock.

4.4 Indicated Power and Cylinder Pressure Trace

Figure 3 presents the variation of the pressure as a function of crank angle for various loads at the MBT timing.

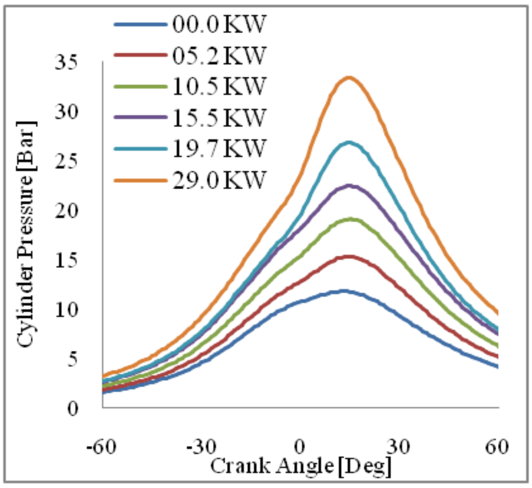

Figure 3: Pressure – crank angle trace at various loads

The pressure crank angle trace brings out one important factor with respect to the MBT timing. As per literature [7][13][19-21] if the peak pressure in the cylinder occurs at around 15 Deg after top dead center (ATDC) then the engine is set at MBT timing. The above figure and corresponding data give a position of peak pressure value of 16 Deg ATDC thereby confirming 24 Deg BTDC as the MBT ignition timing for the current operating condition and fuel. Plotting the pressure volume trace permits evaluation of indicated work as the area under the curve. Figure 4 presents the pressure volume curve that is used to determine the indicated work delivered by the engine at each of the loads. The area under the curve gives the work done by a single engine and symmetric behavior is assumed.

Table III presents the indicated and brake power along with the corresponding mean effective pressure values. The peak load IMEP at 5.64 bars is lower than the corresponding natural gas IMEP of 8.00 bars as reported by Dasappa et al [12]. Frictional losses in the range of around 11.5 kW are observed for the entire range giving a frictional mean effective pressure (FMEP) of around 1.5 bars which is consistent with the literature reported values for multi cylinder engines at 1500 rpm [22-25].

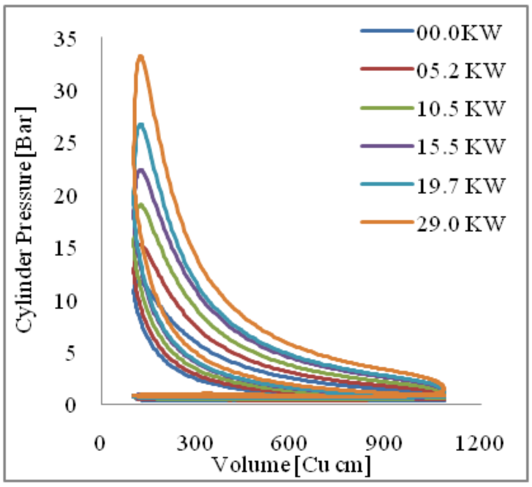

Figure 4: Pressure-Volume trace at various loads

Table III: Power and Mean effective pressures

BP (kW)	IP (kW)	BMEP (Bar)	IMEP (Bar)
00.0	11.04	0.00	1.50
05.2	16.19	0.71	2.20
10.5	22.07	1.43	3.00
15.5	27.96	2.11	3.80
19.7	32.37	2.68	4.40
29.0	41.20	3.77	5.64

4.5 Efficiency and Specific biomass consumption

Figure 5 presents the variation of brake and indicated thermal efficiency with load, dots indicate the experimental values while the lines are second order polynomial fits to represent the trend.

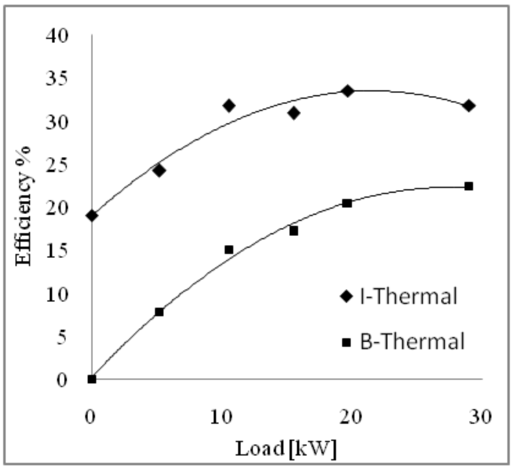

Figure 5: Indicated and brake thermal efficiency

A maximum brake thermal efficiency of 23% is observed amounting to 18% conversion efficiency from biomass to electricity with 80% cold gas efficiency for the gasifier [26].

The specific biomass consumption gives a more direct indication of the amount of fuel consumed per kWh of energy developed. Figure 6 presents the variation of indicated and brake specific biomass consumption with load. Specific biomass consumption trend indicates a value of 1.30±0.05 kg/kWh based on shaft power output. Literature reports SBC value close between 1 to 1.2 kg/kWh for engine systems with overall efficiency values of 30% [5][27-29].

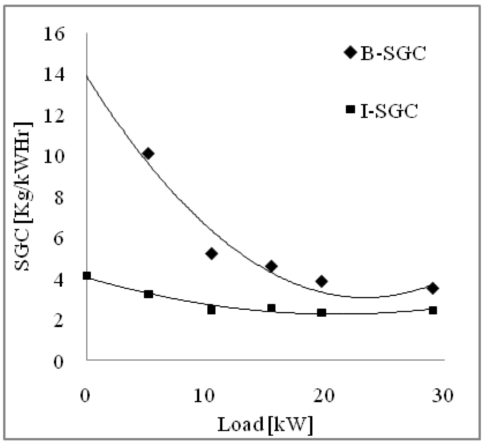

Figure 6: Indicated and Brake Specific Biomass Consumption variation with load

4.6 Energy Balance

Figure 7 presents the energy balance at the peak load of 29 kW.

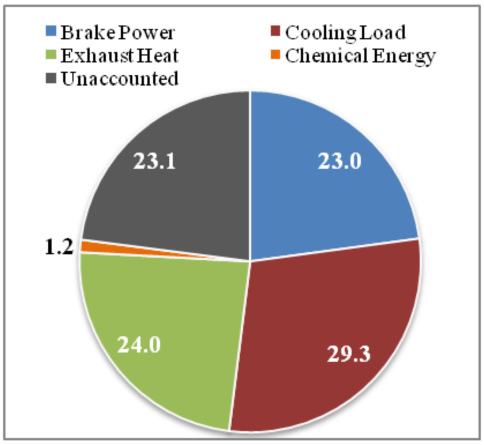

Figure 7: Heat balance at full load

As per literature, a typical spark ignited engine has the following energy distribution represented as percent of heat released in the cylinder; Brake power 25-28%, heat lost to the coolant 17-26%, heat carried away by the exhaust 36-50%, exhaust chemical enthalpy due to incomplete combustion 2-5% and miscellaneous or unaccounted portion around 3-10% [7][30-31]. Along similar lines, Sridhar et al [2] reports, on the performance of a diesel engine converted for producer gas operation at a compression ratio of 11.5 and ignition timing of 15 Deg BTDC distribution of energy as 29.7%, 45% and 25.3% for brake power, exhaust load and cooling + miscellaneous load respectively with the chemical enthalpy respectively. The recorded values in the present work indicate higher cooling and other loads but low exhaust energy. Heywood [7] has indicated that bowl-in-piston kind of engines could experience nearly 10% higher heat transfer and the current engine being a shallow bowl in piston type, has correspondingly resulted in higher cooling load. On the chemical energy in the exhaust, one of the possible reasons for lower value as compared to reported values may be due to the advanced ignition. With the ignition timing at 24 Deg BTDC, an advance by almost 9 Deg cause higher in-cylinder temperatures and lower exhaust temperatures. It is a well established fact that, advancing the ignition timing leads to higher in-cylinder temperatures and subsequently higher heat fluxes across the cylinder wall as brought out by Demuynck et al [32] and Shudo et al [33]. Thus the present observation is consistent with the results from literature. This argument is further strengthen by the fact that, NOX in the exhaust has been observed at around 0.22 g/MJ while in the work by Sridhar et al values close to 0.05 g/MJ have been reported suggesting higher peak cylinder temperatures in case of advanced ignition.

Higher in-cylinder temperatures also cause higher wall and engine component temperatures [32] which is reflected in enhanced other losses. The chemical enthalpy due to incomplete combustion is well within the reported value.

4.7 Emissions

Table IV presents the composition of exhaust emissions in terms of CO, HC and NO_X at full load and corresponding emission norms in India and Europe [34].

The indicated values are corrected for 5% oxygen on dry basis in the exhaust.

Table IV: Emissions at full load against norms

	NO_X [g/MJ]	HC [g/MJ]	CO [g/MJ]
Indian Norms			
	2.56	0.360	0.970
European Union Norms			
	1.30 (NO_X + HC)		1.390
Experimental Results			
	0.22	0.011	4.904

Guidelines for Safe and Eco-friendly Biomass Gasification [35] specifies a limit of 500 mg/m^3 for NO_X and 650 mg/m^3 for CO while the recorded values stand at 123.5 mg/m^3 and 2659.7 mg/m^3 respectively for NO_X and CO. Though NO_X is well within the specified limit, CO seems to be on the higher side. This could be attributed to partial or incomplete combustion at higher loads. This aspect needs further investigation towards identifying a suitable catalytic convertor.

5 EMISSION REDUCTION POTENTIAL IN INDIA

The primary advantage of gasification engine systems stems from the fact that, in comparison with conventional fossil fuels, biomass on a relative time scale can be treated as near carbon neutral [36]. The following calculations give an indication of the potential of gasifier – engine system towards mitigating CO_2 emission in the Indian context.

- Specific Biomass Consumption of gasifier – engine system is around 1.2 Kg/kWh.
- Specific fuel (Coal) Consumption of a thermal power plant in India is around 0.72 kg/kWh [37]
- Carbon Dioxide emission from gasifier – engine system is around 0.85 kg/kWh
- Carbon Dioxide emission from a coal based thermal power plant ranges from 0.7 to 1.8 kg/ kWh depending on the plant load factor and quality of Coal [37]
- Biomass availability in India is around 565 million tonnes per year. Surplus biomass resources (excluding animal feed, cooking, or other purposes) and available for power generation is about 189 million tonnes per year [38].
- 50% of the total available biomass for power generation is considered for use in the gasification – engine system owing to dispersed nature of the biomass.

On the basis of above considerations, gasifier – engine system can develop power to the extent of 800 GWh amounting to around 14% of power generation levels for year 2009 [39-40]. At these levels, the gasifier – engine system would emit around 75 million kilos of CO_2 annually while a coal fired thermal power plant would emit around 105 million kilos of CO_2 annually.

Even if sequestration of CO_2 by biomass is neglected, just displacement of coal provides annual emission savings of around 30 million kilos of CO_2. If 50% of the emitted CO_2 by the gasifier – engine system is absorbed during the growth of biomass either in natural vegetation or in energy plantation [41], annual CO_2 emission mitigation reaches around 65 million kilos amounting to around 5% of total annual CO_2 emissions in India and around 8.5% of CO_2 emissions due to power generation [42]. Thus, in the Indian context, adoption of gasification – engine system has a potential to reduce the net carbon emission to the tune of 5%.

6 CONCLUSIONS

The paper reports perfomance of a multi-cylinder natural gas engine fired with producer gas using incylinder investigations. The peak power obtained on the 1.083 x 6 = 6.5 litre cylinder is 29 kW. Peak load SBC evaluates to around 1.25+0.05 kg/kWh. Emission measurement from the engine exhuast suggests that Nox and HC are within the limits, while CO is higher. A look at the energy balance for the engine has brought out the fact that the energy distribution is comparable with literature reported values for gasoline fired engines while some difference is observed in case of similar work carried out earlier by Sridhar et al [5], with the difference originating due to the change in the ignition settings of nearly 10 Deg. On the emissions front, the producer gas fired engine fares very well on the NOx and HC front while the CO emissions are higher than the permissible limits. Investigations are underway to furter establish the cause and identify suitable catalytic convertor to handle CO emissions. On the statistics of CO_2 emissions, an assessment in the Indian context brings out the fact that close to 8.5 % of the CO_2 emitted owing to thermal and electrial power generation translating in absolute terms to around 65 million kilos can be displaced by adopting gasification – engine system wherever feasible.

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