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Biomass derived producer gas as a reciprocating engine fuel—an experimental analysis

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ABSTRACT

This paper uncovers some of the misconceptions associated with the usage of producer gas, a lower calorific gas as a reciprocating engine fuel. This paper particularly addresses the use of producer gas in reciprocating engines at high compression ratio (17 : 1), which hitherto had been restricted to lower compression ratio (up to 12 : 1). This restriction in compression ratio has been mainly attributed to the auto-ignition tendency of the fuel, which appears to be simply a matter of presumption rather than fact. The current work clearly indicates the breakdown of this compression ratio barrier and it is shown that the engine runs smoothly at compression ratio of 17 : 1 without any tendency of auto-ignition. Experiments have been conducted on multi-cylinder spark ignition engine modified from a production diesel engine at varying compression ratios from 11:5 : 1 to 17 : 1 by retaining the combustion chamber design. As expected, working at a higher compression ratio turned out to be more efficient and also yielded higher brake power. A maximum brake power of 17:5 kWe was obtained at an overall efficiency of 21% at the highest compression ratio. The maximum de-rating of power in gas mode was 16% as compared to the normal diesel mode of operation at comparable compression ratio, whereas, the overall efficiency declined by 32.5%. A careful analysis of energy balance revealed excess energy loss to the coolant due to the existing combustion chamber design. Addressing the combustion chamber design for producer gas fuel should form a part of future work in improving the overall efficiency.

Keywords: Biomass; Compression ratio; De-rating; Producer gas; Spark ignition engine

1. Introduction

With the renewed interest in biomass energy by necessity, biomass-based technologies are achieving prominence not only as rural energy devices but also as industrial power plants. Gasification is one such process where clean gas could be generated using a wide variety of bio-residues as the feed stock and in turn use the fuel gas for power generation purposes. These are being used in standard diesel engines in dual-fuel mode of operation so as to obtain diesel savings up to 85%. Operation of engines on gas alone has been explored in some limited sense by a number of researchers ever since World War II. In the

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present times, adopting these technologies has immense economic benefits, a route pursued by a number of researchers is in [1–5].

Nomenclature

ABDC	after bottom dead centre
ATDC	after top dead centre
BBDC	before bottom dead centre
BTDC	before top dead centre
CO	carbon monoxide
CR	compression ratio
IC	internal combustion
IGN	ignition
IISc	Indian Institute of Science
IMEP	indicated mean effective pressure
LPG	liquified petroleum gas
MBT	minimum advance for brake torque
NO	nitric oxide
NO_2	nitrogen dioxide
NG	natural gas
PHI	fuel–air equivalence ratio
SI	spark ignition
TDC	top dead centre

Development of gas engines using producer gas has been explored ever since World War II. It is estimated that over seven million vehicles in Europe, Australia, South America and Pacific Islands were converted to run on producer gas during World War II. These engines were spark ignited engines, mostly in the lower compression ratio bracket operating either on charcoal or biomass derived gas. Extensive fieldwork has been carried at National Swedish Testing Institute of Agricultural Machinery [1] by mounting gas generator and engine set on trucks and tractors. There have also been sporadic installations at Paraguay and Sri Lanka [1] for power generation application.

The question of power generation using producer gas has been addressed in recent times by a few researchers [1,2,6] and attempts have been made to convert standard compression ignition engine to a gas engine with the relaxation imposed on the compression ratio (CR) and others [3] operating a supercharged SI engine to realise the rated output. Also, one researcher [4] has reported working on producer gas fuelled engine at high CR (16:5 : 1) for water pumping application without any sign of knock. There appears no earlier work on a systematic study on the engine behaviour using producer gas fuel. The other important reason that appears responsible is the non-availability of standard and proven gasification systems, which could generate gas of consistent quality on a continuous basis for engine applications.

Systematic studies are essential from the viewpoint of establishing the highest useful compression ratio (HUCR) for producer gas fuel and also indirectly establish the octane rating for the fuel. This paper reports work on a producer gas fuelled spark ignition engine converted from a production diesel engine.

A well researched, tested and an industrially proven gasifier system capable of generating consistent quality was employed as the gas generator for testing purpose [7,8]. The engine has been tested and verified at the highest compression ratio of 17 : 1 in order to establish knock-less performance by capturing the pressure-crank angle trace. Subsequently the engine has been tested at varying CRs so as to arrive at an optimum CR for maximum brake power and effiency.The overall energy balance has been analysed and the shortcomings identified. Also the emission levels in terms of CO and NO have been examined.

2. Misconceptions and clarification

Prior to this development there have been two misconceptions regarding producer gas fuel and they are identified as follows: (1) auto-ignition tendency at higher CR when used in reciprocating engine, (2) large de-rating in power due to calorific value of the fuel being low.

It was thought that these perceptions had no reasonable basis. Indeed, the basis for the contrary seemed to exist. Firstly, producer gas being a mixture of many gas species with large fraction being inert should have higher octane rating when compared to natural gas and biogas. The gas contains a large fraction of inert gases like CO2 and N2 accounting to 12–15% and 48–50%, respectively, and these could act as knock suppressors [9]. However, so far there has not been any research of octane rating test conducted on producer gas fuel. Moreover, it is not clear if any established test procedure exists for producer gas like the Methane number test for natural gas and biogas. One crude way of assessment is to test the fuel gas in standard engines and place them accordingly in the octane rating table.

Secondly, there is a general thinking that producer gas being a lower calorific fuel, the extent of de-rating would be large when compared to high calorific value fuels like natural gas (NG) and liquefied petroleum gas (LPG). The de-rating if any could be due to two possible reasons. Firstly, with the lower energy density fuels there is a net decrease in number of molecules when compared to high-energy fuels like diesel, gasoline, NG or LPG. This contributes to some de-rating in case of low energy density fuels [10]. De-rating of power on account of calorific value will be small because of marginal differences in the energy release per unit mixture (air+fuel gas) [11]. This can be explained as follows. The calorific value of producer gas varies between 4.7 and 5:0 MJ N m−3 as against 30 MJ N m−3 for NG. The energy density per unit (producer gas +air) mixture is only 15–20% lower than NG and air mixture even though the calorific value of producer gas is one-eighth of NG. This is because the stoichiometric air=fuel ratio for producer gas is 1.2 as compared to 17 for NG. Hence the extent of de-rating with producer gas would not be marginal compared to NG fuelled operation at comparable operating conditions. This gap could be nullified by working producer gas at higher CR when compared to NG. The upper limit of CR for NG has been identified to be around 15:8 : 1 based on a recent work [10].

3. Earlier work

Shashikantha et al. [2] has reported related work on a converted diesel engine at CR of 11:5 : 1. In addition to the change in CR, the combustion chamber of the original engine i.e. bowl-in piston (hemispherical) was modified to Hesselman (shallow W) with an aim of achieving a higher level of turbulence by squish rather than swirl. With the above modification a power output of 16 kWe has been reported in gas mode with efficiency in the range of 21–24% against a rated output of 17 kWe in diesel mode. However, the same authors [6] subsequently claim a lower output and efficiency of 11:2 kWe and 15%. These authors do discuss the knock tendencies at higher CR but no experimental evidence seems to be provided in support. Measurements have been reported of various parameters including that of exhaust emissions, however no measurements have been made with respect to gas composition, which is considered essential from the viewpoint of establishment of input energy.

The first experimental work in the higher CR range has been reported by Ramachandra [4] on a single cylinder diesel engine (16:5 : 1 CR) coupled to a water pump. A power de-rating of 20% has been reported at an overall efficiency of 19% without any signs of detonation. This work does not report any other measurement like the pressure-crank angle diagram in order to rationalise some of the results. Work on gasoline engine operation on producer gas has been reported by Parke [12] with de-rating claims of 34%, compared to gasoline operation. The same authors [3] suggest supercharging to enhance the engine output. Martin and Wauters [5] have reported work using charcoal gas and producer gas on an SI engine with a de-rating of 50% and 40%, respectively, at a CR of 7 : 1. However, the same authors claim 20% de-rating when worked with producer gas at a CR of 11 : 1. The authors present a CR barrier of 14 : 1 and 11 : 1 for charcoal and producer gas, respectively, with inadequate experimental justification. From the literature survey it appears that no experimental evidence is available to support the phenomenon of knock in producer gas engines, even though it is believed knock would occur at higher CR.

4. Current investigations

It is a well-acknowledged fact that it is desirable to operate an internal combustion engine at the highest possible CR so as to attain higher overall efficiencies. But the gain in efficiency beyond a certain CR can be expected to be marginal due to other influencing factors such as heat loss and friction. In the case of an SI engine the limitation of CR comes from the knock sensitivity of the fuel. It has been experimentally investigated that the upper limit for compression ratio for SI engine operation is 17 : 1 beyond which there is a fall in efficiency [13]. The above conclusion is based on extensive tests with iso-octane as the fuel also doped with anti-knock agent. If one were to consider this as the upper limit and since no other work has been conducted at higher CR for SI engines, choosing a production engine in the above range for the current investigation seemed very appropriate.The current investigation was conducted on a commercially available diesel engine so as to explore the possibility of working at the existing CR of 17 : 1 and optimising the same if required. At the onset of investigation, it was perceived that increase in CR could have conflicting effects on the power output of the engine.This could be explained as follows. It has been universally recognised that turbulent flame speed [9] plays a vital role in the heat release rate during the combustion process in an engine cylinder. The turbulent flame speed can be treated as an enhanced form of laminar flame under the influence of time varying turbulence [9] within the combustion chamber of the engine. The laminar flame speed is again a function of initial pressure, temperature and the mixture composition. An earlier computational work by Mishra [14] indicated that the laminar flame speed for stoichiometric producer gas and air mixture could decrease by one-tenth as the initial pressure is enhanced by a factor of 40. However, these calculations were made at an initial temperature of 300K, and the initial temperature at which combustion starts is high in the case of internal Combustion engines. The influence of initial pressure and temperature on laminar flame speed can be explained in simple terms as follows. The increase in the unburned gas temperature results in increase in adiabatic flame temperature and hence the average reaction rates. The increase in the reaction rate is a result of the increase in the number of radicals released—thus contributing to increase in the flame speed, whereas the rise in pressure can result in reduction in the amount of radicals released thus retarding the flame speed. Therefore, the conflicting nature of the effects of initial pressure and temperature needs to be recognised. The effect of these at varying CR is an additional feature that needs to be recognised in order to arrive at the optimum CR. Consequently, the present investigation was started with an assumption that the optimum compression ratio would be between 12 : 1 and 17 : 1 for maximum power output and overall effciency.

In the current investigation, CR was the parameter that was varied. The influence of CR on power, effciency and emissions has been studied in some detail. Minimum ignition advance for best torque (MBT) has been determined at different Rs. The variation of cylinder pressure with time has been captured using a piezo-based transducer. The overall energy balance has been projected.

5. Conversion methodologies

A three cylinder, direct injection diesel engine of 3:3 l capacity, with a CR of 17 : 1 was converted into a spark ignition engine

1. Insertion of spark plug in place of fuel injectors without changing its location (centrally located).

2. Adaptation of a distributor type battery based ignition system with a provision to advance=retard ignition timing. The set ignition timing was checked using a stroboscope.

3. The combustion chamber design comprising a flat cylinder head and bowl-in piston was retained. No attempts were made to change the combustion chamber design except that the thickness of the cylinder head gasket was varied to accomplish different CRs of 17 : 1, 14:5 : 1, 13:5 : 1 and 11:5 : 1.

4. For in-cylinder pressure measurement, provision was made on one cylinder head by drilling a 1:5 mm diameter hole for

to drive a 25-kVA alternator. The salient features of the engine are given in Table 1. Modifications attempted on the engine for conversion are as follows:

pressure measurement and fitting an optical sensor on the crankshaft for crank angle measurement.

6. Experimental set-up and measurement scheme

The well-researched, tested and industrial version of IISc's-open top down draft, twin air entry 75 kg h^{-1} solid bio-residue gasifier system [7] formed the gas generator. This state-of-the art technology has undergone extensive testing both in India [8] and overseas [15] and proven to be a world-class system. The system has qualified for long hours of continuous operation in meeting the industrial requirements in terms of generation of consistent quality gas. The overall details of the gasifier system are presented in Fig. 1. As shown in the figure, the system had the provision to test the quality of the gas prior to supply to

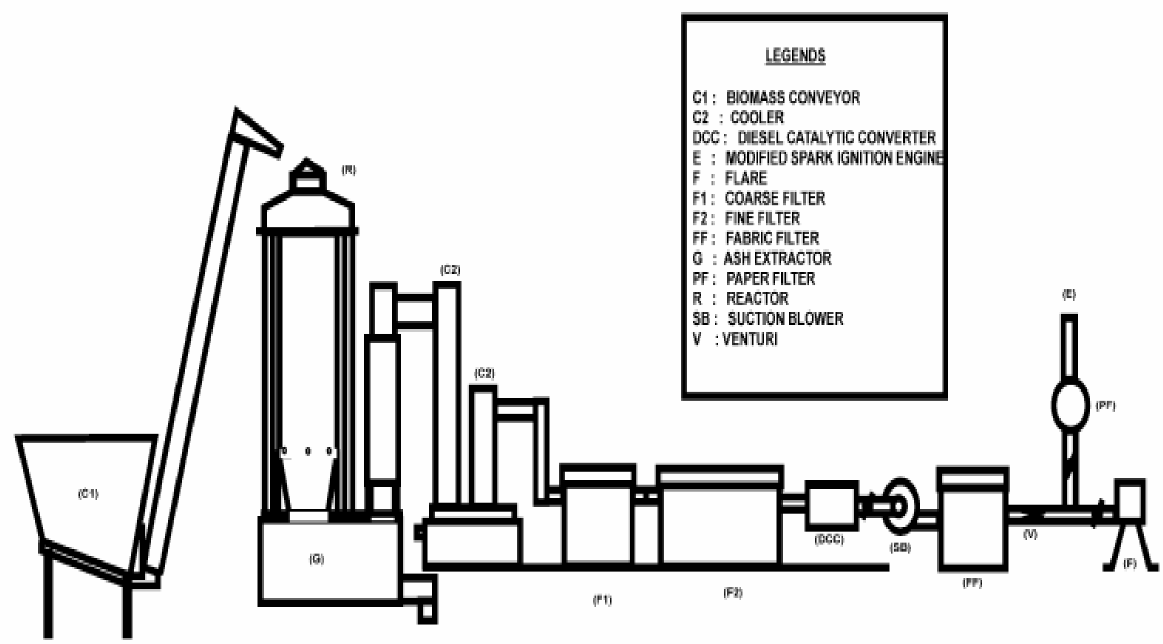

Fig. 1. Schematic of IISc's open top solid biomass gasifier system coupled to a spark ignition engine.

Table 1
Engine configuration details

Engine make, model	Kirloskar, RB-33 coupled to a 25 kVA alternator
Rated output (in diesel)	28 kW at 1500 RPM
Output measured (in diesel)[a]	24 kW (21 kWe) at 1500 RPM
Bore × Stroke	110 × 116 mm
Number of cylinders	3
Type of cooling	Water
Compression ratio	17 : 1
Bumping clearance	1 mm
Combustion chamber	Flat cylinder head and bowl-in piston type
Squish area	70%
Ignition system	Battery-based distributor type with ignition advance/retard facility
Spark plug type and location	Cold, offset from centre of combustion chamber by 8 mm
Valve port	Tangential—directed type
Valve timing	Inlet valve opening—26 BTDC Inlet valve closing—66 ABDC Exhaust valve opening—64 BBDC Exhaust valve closing—38 ATDC
Air/Fuel ratio in diesel mode at peak load (24 kw)	$20.5 \pm 0.5 : 1$
Alternator efficiency[b]	87%
Gasification efficiency[c]	80%

[a] At Bangalore, 1000 m above sea level.
[b] As per manufacturer's specifications.
[c] See Ref. [7].

the engine. At the engine intake, a carburetor is provided for proportioning air and fuel flow. As there were no carburetors commercially available to cater to producer gas, a locally made carburetion system and manually controlled valve were used for proportioning.

Measurements were made with respect to the following parameters:

(A) Producer gas compositions using on-line gas analysers. The gases analysed were CO, CO_2, CH_4, O_2 and H_2. The N_2 concentration was deduced by difference. The CO, CO_2, CH_4 components were determined using infrared gas analysers and the H_2 component using a thermal conductivity-based analyser. The O_2 measurement system was based on chemical cell.

(B) In-cylinder pressure variation data synchronized with the crank angle measurement was acquired on a computer for every one-degree crank angle. The pressure measurement was accomplished using a pre-calibrated Piezo based pressure transducer (M=s PCB make).

(C) Measurement of voltage and current across three phases and frequency for power output calculations—the load bank constituted of resistors.

(D) Air and gas FLow to the engine using pre-calibrated venturimeters.

(E) Engine exhaust analysis—O_2, CO_2; CO, and NO and temperature.

7. Experimental procedure

Experiments were initiated on the engine only after the gasifier system stabilised i.e. attained steady state operation in terms of generation of consistent quality gas. The typical time scale for attaining steady state of operation from the cold start was 2–3 h. 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 minimise errors in long duration operation. Typically gas composition at the time of start of the engine test was 19 ± 1% H_2; 19±1% CO; 2% CH_4; 12±1% CO_2; 2±0:5% H_2O and rest, N_2. The mean calorific value of gas varied around 4:65±0:15 MJ N m^{-3}. The feedstock used for gasification is Causurina species wood with moisture content between 12% and 15% on dry basis (sun-dried wood).

Once the gas composition stabilised, the engine was operated for a few minutes at 1500 RPM at no-load condition. All the tests on the engine were conducted around a constant speed of 1500±50 RPM. The throttling for speed control and air and fuel proportioning was achieved using manually operated valves.

Experiments were conducted at CRs of 17 : 1, 14:5 : 1, 13:5 : 1 and 11:5 : 1 and these CRs were achieved by varying the thickness of the cylinder head gasket. The compression ratio values are based on the cylinder’s geometric measurements and were verified by matching the motoring curve with an engine simulation curve. The engine was tested at different ignition timing settings to determine the MBT at different CRs. With a set ignition timing, the air and fuel were tuned to achieve maximum power. Measurements were initiated 10–15 min after attaining stable operation. The in-cylinder pressure data with a resolution of 1° crank angle was acquired on a computer in excess of 100–150 consecutive cycles. Prior to the start of these tests, the TDC was accurately determined using a dial gauge and synchronized with the optical crank angle measuring system with an accuracy of ±0:5°.

8. Results and observations

8.1. Performance

The first and the foremost result of these tests is that the engine worked smoothly without any sign of knock at a high CR of 17: 1. There was no sign of audible knock during the entire load range. Moreover, the absence of knock was clear from the pressure-crank angle recordings both at full load and part load.

The engine delivered a maximum output of 17:5 kWe (20 kW shaft power) at a CR of 17 : 1 with an overall efficiency of 21% compared to 21 kWe (24 kW shaft power) output at 31% efficiency in diesel mode. The overall efficiency calculated is based on the ratio of the shaft output delivered to the energy content in the biomass. The overall efficiency in gas mode is based on the ratio of mechanical shaft output to the energy content in the biomass. The useful output and efficiency decreased with the lowering of CR. A maximum output of 15:3 kWe (17:6 kW shaft power) at an overall efficiency of 18% was obtained at a CR of 11:5 : 1. The variation of brake power with CR is shown in Table 2. The power output at an intermediate CR of 14:5 : 1 and 13:5 : 1 were 16.4 and 16:2 kWe, respectively, and with overall efficiencies being 20%. The overall efficiency at 13.5 CR is the same as that at 14:5 : 1 on account of leaner operation.

Table 2
Engine output at MBT

CR	IGN, BTDC	PHI	H_2/CO	Max Power (kWe)
17:1	6	1.05	1.01	17.5
14.5:1	10	1.04	1.00	16.4
13.5:1	13	0.86	1.08	16.2
11.5:1	16	1.00	1.06	15.3

The extent of de-rating in brake power is about 16.7% at a CR of 17 : 1 and increased as high as 26% at 11:5 : 1 when compared to diesel mode of operation. The gain in overall efficiencies from CR 11.5 to 17 : 1 works out to be 16.6%, which means an increase of 3% per unit CR increment. However, the incremental gain per unit CR from 14.5 to 17 : 1 is 2%. These figures are well within the range of 1 to 3% gains per unit incremental of CR [9]. The fuel–air equivalence ratio (PHI) at which the maximum power was derived was around 1.00–1.06 with the exception of 0.86 at CR of 13:5 : 1. The air to fuel ratio is tuned from the viewpoint of deriving maximum output and therefore the efficiency figures are necessarily not the maximum that can be obtained. It may be possible to achieve higher overall efficiencies by operating at leaner conditions.

The peak shaft output at varying CR was found to be sensitive to the producer gas composition. The hydrogen fraction in the fuel gas dictated the ignition timing setting. Therefore the minimum advance for brake torque (MBT) varied with the change in hydrogen content in the fuel gas. A higher fraction of hydrogen means that the ignition timing has to be retarded in order to benefit from the increase in the flame speed. With a faster burn rate the optimum spark timing has to be closer to the TDC, the mixture temperature and pressure at the time of initiation of spark will be higher and hence the laminar flame speed at the start of combustion will also be higher. Therefore optimizing the ignition timing based on hydrogen fraction is vital from the viewpoint of deriving maximum shaft output.

Since the mixture flame speed is a strong function of hydrogen content in the gas, the MBT will differ based on the actual fuel gas composition. For a gas composition containing 20:5 ± 0:5% H_2 and 19:5 ± 0:5% CO, the MBTs have been identified as shown in Table 2, the measurement accuracy of MBT is within 3° crank angle. The MBT in the present case has turned out to be about 6–10° BTDC at a CR of 17 : 1 and has gone up to as high as 14–16° BTDC for a CR of 11:5 : 1. These values are much lower compared to 30 to 40° BTDC at a CR of 11.5 based on earlier work [2,6,12] but matches with 10° BTDC [4] at a CR of 17 : 1. The mechanical efficiency of the engine at a CR of 17 : 1 is about 80% and increases to as high as 87% at a CR of 11:5 : 1. The increase in mechanical efficiency is attributed to the reduction in rubbing friction [16] due to lower cylinder pressures encountered at lower CRs. The mechanical efficiency values are based on indicated power measurement (based on integration of pressure–volume diagram) and these were found to be identical with the Morse test results.

8.2. Pressure-crank angle data

The pressure–crank angle recording at all the CRs did not show any trace of knock for all ranges of load including that of peak load and this is visible from the pressure–crank angle diagrams as shown in Fig. 2. Faster burn rate due to presence of hydrogen in the fuel gas may be one factor for the non-knocking performance at higher compression ratio. The faster burn rate accompanied by retarded ignition timing setting obviates any auto-ignition tendency of the end gas. Increasing the flame speed or retarding the ignition timing setting is one possible way of reducing knock tendency and this is well acknowledged in the literature [9]. The maximum indicated mean effective pressure (IMEP) was obtained at ignition timing corresponding to the maximum shaft output. The IMEP obtained at varying CR as a function of ignition timing is shown in Fig. 3. The max IMEP recorded was 595 kPa at a CR of 17 : 1 and declined to about 485 kPa at a CR of 11:5 : 1. The maximum IMEP was obtained at a PHI of 1.05 and this falls well within the anticipated PHI of 1.0 to 1.1 [9]. The point of occurrence of peak pressure at all CRs occurred at 18–19° ATDC, except that at 13:5 : 1 which occurred at 17° ATDC, which is close to the generally acknowledged value of 17° ATDC for MBT [9,17]. Therefore, for CR of 17 : 1, 14.5 and 11:5 : 1 the MBT should be more advanced

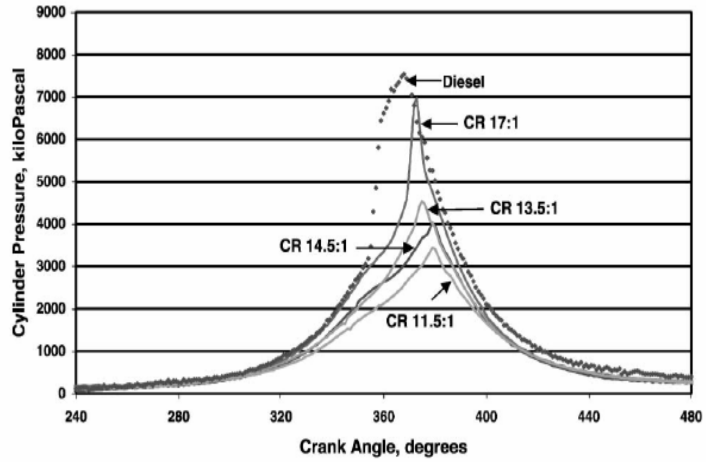

Fig. 2. Pressure–crank angle diagram at varying compression ratio; CR 13 : 5 : 1 is at MBT and CR 17 : 1, 14.5 : 1 and 11.5 : 1 the ignition advance is sub-optimum. Dotted plot shows the diesel mode at 90% of full load operation.

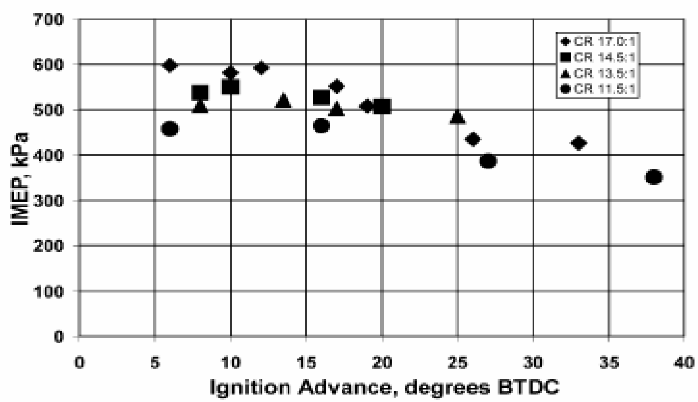

Fig. 3. Variation of IMEP with ignition timing and CR.

Table 3
Cylinder peak pressures and their occurrence

CR	Peak pressure (kPa)	Occurrence of peak pressure, degrees ATDC
17 : 1	5730	19
14.5 : 1	4330	19
13.5 : 1	4850	17
11.5 : 1	3470	19

(lie within 2–3◦) from what was actually measured. The variation of IMEP or the net useful output within this close range would be marginal. The peak cylinder pressures and their point of occurrence is shown in Table 3. The peak pressure at 14:5 : 1 CR is lower than 13:5 : 1 probably due to a slight departure from MBT. The coefficient of variation of the IMEP at all CRs and ignition settings occurred well within 3–3.5%, implying low cycle-to-cycle variation. The reason for low cyclic variation is the faster rate of combustion 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 bowl-in piston combustion chamber design with increased squish effect.

8.3. Energy balance

Fig. 4 represents the overall energy balance at a CR of 17 : 1. The energy balance at MBT showed that about 32.5% of the energy was realised as useful output (indicated power), about 27% was lost through the exhaust (including the CO in the exhaust) and the remaining 40% to the cooling water (inclusive of radiative losses, etc). As expected with the advancement of ignition timing, the loss through exhaust mode reduced and increased to the coolant route. The energy balance in gas mode showed that a large fraction of the energy was lost to the cooling water when compared to the diesel mode. Fig. 5 compares the energy balance in gas and diesel mode (at rated output of 21 kWe) at a CR of 17 : 1, the energy loss at maximum power delivered to the coolant and miscellaneous is about 40% compared to 33% in diesel and whereas, the energy loss through exhaust in gas mode decreased by 5%. The indicated power and thereby the thermal efficiency being higher in diesel mode is evident from the pressure–time curve shown in Fig. 2. The energy balance at varying CRs is shown in Fig. 6. There was an increase in the loss of energy through exhaust (which includes the energy in the form of CO) with the reduction in the CR, whereas, the loss through the coolant and miscellaneous was higher at higher CR. The useful energy is 32.5% at a CR of 17 : 1 and has declined to 25.7% at a CR of 11:5 : 1. The useful energy at a CR of 13:5 : 1 has stood about the same as that at 14:5 : 1 due to a relatively leaner operation. The increased amount of heat loss to the cooling water as a whole in gas operation

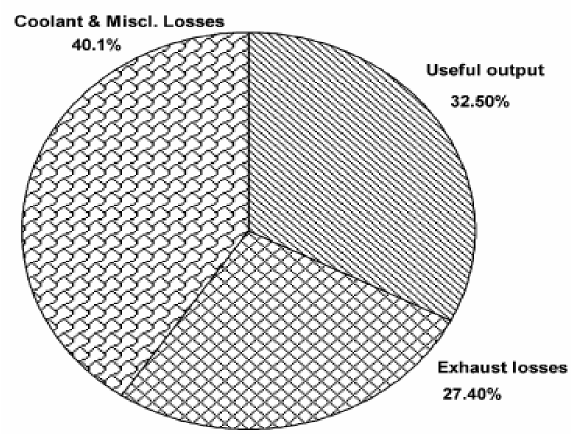

Fig. 4. Overall energy balance in gas mode at CR 17 : 1.

could be attributed to the engine combustion chamber design. It has been quoted in the literature [9] that with engine geometries such as bowl-in-piston there will be 10% higher heat transfer. The heat transfer to the coolant in the current case falls well within this range. With the increase in compression ratio the overall conversion efficiencies must improve thermodynamically, similarly, the heat loss to the coolant and exhaust should have reduced. However, increased energy loss to the coolant at a higher compression ratio is probably due to increase in heat transfer coefficient.

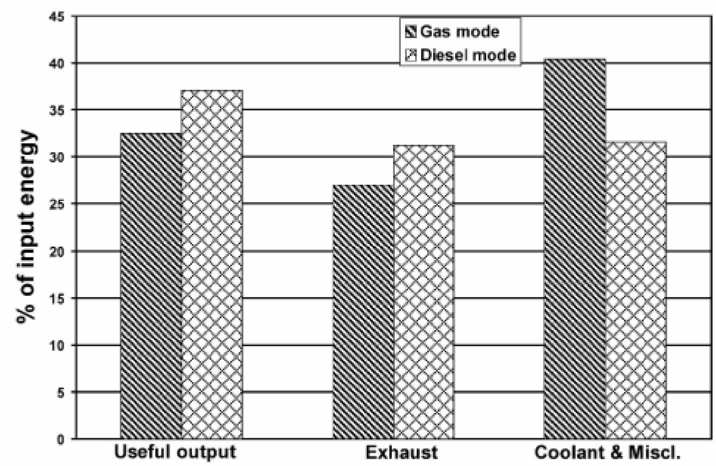

Fig. 5. Overall energy balance in gas and diesel at 17 : 1 CR at peak output.

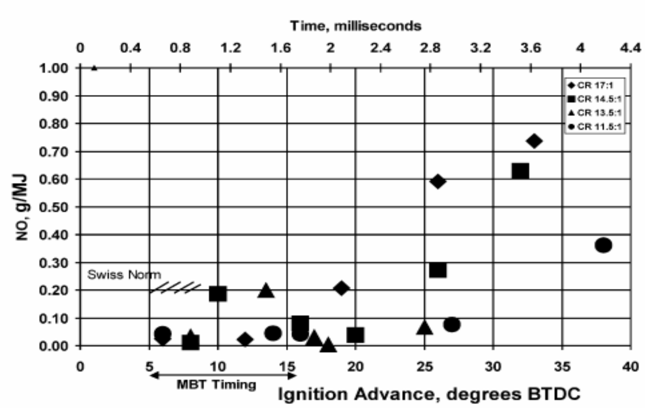

Fig. 7. Variation of NO with ignition advance.

accounted for because it forms a small part (□ 5%) of NOx generated [9]. The NO level has been represented in milligram per unit MJ of input energy. These results have been compared to the Swiss norms because of some earlier collaborative work with the Swiss scientists; it was indicated that Swiss norms were stringent with respect to the emission levels. The NO level reduced with the retardation of ignition timing and this feature was observed for all CRs. The NO level was observed to be maximum at the highest compression ratio with advanced ignition timings, whereas for the MBT range of 6–20° BTDC the NO was roughly about the same in almost all the cases. However, there was one exception of NO being higher at MBT for a CR of 13.5 due to a leaner operation.

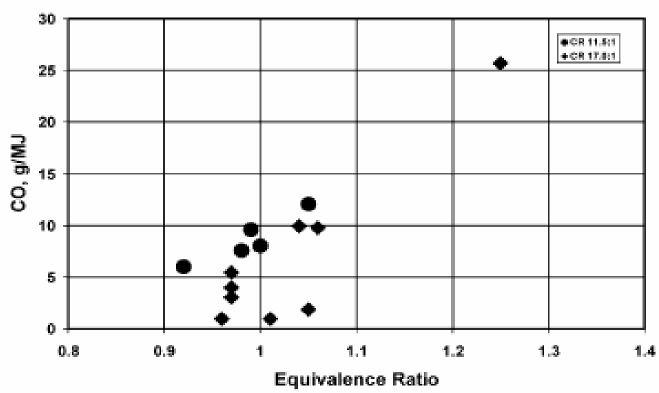

Fig. 8. Variation of CO with equivalence ratio.

It is a well-known fact that NO generation is strongly dependent on the temperature and also residence time in the combustion chamber. With the flame speed of the gas mixture being high, the ignition setting is retarded whereby the residence time in the high temperature combustion chamber is automatically reduced. Therefore, the low NO levels at retarded ignition setting is an expected and consistent behaviour. The above results match well with those quoted in the literature [9], which show small to modest variation of NO with CR. The variation of carbon monoxide (CO) with PHI is shown in Fig. 8. The CO levels have been represented in grams per MJ of input energy. The trend of CO with PHI is clear from the figure. The CO levels were lower at the highest CR, and this could be attributed to higher temperatures, leading to relatively complete combustion.

9. Conclusions

Performance of the engine at higher CR has been smooth and it has been established beyond doubt that the operating engines using producer gas in SI mode at higher compression ratios is technically feasible. The cylinder pressure–crank angle trace has shown smooth pressure variations during the entire combustion process without any sign of abnormal pressure raise. 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 3%. The faster burning process has been identified to be due to the higher flame speed of the fuel gas mixture and the same has been attributed to the hydrogen content in the gas. So, increased quantity of hydrogen in the fuel gas mixture is desired from the viewpoint of approaching an ideal cycle operation. The perceived negative influence of pressure on flame speed at higher CR does not seem to exist based on the above results. The maximum de-rating in power is observed to be 16% in gas mode when compared to diesel operation at comparable CR. The extent of de-rating was much lower when compared to any of the previous studies [3–6]. This number matches with a similar kind of de-rating reported with NG operation [10]. However, the overall efficiency drops down by almost 32.5% compared to normal diesel mode of operation. A careful analysis of the energy balance revealed excessive heat loss to the coolant at all CRs and this resulted in engine overheating within 30–40 min of operation at full load. This phenomenon of excessive heat loss could be attributed to bowl-in piston combustion chamber design. Higher heat loss to the cylinder walls can be expected because the combustion chamber is originally meant for diesel operation with inherent swirl. Hence suitable modification of the combustion chamber is essential from the view point of reducing the energy loss to the coolant. Reduction in 10% heat loss to the coolant would amount to improvement in overall efficiencies by 3%. The study of the engine combustion chamber and modifications can be looked upon as future work.

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