Energy 63 (2013) 317–333

Contents lists available at ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

# Performance analysis of an internal combustion engine operated on producer gas, in comparison with the performance of the natural gas and diesel engines

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ARTICLE INFO

*Article history:*
Received 29 March 2013
Received in revised form
13 August 2013
Accepted 11 October 2013
Available online 5 November 2013

*Keywords:*
Producer gas
Internal combustion engine
Compression ratio
Expansion ratio
Power generation efficiency

ABSTRACT

The growing economy and changing lifestyle have increased the demand for modern energy, like electricity. Globally 1.3 billion people are without access to electricity. In India, 289 million people do not have access to electricity. Decentralized distributed power generation using renewable energy is a competitive alternative for energy supply to all, with a sustainable growth. The performance of an internal combustion engine fueled with 100% producer gas was studied at variable load conditions. The engine was coupled with a 75 $kW_e$ power generator. Producer gas generated from a downdraft gasifier system was supplied to the engine. The overall power generation efficiency of 21% was achieved above 85% load. The power generation efficiency of the producer gas engine was estimated at variable load conditions. The influencing factors of the power generation efficiency of a producer gas engine, such as volumetric efficiency, energy density of the fuel mixture, adiabatic flame temperature, compression ratio and expansion ratio were studied in detail. A relation between volumetric efficiency, expansion ratio, compression ratio and thermal efficiency was established and verified. The efficiency of the engine estimated using the new method has a correlation coefficient of 0.99 with the efficiency estimated using the energy input and output.



## 1. Introduction

Globally 1.3 billion people do not have access to electricity, 84% of these people live in rural areas. In India about 289 million of people who account for 25% of the population do not have access to electricity [1]. Biomass fuels are still contributing to 14% of the world energy demand and 38% of the developing countries [2]. In the past six decades India's energy need has increased by 16 times and the installed electricity generation capacity by 84 times [3]. With the economy projected to grow at 8%–9% per annum and the improving standards of millions of population, the energy demand is likely to grow significantly. Biomass is a potential renewable source for power generation [4]. Deployment of biomass gasification technology can meet the triple bottom line for growth of – local energy, local employment and local economy [5]. Biomass gasification is one of the potential options for DDG (decentralized and distributed generation) of electricity [6]. Electricity generation through biomass combustion and gasification was considered as a potential source to meet the rural energy needs [7]. India is having a biomass power potential of 16,000 MW from agro residues and 45,000 MW from plantations [8].

In India, a national program (Phase I) was launched during 1987–1993 for technology development and demonstration of biomass gasifier systems. This program focused on promotion of irrigation pumps in the range of 3–10 hp and decentralized power generation in the range of 3–100 $kW_e$ [9,10]. Dual fuel engines were used during this program. A national program of Phase II was implemented during mid and end of 90's. During the second phase, the gasifier system was used in the range of 10–100 $kW_e$ power generation system for rural electrifications. Based on the experience of this program, the need for compatibility of the engine in terms of ease of operation and economic viability was emphasized in Ref. [9]. DPS (distributed power system) is proposed as one of the options for economic, environmental and energy security [11]. During 2005, a VESP (village energy security program) was launched by the MNRE (Ministry of New and Renewable Energy), India. This program aimed at providing the total energy required by the remote villages. Only 45 biomass gasifier based power plants were installed out of 95 plants sanctioned for the program [12]. Only 34 plants out of the 45 plants installed were functional due to various reasons. The status of gasification technology for operating

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0360-5442/$ – see front matter 
http://dx.doi.org/10.1016/j.energy.2013.10.033

an internal combustion engine cannot be ignored. There is a need to understand the performance of the ICE (internal combustion engines), when running on producer gas and identifying ways to remove any barriers to their performance. Generally, internal combustion engines are fueled with producer gas either on dual fuel mode (along with diesel) or on 100% producer gas.

### 1.1. Internal combustion engines operated using producer gas on dual fuel mode

The history of gasification development starting from 1699 to 1970 as reported in Ref. [13]. It also reports that the first patent regarding gasification was obtained by Robert Gardner during 1788. The target and benchmarks for gasification were reported in 1906 [14]. Biomass gasification technology has been in existence for more than 80 years since world war two [15–17]. The first attempt to use producer gas to operate an internal combustion engine was carried out in 1881 [18,19]. The first well reported conversation about using a producer gas engine for operating tractors was during 1931–1934 [20]. Initially, diesel engines were operated along with producer gas in dual fuel mode. In dual fuel operation, 60–65% of diesel replacement was obtained while using an engine with a capacity of 5.25 kW [3,21].

A dual fuel engine was operated with a power generation efficiency of 19%, with a diesel replacement of 59% [22]. A maximum efficiency of shaft power was obtained at 21% in dual fuel operated engine [23]. The specific fuel consumption to generate one kWh of electricity was reported as 1.28 kg of fuel wood and 65 mL of diesel [24]. Presently most of the engines are working at a very low efficiency. Particularly the producer gas engines are working with an overall efficiency in the range of 20–22%. Efficiency closer to 20% is achieved only at the maximum operating capacity of the engine. At part load operating conditions, the efficiency of the engine is much lower. It is essential to operate the biomass gasifier based power generation system at higher efficiency to reduce fuel consumption and substitute the use of fossil fuel. Energy efficiency issues are discussed in the context of technology and trends in energy use [25]. While encouraging technology driven economic growth there should be a focus for reduction of GHG (green-house gas) emission [26]. Improvement in performance efficiency of engines by reducing fuel consumption helps to reduce GHG emissions substantially and achieve sustainable growth. The present study involves a detailed performance analysis of a biomass gasifier coupled with a producer gas engine for improving the overall efficiency of the system. Electrical output from the engine and energy flow through flue gas and radiator cooling fluid was monitored at variable load conditions.

Internal combustion engine operated using producer gas was studied during 1896 using different types of engines [14]. This report also highlights a serious limitation to the gasifiers for production of good quality gas in those days. This report mentions about 11 types of difficulties related to gas quality and technology status which act as a barrier for promotion of producer gas engines. Most of the difficulties mentioned in this report remain unsolved even with today's status of technology.

### 1.2. Internal combustion engines to run on 100% producer gas

Initially, diesel engines were run along with producer gas on dual fuel mode to avoid complications involved in modifying an engine to run on 100% producer gas. Due to increase in cost of diesel and its scarcity in rural areas it was preferred to run engines on 100% producer gas. The increasing cost of diesel price makes it too expensive to generate power on dual fuel mode [27]. A diesel engine needs more modifications to run with 100% producer gas. The engine modifications needed are introduction of a spark ignition system, a gas carburetor (fuel intake manifold) for supplying the required fuel mixture and a governor to control the throttle valve for controlling the fuel flow according to the operating load and hence, maintain engine speed. The engine manufacturers are not in favor of engines operating with 100% producer gas due to the impurities in the gas. Now engine manufacturers like Cummins are manufacturing heavy duty IC engines having 12 cylinders which can run on 100% producer gas [28].

### 1.3. Impurities in producer gas

The real difficulty is not generating combustible gas from a gasifier but obtaining a good quality gas which can be used to economically operate IC engines for a longer duration [20]. Tar formation remains a technical hurdle for the development of biomass gasification [29]. The key issue for successful application of producer gas engines is the removal of tar and further development of the system for obtaining cleaner gas [30]. The problems related to tar and particulate matters in the producer gas were discussed in context with power generation in Ref. [31]. Gas purification by removing contaminants like tar and particulate matters and problems associated with catalyst based gas purification were discussed by Ref. [32]. This paper also highlights the need for an advanced gasification system to produce cleaner gas.

Currently many institutions are involved in technology development to improve the gas quality and make the system user friendly [33,34]. The tar content in the producer gas was reduced to 19–34 mg $Nm^{-3}$ by using a charcoal coupled two stage wood gasifier [35]. The producer gas generated from a downdraft gasifier has less tar content and is suitable for running IC engines [36]. To have an economical long term operation of an engine it is important to reduce harmful contaminants in the gas such as hard solid particles and corrosive compounds [37]. This paper also reports about the effect of particles present in producer gas on the performance of the engine. Particles larger than $1 \pm 2$ μm can bridge the protective oil film between the moving parts and cause abrasive wear. Smaller particles may contaminate and thicken the lube oil which can cause wear in the moving parts of the engines. Hence, the particulate matters in the producer gas and its size should be within the allowable range to ensure longer durability of the engine.

## 2. Methodology

Performance of a producer gas engine coupled with a power generator having a design capacity of 75 $kW_e$ was studied and analyzed. The objective of this study was to analyze the performance of an internal combustion engine operated using 100% producer gas. The present study focuses on analyzing the property of producer gas and its influence on the performance of producer gas engines. The quality of producer gas depends upon the type of the gasification reactor used in the system. A downdraft type gasification reactor was designed to operate the producer gas engine. The experimental study and analysis were carried out by monitoring the performance of the gasifier system and the producer gas engine. An engine designed to run on natural gas was used to operate on 100% producer gas. The components of the gasifier system, analysis of the producer gas and performance of the producer gas engine are discussed in this paper. A diagram depicting the components of the biomass based power generation system with the producer gas engine is shown in Fig. 1. From Fig. 1 it may be noted that a biomass gasifier power plant consists of a biomass gasifier, gas cleaning train, manifold for supply of appropriate air–gas fuel mixture and a producer gas engine.

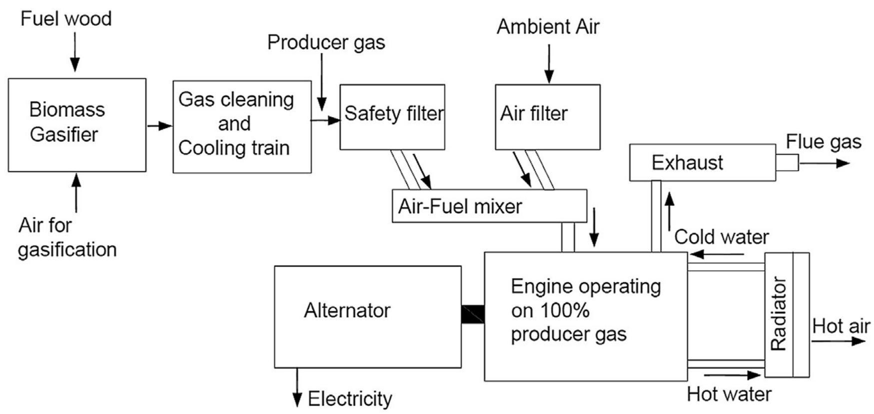


**Fig. 1.** Components of the biomass based power generation system.

### 2.1. The experimental setup

The experimental setup shall enable measurement of different parameters selected for the experimental analysis. The experimental setup for monitoring the performance of the producer gas engine included gas sampling ports, temperature sensors, equipment for measuring flow rate of air and gas and power output. The experimental setup was created for the measurement of various operating parameters related to the gasifier and the producer gas engine. Fuel wood consumption, gas production rate and quality of the producer gas were monitored at the gasifier side. Gas flow rate, air flow rate, electrical power output, energy flow through the exhaust and the radiator were monitored at the engine side. K type temperature probes were installed to monitor the gas temperature and water temperature at various locations. A hot wire anemometer was used to monitor the flow rate of gas and air. Provisions were made at the engine's gas inlet and air inlet for installing the flow sensors of the hot wire anemometer. ICT (induction coil transformers) were installed at each phase to monitor the electrical output of the alternator coupled to the 100% producer gas engine. A load bank consisting of electrical heaters was installed to study the performance of the producer gas engine at variable load conditions.

### 2.2. Equipment used during the experiment

Several equipments were installed for monitoring the performance of the engine and parameters such as gas quality and gas flow rate influencing the performance of the engine. Fuel wood consumption was monitored by weighing the biomass fed into the gasifier at regular intervals and at different loads. Gas flow rate and air flow rate were measured by using a hot wire anemometer. The operating load of the engine was monitored by a Watt meter. Total power generated during the experiment was monitored using a three phase energy-meter. Digital thermometer was used to monitor the fluid temperature at the exhaust and the radiator of the producer gas engine. The details of the equipment used during the experiment are presented in Table 1.

### 2.3. Experimental conditions

The performance efficiency of the biomass gasifier system and the producer gas engine was analyzed at variable load conditions. Energy balance analysis was carried out at 97% and 31% load. Energy lost through the radiator and the exhaust was monitored to carry out the energy balance analysis. The producer gas engine was operated at variable load conditions to monitor the biomass consumption rate, gas consumption rate, performance efficiency and specific fuel consumption rate of biomass and producer gas. The producer gas engine was operated at variable load conditions by varying the load from 7 to 97% of the design capacity. During the experiment fuel wood from the same lot was used to avoid any variation in the fuel quality. Experimental conditions of the present study at the maximum operating load of 73 $kW_e$ are presented in Table 2. The maximum operating load of 73 $kW_e$ corresponds to 97% load of the design capacity.

The results obtained on gas quality and producer gas engine's performance were compared with published results available in various research papers and reports. The performance of the producer gas engine is compared with the performance of diesel engines and natural gas engines. The efficiency of the producer gas engine is discussed with reference to the energy density of the fuel mixture, Wobbe number, adiabatic flame temperature, volumetric efficiency, compression ratio, expansion ratio and combustion pressure.

**Table 1**
Details of the equipment used during the experiment.

| S. No. | Instrument | Measurement | Least count | Error level |
|---|---|---|---|---|
| 1 | Hot wire anemometer | Flow measurement | 0.1 m $s^{-1}$ | ±1% |
| 2 | Pressure differential meter (PDM) | Pressure drop | 0.1 mm | ±0.2% |
| 3 | Digital temperature indicator | Temperature measurements | 0.1 °C | ±1% |
| 4 | Weighing balance | Fuel feeding rate | 100 g | ±20 g |
| 5 | Watt meter | Monitoring the power output | 1 W | ±0.5% |
| 6 | Energy meter | Monitoring the energy output | 1 kWh | ±0.5% |
| 7 | Gas Chromatograph | Gas component analysis | 0.01% | ±1% |

**Table 2**
Experimental conditions.

| S. No. | Components | Unit | Value |
|---|---|---|---|
| 1 | Maximum power output | $kW_e$ | 73 |
| 2 | Fuel wood consumption at the maximum load | kg $h^{-1}$ | 86 |
| 3 | Calorific value of the fuel wood | MJ $kg^{-1}$ | 18.0 |
| 4 | Gas flow rate at the maximum load | $m^3$ $h^{-1}$ | 241 |
| 5 | CV of gas | MJ $Nm^{-3}$ | 5.7 |
| 6 | Air flow | $Nm^3$ $h^{-1}$ | 289 |
| 7 | Flue gas | kg $h^{-1}$ | 610 |
| 8 | Ambient temperature | °C | 30 |
| 9 | Flue gas temperature | °C | 565 |
| 10 | Hot water from engine | °C | 73 |
| 11 | Cold water from radiator | °C | 63 |

The energy density of the fuel mixture is estimated by Eq. (1).

$$E_d = C_{vf}/\left(V_f + V_a\right) \tag{1}$$

The compression ratio is expressed by Eq. (2).

$$C_r = (S_v + D_s)/D_s \tag{2}$$

The volumetric (filling) efficiency of the engine is calculated by Eq. (3).

$$V_e = \left(V_f + V_a\right)/V_s \tag{3}$$

The expansion ratio is estimated by Eq. (4).

$$E_r = \left\{\left(V_f + V_a\right)_{T1} \times [(T_1 + A_t)/T_1]\right\}/\left(V_f + V_a\right)_{T1} \tag{4}$$

The efficiency of the producer gas engine using the energy input and energy output can be estimated by Eq. (5).

$$\eta_e = (E_o/E_i) \times 100 \tag{5}$$

The efficiency of the engine without considering the volumetric filling efficiency and friction loss is defined in Ref. [38]. The effect of the volumetric efficiency and friction loss in the efficiency of the engine was reported in Ref. [39]. At the maximum load the efficiency of the engine based on its compression ratio and volumetric efficiency can be estimated by Eq. (6).

$$\eta_c = \left(1 - \left[1/_{\varepsilon^{1-k}}\right]\right) \times V_e \tag{6}$$

In the case of a producer gas engine the expansion ratio ($E_r$) of the fuel mixture is less than the compression ratio ($C_r$). The actual working efficiency of an engine using the volumetric efficiency, compression ratio and the expansion ratio can be theoretically estimated by Eq. (7). $\eta_c$ is a function of volumetric efficiency which varies along with the load. Eq. (7) can be used to estimate the efficiency of the engine at any required operating load (even at part load operating conditions).

$$\eta_{E_r} = \eta_c \times (E_r/C_r) \tag{7}$$

The combustion pressure at any given position of the piston with respect to the TDC (top dead center) can be estimated by Eq. (8).

$$P_\theta = [(S_{v-\theta} + D_s)/D_s] \times E_r \tag{8}$$

The Wobbe number and the energy density of the fuel mixture are interrelated. As the Wobbe number is a function of the calorific value of the fuel, it has a strong influence on the expansion ratio of the fuel mixture.

The Wobbe number of the fuel mixture can be estimated by Eq. (9).

$$W_n = C_v/\left(\sqrt[2]{\rho_g}\right) \tag{9}$$

## 3. Producer gas engines

In the beginning, producer gas engines were operated on dual fuel mode along with diesel. In dual fuel mode, consumption of diesel was reduced to the range of 60–80%. Hence, there was a need to provide diesel for the balance of 20–40% fuel requirement of the engine. Due to increasing diesel price, it is preferred to operate engines with 100% producer gas. Thus, there is a need to design producer gas engines which can run on 100% producer gas at a higher efficiency and with a minimum use of biomass. The design of producer gas engine should comprise of a well-designed intake manifold, suitable spark ignition system with appropriate advanced ignition timing and appropriate compression ratio. Producer gas unlike conventional fuels like petrol, diesel, natural gas, etc. has different combustion characteristics when compared to the conventional fuels like petrol, diesel, and natural gas. On comparison of the ignition systems, diesel engines have a compression ignition system while the petrol and natural gas engines have a spark ignition system. Producer gas engines also require a spark ignition system for ignition of the fuel mixture comprising of air and producer gas. The stoichiometric ratio of air required for an amount of producer gas varies as compared to other fuels. Hence, there is a need for a well designed intake manifold for supply of air and producer gas in the appropriate ratio at variable load conditions.

As ready-made producer gas engines are not available in the market, two options are available for modifying existing engines to run on 100% producer gas. One option is to modify a diesel engine to run on 100% producer gas while the other option is to modify a readily available natural gas engine. Modifications required for a diesel engine to run on 100% producer gas include installing a spark ignition system along with a new governor mechanism and reduction of compression ratio. On the other hand, modifications required for a natural gas engine to run on 100% producer gas is minimum as natural gas engines are already equipped with a spark ignition system and a governor mechanism to control the throttle valve. As compared to diesel engine natural gas engines are designed with low compression ratio. Due to these reasons the natural gas engines are required a minimum modification to run on 100% producer gas. However, in both the options an appropriate intake manifold (for gas and air) has to be installed to run the engine on 100% producer gas. The choice of engine for modification i.e. diesel or natural gas engine need to be selected considering conditions such as types of engines available and availability of skilled persons in the locality. To avoid search for skilled technicians and complications associated with modifying diesel engines, preference is given to procure and modify a natural gas engine to run on 100% producer gas.

**Table 3**
Engine configurations.

| S. No. | Component | Unit | Details |
|---|---|---|---|
| 1 | Rated engine capacity (in natural gas) | hp | 133 |
| 2 | Rated engine speed | RPM | 1500 |
| 3 | Rated power output (in natural gas) | $kW_e$ | 85 |
| 4 | Power output (in producer gas) | $kW_e$ | 75 |
| 5 | Power de-rating (in producer gas) | Fraction | 0.14 |
| 6 | Bore × stroke | mm | 132 × 150 |
| 7 | Number of cylinders | No. | 6 |
| 8 | Compression ratio | Proportion | 12:1 |
| 9 | Number of strokes per cycle | No. | 4 |
| 10 | Type of cooling | – | Water cooling |
| 11 | Ignition system (with advance/retard facility) | – | Cam shaft and contact breaker |
| 12 | Ignition timing | Degree | 28° BTDC |
| 13 | Air to fuel ratio (in producer gas) | Proportion | 1.2:1 |
| 14 | Alternator efficiency | Fraction | 0.85 |
| 15 | Specific fuel consumption at peak load (producer gas) | $Nm^3\ kWh^{-1}$ | 3.4 |
| 16 | Specific fuel consumption at peak load (fuel wood) | $kg\ kWh^{-1}$ | 1.2 |

### 3.1. The engine specifications

In the present study, a natural gas engine was procured with a design capacity 75 $kW_e$ power (after considering about 15% of de-rating in the design capacity) and was operated after suitable modifications, to run on 100% producer gas. The hydraulic governor, installed to control the natural gas engine, was fine tuned to maintain the engine speed at 1500 RPM (revolution per minute), while running on producer gas. The ignition system of the engine is activated by a contact-breaker and a cam-shaft connected to the crank shaft. The design configuration of the engine used in this study is presented in Table 3.

### 3.2. Engine manifold

The producer gas composition and stoichiometric air requirement of the producer gas is entirely different from that of natural gas. Generally, natural gas is supplied to the engine from a high pressure cylinder using pressure regulators. The inlet pressure is positive in case of natural gas engines. In most of the cases, the producer gas pressure at the inlet of the engine is below atmospheric pressure. Gas to air ratio of producer gas is 1:1.2 whereas gas to air ratio for natural gas is 1:13.5. Due to variations in inlet pressure, heating value of the fuel and stoichiometric air requirement a fuel intake manifold was designed to operate the engine on 100% producer gas. The fuel mixture intake manifold was designed to supply the appropriate fuel mixture of air and gas required to the engine according to the variation in the operating load conditions. A diagram of the fuel intake manifold designed to operate the engine on 100% producer gas is shown in Fig. 2. From Fig. 2 it may be noted that the manifold comprises of two separate chambers for entry of gas and air into the engine. The intake manifold is provided with a mixing chamber to produce a homogeneous mixture of producer gas and air. Individual valves were installed at the inlet of the manifold to maintain the required gas to air ratio. The outlet of the designed fuel intake manifold has a venturi arrangement for achieving a homogeneous fuel mixture. The outlet of the manifold is connected to the engine inlet through a throttle valve. The hydraulic governor connected to the throttle valve, controls the flow of fuel mixture into the engine according to the operating load of the engine.

### 3.3. Engine governor

The producer gas engine was designed to run at a speed of 1500 RPM generating three phase power at 415 V, at a frequency of 50 Hz. A hydraulic governor installed in the natural gas engine was retained to control the engine speed when operating on 100% producer gas. It was connected to the throttle valve located at the outlet of the designed fuel intake manifold. The engine speed tends to reduce as load increases and when the engine speed reduces the hydraulic governor opens the throttle valve to a certain degree to increase the flow of fuel mixture into the engine and to maintain the engine at the design speed. When there is a reduction in the load the speed of the engine tends to increase and the governor will reduce the flow of fuel mixture into the engine by closing the throttle valve to a certain degree. The link of the governor and the throttle valve was tuned to run the producer gas engine at the required engine speed of 1500 RPM when running on 100% producer gas.

## 4. Reactor design

A downdraft gasifier with a gasification reactor having a SGR (specific gasification rate) of 0.2 $Nm^3\ cm^{-2}\ h^{-1}$ was used to generate the producer gas from fuel wood. The rector of the gasifier consisted of multiple layers of insulation to reduce heat loss from the reactor. Hot air was supplied to the reactor for obtaining good quality producer gas with low impurities. A shell and tube heat exchange was installed to the downdraft gasifier system to use the sensible heat in the energy available from the hot producer gas obtained from the reactor and was used to heat the air supplied for gasification. A shell and tube heat exchanger was used to generate hot air. The hot air temperature was at 260 °C, when the hot gas temperature was at 610 °C. The reactor was working at a cold gas efficiency of 88%. The heating value of the producer gas generated was 5.6 MJ $Nm^{-3}$. The gas quality and cold gas efficiency of the producer gas vary according to the design specifications of the reactor. Nitrogen content in the producer gas is one of the factors influencing the heating value of the producer gas. Air is the most widely used as oxidant for gasification of biomass to avoid the requirement for production of oxygen. Use of air for biomass gasification results in production of producer gas with low heating value to the order of 4 MJ $Nm^{-3}$–6 MJ $Nm^{-3}$ [40]. The nitrogen content in producer gas varies from 51% to 59% when using olive, peach and pine [41]. The calorific value of the producer gas produced from these materials varies from 3.6 to 4 MJ $Nm^{-3}$. The

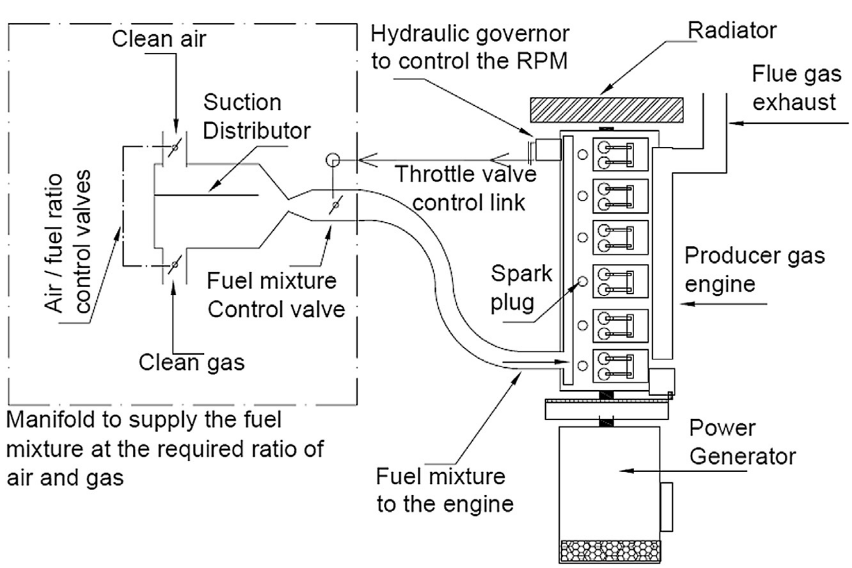


**Fig. 2.** A schematic diagram of the fuel intake manifold with the producer gas engine.

present gasifier system used in this study generates producer gas having a calorific value of 5.6 MJ $Nm^{-3}$.

### 4.1. ER (equivalence ratio)

The ER (equivalence ratio) is an important factor influencing the CV (calorific value) of the producer gas and hence, the efficiency of the engine. ER is the ratio of the quantity of the air supplied for gasification of fuel wood to the quantity of air required for the complete combustion of the fuel wood. Higher ER will result into higher nitrogen content and lower heating value of the producer gas. Whereas, lower ER will result into lower nitrogen content, this will lead to higher heating value of producer gas. Reduction in the ER bellow the required level will lead to reduction in the reactor temperature and can lead to increase the impurities like tar. It is essential to ensure that the ER is at optimum level to ensure the gas quality. In the present system, the producer gas generated with a high heating value of 5.6 MJ $Nm^{-3}$ at an ER of 0.35. The heating value of 5.3 MJ $Nm^{-3}$ of producer gas was reported at an ER of 0.32, in Ref. [42].

### 4.2. Charcoal recovery, in the ash pit

Ash removal system influences the cold gas efficiency of the gasifier system to a large extent. The ash removal system needs to be designed in such a way that it removes only the ash from the reactor while retaining the charcoal in the reactor. At least 3 kg of fuel wood is required to produce one kg of charcoal. It indicates that for every one kg of charcoal removed from the reactor the fuel wood consumption increases by 3 kg. Removal of charcoal from the reactor can be estimated by the charcoal recovered from the ash pit. Increase in charcoal recovery rate from the ash pit will decrease the cold gas efficiency proportionately. An efficient ash removal system was introduced in the present gasifier system to removes only the ash from the reactor and thus, ensures complete gasification of charcoal. When the system was operated at 85 $\pm$ 6 $kW_e$ the wood consumption rate was 110 $\pm$ 10 kg [24]. At this fuel consumption rate, the charcoal and ash extraction was in the range of 3.5–4.5%. The gasification efficiency is reported as 40–52% with a charcoal carry-over of 14–28% [43]. The present system works with a cold gas efficiency of 88% when the ash removal rate was less than 1%.

## 5. Results and discussion

The results obtained during the experiment of operating 100% producer gas engine are discussed in the following chapters. The engine's performance was evaluated at full load and part load operating conditions. Published references on performance of 100% producer gas engines at variable load conditions were found to be rare. The published results having the performance of producer gas engines operated at maximum load were compared with the performance results of the present producer gas engine operated at its maximum load. The performance of the present producer gas engine is also compared with the performance of diesel engines and natural gas engines at different load conditions. The relationship between energy content in the fuel mixture, adiabatic flame temperature, volumetric efficiency, expansion ratio, compression ratio and power generation efficiency was analyzed in the context of the performance efficiency of the producer gas engine.

### 5.1. The reactor's performance

The calorific value of the producer gas was obtained by measuring the calorific value of combustible gases constituted in the producer gas. In the present study, the energy content of the producer gas generated was 5.6 MJ $Nm^{-3}$. The cold gas efficiency of the gasification reactor of the present system was 88% at an ER of 0.35. At an ER of 0.38 the maximum heating value of the producer gas obtained was 5.6 MJ $Nm^{-3}$ [44]. During gasification of woodchips a cold gas efficiency of 76.7% was obtained with an ER of 0.29 [45]. The cold gas efficiency of 69–72% was obtained at an ER of 0.3–0.35 [46]. An ER of 0.2–0.4 was observed for gasification of biomass [47]. The calorific value of producer gas obtained was 5.35 MJ $Nm^{-3}$ at an ER of 0.28 [48]. The calorific value of the producer gas was 5.6 MJ $Nm^{-3}$ at an ER of 0.35 [48]. It may be noted that the gasifier system used for the present study works at an ER of 0.35 and generates the producer gas with the calorific value of 5.6 MJ $Nm^{-3}$. In the present system, higher cold gas efficiency was achieved using multi-layer insulation in the reactor, by supplying hot air into the reactor by recovering waste heat from the hot producer gas drawn from the reactor and reducing the charcoal falling rate into the ash pit. The present system was operated with reduction in nitrogen content in the producer gas and increasing the heating value of the gas to 5.6 MJ $Nm^{-3}$. The increase in calorific value of the producer gas was obtained by minimizing heat loss from the reactor and injecting hot air into the reactor, at an ER of 0.35. Reducing the charcoal yield rate in the ash pit using an efficient ash removal system also contributed to increase the cold gas efficiency of the system to 88%.

Equivalence ratio and charcoal return into the ash pit have a strong influence on the cold gas efficiency of the gasification system. It is important to have an appropriate ER to convert the biomass into producer gas. Also, it is important to ensure that maximum amount of charcoal is converted into combustible gas and reduce the falling as charcoal into the ash pit. The ER, calorific value of the producer gas, quantity of ash return rate (along with charcoal) and cold gas efficiency of the present system were compared with that of other systems. The comparison of the performance analysis of the present system and other systems referred is presented in Table 4. It may be noted from Table 4, in Ref. [42] the cold gas efficiency is reduced due to the change in ER and higher charcoal return rate. Similarly, in Ref. [43] the cold gas efficiency was low due to high rate of charcoal return. Ash removal rate was not reported in Refs. [44–46]. A cold gas efficiency of 78% is reported in Ref. [44], which is 10% lower than the present system. A cold gas efficiency of 77% is reported in Ref. [45], which is 11% lower than the present system. A cold gas efficiency of 72% is reported in Ref. [46], which is 16% lower than the present system.

#### 5.1.1. Components of the producer gas

The heating value of producer gas mainly depends upon the presence of combustible gases in the producer gas. The components of the producer gas are influenced by factors such as ER, reactor temperature and efficient ash removal system. A high ER will intensify the combustion rate and will increase $CO_2$ while a low ER will result in low gasification efficiency and will lead to production of more charcoal. Both these scenarios will lead to a reduction in heating value of the producer gas. In the present study, a reactor with multi-layers of insulation was operated at an ER of 0.35 and at temperature in the range of 1100–1150 °C. A higher reactor temperature enhances the reaction rate and increases the production of combustible gases.

The components of the producer gas generated from the gasifier were analyzed using a gas chromatograph. The results of the producer gas analysis are presented in Table 5. The components of the producer gas presented in Table 5 are on dry basis. The moisture present in the producer gas was removed using two chillers connected in series. From Table 5, it may be noted that the combustible components of the producer gas constitutes 23% of $H_2$, 21% of CO and 0.9% of $CH_4$. The non-combustible components of the producer

**Table 4**
A comparison of the performance results of the gasifier systems.

| S. No. | Design parameters | Unit | Present study | Ref. [24] | Ref. [42] | Ref. [43] | Ref. [44] | Ref. [45] | Ref. [46] | Ref. [48] |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | Equivalence ratio | Mass fraction | 0.35 | na | 0.32 | 0.25 | 0.38 | 0.29 | 0.35 | 0.29 |
| 2 | Calorific value of the gas | MJ $Nm^{-3}$ | 5.6 | 4.5 | 5.3 | 3.9 | 5.6 | 5.2 | 4.8 | 5.5 |
| 3 | Ash removal rate | % of mass fraction | 1.0 | 4.0 | 4.6 | 21 | na | na | na | na |
| 4 | Cold gas efficiency | % of energy fraction | 88.0 | 77.0 | 52 | 49.1 | 77.97 | 76.7 | 72.0 | 57.9 |

na: data relating to parameters are not available.

gas constitute 9% of $CO_2$ and 46% of $N_2$. Among the combustible gases of the producer gas, $CH_4$ has a higher calorific value, but its presence is very little in percentage. The CV (calorific value) of CO is 12.7 MJ $Nm^{-3}$, CV of $CH_4$ is 36 MJ $Nm^{-3}$ and the CV of $H_2$ is 11.0 MJ $Nm^{-3}$. According to the contribution of the combustible gas components the calorific value of the producer gas is 5.6 MJ $Nm^{-3}$. From Table 5, it may be noted that, only 45% producer gas comprises of combustible gases and 55% as non-combustible components. Hence, producer gas is known as lean gas or low calorific value gas. Since, major portion of producer gas is non-combustible; the producer gas engines are operating at a de-rated design capacity. The 55% of the non-combustible gas present in producer gas affects the engine's efficiency in two ways. When using the producer gas for operating engines, most of the heat generated during combustion phase is absorbed by the non-combustible components of the producer gas itself. These non-combustible components lower the energy density of producer gas and reduce the adiabatic flame temperature. Reduction in adiabatic flame temperature results in low expansion ratio of the fuel mixture, which reduces the efficiency of the engine. When the moisture content in the fuel wood was in the range of 10–15%, the calorific value of the producer gas was 4.5 $\pm$ 0.3 MJ $kg^{-3}$ [24]. The producer gas from the present system has a calorific value of 5.6 MJ $Nm^{-3}$. The calorific value of 5.6 MJ $Nm^{-3}$ was achieved by using multi-layer insulation in the reactor and supply of hot air for gasification.

### *5.2. Adiabatic flame temperature*

Adiabatic flame temperature is one of the important factors which influence the expansion ratio of a fuel mixture and efficiency of the engine. Producer gas has a peak flame temperature of 1800 K while the natural gas has a peak flame temperature of 2210 K, at an air gas ratio of 1.35 and 17.2 respectively [49]. Adiabatic flame temperature has a strong influence on the combustion pressure. The fuel mixture's expansion ratio depends on the adiabatic flame temperature. Combustion pressure is a function of expansion ratio and compression ratio. Thus, the adiabatic flame temperature of the fuel mixture is one of the factors influencing the efficiency of the producer gas engine.

### *5.3. Importance of robust gas cleaning system*

The producer gas exits the reactor at a very high temperature of 500–600 °C. Apart from a very high temperature, the producer gas exits from the reactor also contain a lot of impurities such as tar and dust. In order to use producer gas to run an IC engine, the temperature of the producer gas has to be reduced to close to ambient temperature. As engine run on high temperature producer gas will lead to reduction in efficiency of the engine performance. A high temperature fuel mixture will reduce the filling efficiency and subsequently will reduce the performance efficiency of the engine. The impurities in producer gas also need to be reduced to ensure smooth operation of the engine. For a sustainable operation of the gasifier based power plant it is essential to have a gas cleaning system which works efficiently with a low maintenance cycle. The new generation power units with high speed engines are more delicate and display higher sensitivity to impurities as compared to the old generation power units [50]. For operating IC engines using producer gas, the allowable tar level in the producer gas should be less than 30 mg $Nm^{-3}$ [50]. This report indicates a large variation in the recommendation of the allowable tar level in producer gas for running the IC engines. The recommended tar level in producer gas (based on 14 references) for a smooth operation of the IC engines varies from 10 to 100 mg $Nm^{-3}$ [50]. The maximum allowable limit for tar level in producer gas is quoted as 50 mg $Nm^{-3}$ and tar reduction is reported as the most challenging problem in gasification in Ref. [51]. After cooling the producer gas to 45 °C, by using a wet-electrostatic precipitator the tar content was brought down to less than 25 mg $Nm^{-3}$ [52].

High tar content is one of the key technical barriers for large-scale implementation of gasification technology [53]. Activated carbon was used to for an effective removal tar from producer gas [53]. The present system is developed with a primary focus on tar reduction at the reactor itself by increasing the reactor temperature. The temperature of the reactor was increased by using multiple layers of insulation and supply of hot air for gasification. In the present system venturi scrubbers, chiller, fabric filter and paper filter were used to reduce impurities in the producer gas. Tar and dust in the producer gas was estimated using the test protocol [54]. In the present system, the tar content of 350 mg $Nm^{-3}$ in the raw gas was brought down to 30 mg $Nm^{-3}$, which is in the required range of less than 50 mg $Nm^{-3}$. Similarly, the dust content was brought down from 800 mg $Nm^{-3}$ (in the raw gas) to 25 mg $Nm^{-3}$ (in the clean gas). When the moisture content of the biomass varies from 10 to 20% the particulate and tar content reaches a maximum of 1000 mg $Nm^{-3}$ [16]. An extensive gas cleaning train is used to bring down the impurities in the gas to an acceptable level for smooth operation of the IC engines. Producer gas quality is an import factor to economically run the IC engine for a longer duration. Sand filters, venturi scrubbers and catalyst crackers are referred as high efficiency equipment for tar reduction [30]. This paper also reports that wet electrostatic precipitator and fabric filters were used for efficient reduction of dust content in the producer gas. Various methods of gas cleaning and their features were compared in Ref. [55]. This paper also reports stable metals such as Ni and alkali metals such as KOH, $KHCO_3$ and $K_2CO_3$ have been examined for elimination of tar from producer gas. The cost of the catalyst, maintaining them at high temperature and obtaining sustainable efficiency are some of the challenges associated with use of such catalysts. Two-stage and multi-stage gasifiers were developed to reduce impurity level in the producer gas [33,34]. The impurities in the producer gas clog the engine valves and gas filters [40]. Besides the problem of clogging at the engine's valves, other problems faced due to impurities of the producer gas are corrosion, erosion and poisoning of the catalyst used for tar cracking [40].

There is a need for introduction of a dry gas cleaning system to minimize the amount of water required by the gas scrubbers and also to reduce the problems associated with disposal of the water used for scrubbing. About 20 $m^3$ of water needs to be treated daily when operating a 100 $kW_e$ (power) plant using wet scrubbers to

**Table 5**
Components of the producer gas.

| S. No. | Component | Volume fraction in % |
|---|---|---|
| 1 | Hydrogen | 23.0 |
| 2 | Carbon monoxide | 21.0 |
| 3 | Methane | 0.9 |
| 4 | Carbon dioxide | 9.0 |
| 5 | Nitrogen | 46.1 |

clean the producer gas [24]. This paper also reports that water circulation through the scrubber is 50 $m^3\ h^{-1}$ and about one cubic meter of water is lost in evaporation. To avoid problems related to water procurement and treatment dry gas it is important to have a reliable dry gas cleaning system to remove the impurities from producer gas. A dry gas cleaning method is adopted in the two-stage gasifier system [33]. However, reduction of tar level at the reactor itself is more preferable to avoid the load on the gas cleaning system. A robust gas cleaning system is a basic requirement for successful operation and maintenance of a gasifier based power plant.

### 5.4. Performance of the producer gas engine

In the present study a natural gas engine having a design capacity of 133 hp was used to operate on 100% producer gas. The design capacity of the engine's power output in natural gas mode was 85 kW, with an alternator efficiency of 85%. The maximum power output obtained during the experiment was 73 $kW_e$, when operating the engine on 100% producer gas. About 12.4% de-rating was observed when the engine was operated on producer gas when compared to the design capacity of the engine (when operating on natural gas). The producer gas engine delivers about two-third of the power at its maximum load as compared to the performance of engines using liquid fuel [56]. Reduction in energy density and a reduction in compression ratio result in 20% de-rating of the engine as compared to operating it on diesel. While operating an engine on producer gas de-rating from the designed power rating was in the range of 40–50% when the CR was 7:1 and 20% when the CR was 11:1 [57]. With the compression ratio of 10:1–12:1 the theoretical efficiency of an engine works out to be is 60–63% [39]. This report also indicates that in normal operating conditions the volumetric filling efficiency of the fuel mixture is in the range of 70–90%. A reduction in filling efficiency reduces combustion pressure. These two factors result in de-rating of the engine when operating on producer gas.

In the present study, an overall efficiency of 21% was achieved with the compression ratio of 12:1. The maximum power generation efficiency of 21% was achieved at 85% load. An overall efficiency of 21% was achieved at the highest compression ratio of 17:1 [58]. It may be noted in the present study that an overall efficiency of 21% was achieved at a CR of 12:1 at 85% load. This is due to other factors like energy content of fuel mixture, volumetric efficiency, etc. An engine's efficiency of 29% and an overall electrical efficiency of 25% were observed with a two stage gasifier system [59]. A diesel engine operating at 30–35% efficiency will operate at 25–30% efficiency when operating on producer gas [17]. A 70 $kW_e$ producer gas engine was operated at its maximum load with an overall power generation efficiency of 16.1% [60]. In the present research work, the performance of the producer gas engine was studied at variable operating load conditions along with an analysis of the fuel input and the power output. The results obtained during the performance analysis of the producer gas engine based power generation system are presented in Table 6. From Table 6, it may be noted that, the producer gas engine was operated at variable load conditions from 7 to 97% of the design capacity and the efficiency of the engine varies in the range of 3.1–20.7%. There is a large variation in the specific fuel consumption rate ranging from 7.9 to 1.2 kg $kWh^{-1}$. The fact is, at an overall efficiency of 20.7% the specific fuel consumption rate was 1.2 kg $kWh^{-1}$ and about 80% of the energy from fuel wood goes un-utilized. There is a need to improve the system to generate producer gas with high calorific value gas without adding much complications and cost. From Table 5, it may be noted that about 55% of the producer gas is non-combustible component and hence, $N_2$ and $CO_2$ in the producer gas need to be reduced to the most possible level. Improving the existing efficiency status of the producer gas

**Table 6**
Performance results of the producer gas engine.

| S. No. | Load $kW_e$ | Load % | Fuel wood consumption kg $h^{-1}$ | Gas flow rate $Nm^3\ h^{-1}$ | SFC[a] (fuel wood) kg $kWh^{-1}$ | SFC[a] (producer gas) $Nm^3\ kWh^{-1}$ | Efficiency % |
|---|---|---|---|---|---|---|---|
| 1 | 5 | 7 | 39.5 | 112 | 7.9 | 22.4 | 3.1 |
| 2 | 9 | 12 | 41.5 | 118 | 4.6 | 13.1 | 5.2 |
| 3 | 10 | 13 | 41.8 | 119 | 4.2 | 11.9 | 5.8 |
| 4 | 13 | 17 | 47.3 | 134 | 3.6 | 10.3 | 6.7 |
| 5 | 15 | 20 | 47.3 | 134 | 3.2 | 9.0 | 7.7 |
| 6 | 20 | 27 | 54.0 | 153 | 2.7 | 7.7 | 9.0 |
| 7 | 23 | 31 | 55.2 | 157 | 2.4 | 6.8 | 10.1 |
| 8 | 26 | 35 | 58.8 | 167 | 2.3 | 6.4 | 10.7 |
| 9 | 33 | 44 | 67.6 | 192 | 2.0 | 5.8 | 11.8 |
| 10 | 44 | 59 | 72.2 | 205 | 1.6 | 4.7 | 14.8 |
| 11 | 53 | 71 | 79.5 | 226 | 1.5 | 4.3 | 16.1 |
| 12 | 57 | 76 | 81.7 | 232 | 1.4 | 4.1 | 16.9 |
| 13 | 58 | 77 | 82.6 | 235 | 1.4 | 4.0 | 17.0 |
| 14 | 59 | 79 | 84.1 | 239 | 1.4 | 4.0 | 17.1 |
| 15 | 65 | 87 | 80.3 | 228 | 1.2 | 3.5 | 19.6 |
| 16 | 66 | 88 | 78.5 | 223 | 1.2 | 3.4 | 20.4 |
| 17 | 67 | 89 | 79.7 | 226 | 1.2 | 3.4 | 20.6 |
| 18 | 68 | 91 | 79.6 | 226 | 1.2 | 3.3 | 20.7 |
| 19 | 69 | 92 | 81.4 | 231 | 1.2 | 3.4 | 20.5 |
| 20 | 70 | 93 | 83.3 | 237 | 1.2 | 3.4 | 20.4 |
| 21 | 71 | 95 | 85.3 | 242 | 1.2 | 3.4 | 20.2 |
| 22 | 72 | 96 | 85.3 | 242 | 1.2 | 3.4 | 20.4 |
| 23 | 73 | 97 | 86.9 | 247 | 1.2 | 3.4 | 20.4 |

[a] SFC: specific fuel consumption rate.

engine is essential for large scale dissemination of gasification technology.

### 5.5. Factors influencing the efficiency of the producer gas engine

The key factors influencing the efficiency of the producer gas engine are discussed in this chapter. The factors considered for analyzing the efficiency of the producer gas engine are compression ratio, air fuel ratio, the energy density of the fuel mixture, Wobbe number, expansion ratio of the fuel mixture and combustion pressure.

#### 5.5.1. Compression ratio

The CR (compression ratio) of the engine varies depending upon the fuel to be used, manufacturers' design and the model of the engine. The producer gas engine used in this study was having a CR of 12:1. The power generation efficiency was obtained as 21% with a fuel mixture expansion ratio of 6. It is important to note that the expansion ratio was only 50% of the compression ratio.

The diesel engines work with a CR of 17:1 and the upper limit for CR of natural gas engines is 15.8:1. Engines with higher compression ratio are difficult to start and result in increased wear and tear. The efficiency of the engine decreases about 32% when operating a diesel engine on producer gas [58]. Most of the natural gas engines work with the CR in the range of 10:1–12:1 [28]. The efficiency of the engine decreases due to increase in heat loss, friction and cylinder pressure at the time of opening of the exhaust valve [61]. It may be noted that even at a CR of 10:1 the engine efficiency can go up to 34% when the expansion ratio is 10:1 [61]. The CR of 10:1 was said to be technically and economically viable [13].

A 20 kW diesel engine was modified with spark ignition for operation on producer gas with high compression ratio [62]. Experiments were conducted on this engine with the same generator with a CR of 18.5:1 and 9.5:1. The engine having CR of 18.5:1 was operating with efficiency in the range of 30–35%. The engine having CR of 9.5:1 was operating with efficiency in the range of 25–30%. As producer gas engines can work with higher CR, the natural gas engines normally designed with a CR of 12:1 and with a SI (spark ignition) system can be readily operated on 100% producer gas. The Highest efficiency of power generation observed using small scale gas engines was 28% [63].

Apart from CR other factors influencing the performance efficiency of an IC engine are heat loss, friction and expansion ratio [56,61]. In the present study several factors like energy content of the fuel mixture, energy density of the fuel mixture, adiabatic flame temperature, volumetric efficiency, expansion ratio of the fuel mixture and compression ratio of the engine were analyzed in detail for identifying their impact on the performance of the engine.

#### 5.5.2. Air fuel ratio

Air fuel ratio is an important factor which influences the efficiency of an engine. Air fuel ratio below stoichiometric will result in incomplete combustion of the producer gas and hence, lower efficiency. Air fuel ratio above stoichiometric will lead to absorption of energy by the excess air and will result in lower efficiency. So, it is important to supply a limited quantity of excess air (above stoichiometric) to ensure complete combustion and maximum efficiency. In the present study the producer gas engine was operated at a stoichiometric air fuel ratio of 1.2:1. Stoichiometric air fuel ratio of 1.2 and 1.3 was reported for producer gas having a heating value of 5.6 and 6.0 MJ $Nm^{-3}$ [59]. With a combination of natural gas and producer gas, the peak efficiency of the engine was reported when the air fuel ratio was in the range of 1.2–1.3 [64]. Stoichiometric air fuel ratio ($\lambda$) in the range of 1.2–2.8 is recommended for operation of producer gas engine without any significant reduction in the efficiency [59]. The intake manifold has a direct influence on improving the intake aerodynamics of the fuel mixture and combustion [65]. It is important to supply the fuel mixture at the required quantity with an appropriate ratio of gas and air at variable load operating conditions. A carburetor (fuel intake manifold) needs to be introduced to supply an appropriate air fuel mixture to the engine when modifying the diesel engine or natural gas engine to run on producer gas. A carburetor was designed and installed to have an appropriate supply of air fuel mixture by distributing the engine suction to the required proportion. An oxygen sensor was introduced at the exhaust to control an activator installed at the throttle valve to maintain the required air fuel ratio [62]. In the present system the air fuel ratio was maintained by distributing the engine suction and tuning the throttle valves provided at the air and producer gas intake. According to the load conditions, the flow rate of the air and gas mixture was controlled automatically by controlling a throttle valve linked to a hydraulic governor.

#### 5.5.3. The energy density of the fuel mixture

The energy density of the fuel mixture is an important factor which influences the power output of the engine. Producer gas having a heating value of 5.2 MJ $Nm^{-3}$ would result in an energy density of 2.3 MJ $Nm^{-3}$ [66]. This paper reports that the energy density of the fuel mixture of diesel and natural gas is 2.8 MJ $Nm^{-3}$ and 3.0 MJ $Nm^{-3}$ respectively. In the present system the calorific value of the producer gas produced is 5.6 MJ $Nm^{-3}$ and the energy density of the fuel mixture works out to be 2.55 MJ $Nm^{-3}$. These numbers indicate that the energy density of the producer gas and air mixture is 15% lower than diesel and 8.9% lower than Natural gas.

Through steam gasification, the producer gas heat content was raised to 15.69 MJ $Nm^{-3}$ [67]. In air gasification the calorific value of the gas lies in the range of 4–6 MJ $Nm^{-3}$. The adiabatic flame temperature of producer gas is 1750 K for air gasification and 2200 K for gas obtained through steam gasification [67]. It may be noted the adiabatic flame temperature of the producer gas generated through steam gasification is closer to the adiabatic flame temperature of natural gas. This indicates steam gasification can improve the energy density of the producer gas mixture by 20%. Increasing the energy density can increase the engine efficiency which is proportional to the increase in hydrogen content and a reduction in nitrogen content in the producer gas.

#### 5.5.4. Wobbe number

Energy density and Wobbe number are interrelated. A relation between the energy density and Wobbe number can be defined as shown in Eq. (9). When the heating value of natural gas and producer gas is 35.90 MJ $Nm^{-3}$ and 6.47 MJ $Nm^{-3}$, the Wobbe number is 53.66 MJ $Nm^{-3}$ and 7.94 MJ $Nm^{-3}$ respectively [64].

In the present study the calorific value of the producer gas generated from the gasifier system is 5.6 MJ $Nm^{-3}$. The density of the producer gas was estimated at 1.05 kg $Nm^{-3}$ based on the result of the gas components. According to the calorific value and the density of the producer gas generated in the present system the Wobbe number works out to be 5.5 MJ $Nm^{-3}$ (estimated as shown in Eq. (9)). It may be noted that the Wobbe number of producer gas is 6.6 times lower than the Wobbe number of natural gas. At an air fuel ratio of 1:18, the energy density of the fuel mixture comprising of air and natural gas is 3.0 MJ $Nm^{-3}$. At an air fuel ratio of 1:18, the energy density of the fuel mixture comprising of air and diesel is 2.83 MJ $Nm^{-3}$. At an air fuel ratio of 1:1.2, the energy density of the fuel mixture comprising of air and producer gas is 2.59 MJ $Nm^{-3}$. The energy density of fuel mixture of air and producer gas is 15% less than the energy density of air and natural gas. The reduction in

Wobbe number and energy density of the producer gas results in to de-rating of the engine from its design capacity when operating a modified natural gas engine on 100% producer gas.

### 5.5.5. Expansion ratio

The expansion ratio ($E_r$) is a function of the energy density of the fuel mixture, Wobbe number, adiabatic flame temperature and volumetric filling efficiency of the engine. The expansion ratio of a fuel mixture is one of the factors which influence the efficiency of an engine. Expansion ratio is always less than the CR, particularly during part load operation of an engine. This is one of the reasons for the engines to operate with a poor efficiency at part load. Ultimately the poor efficiency leads to consumption of more fuel during part load operation as compared to consumption of fuel when operating at designed load. An engine having a CR of 10 can work with a brake thermal efficiency of 34% and 43% at the expansion ratio of 10 and 30 respectively [61]. This paper also highlights that expansion ratio such as 30 are not practical. So, there is a need for developing an engine which can work with a high expansion ratio under practically feasible conditions.

A peak pressure of 60 bar was observed while operating an IC engine on producer gas [49]. A peak pressure of $59 \times 10^5$ Pa was observed while operating an IC engine on natural gas [64]. Peak pressure is a function of CR, energy density, flame velocity and speed of the piston. When using various combinations of producer gas and natural gas, no significant change was observed either in the peak pressure or the timing of its occurrence in reference to the crank angle [64]. Maximum duration of the occurrence of the peak pressure during the power stroke is from 360° to 370° of the crank angle [64,68]. This duration corresponds to 0.006 s, at an engine speed of 1500 RPM (revolution per minute). This indicates that the engine efficiency is not only dependent on peak pressure but also on energy density and expansion ratio.

In the present study, a volumetric efficiency of 100% was achieved when the operating load of the producer gas engine was above 85%. During the full load operation when the throttle valve is at its maximum opening position the volumetric filling efficiency of the engine reaches maximum. At full load the pressure drop across the throttle valve will be at minimum and the blower just before the fuel intake manifold also contributes to the increase in volumetric efficiency. When the volumetric efficiency is 100%, the producer gas engine was having a maximum expansion ratio of 6.35. This is only 50% of the CR. At 100% filling efficiency expansion ratio of the fuel mixture comprising of natural gas and diesel are 7.34 and 7.56 respectively. The expansion ratio of the fuel mixture comprising of producer gas is 14% lower than the fuel mixture comprising of natural gas. Low energy density of the producer gas fuel mixture results in lower adiabatic flame temperature. The reduction in adiabatic flame temperature results in lower expansion ratio of the fuel mixture. When the engine was operating at and above 80% of the designed load, the expansion ratio was 5.9 and when the engine was operating at the minimum load of 3% the expansion ratio was only 2.7. The efficiency of the engine corresponding to the expansion ratio of 5.9 and 2.7 was 20.4% and 3.07% respectively. These values indicate that the expansion ratio has a strong influence on the efficiency of IC engines.

### 5.5.6. Combustion pressure

In addition to the high CR, the combustion pressure is also an important factor for achieving higher efficiency of the engine. A gasoline engine can produce a combustion pressure of 50 kg $cm^{-2}$ at a CR of 6:1 with the same CR a wood gas engine can produce a combustion pressure of 27 kg $cm^{-2}$ [17]. The energy density of gasoline fuel mixture is 50% higher than the energy density of producer gas mixture. Accordingly, the combustion pressure of gasoline engines is 85% higher than the producer gas engine for the same CR of 6:1. The engine efficiency is directly proportional to the combustion pressure generated during the power stroke. An engine with a CR of 8.2:1 can produce 6.7 kW when running on gasoline and produces 2.3 kW when running on producer gas [69]. It may be noted when the gasoline engine is operated with producer gas it produces less than one third of the rated power, which corresponds to 50% reduction in combustion pressure. The engine operating on producer gas can reach a peak pressure of 60 bar with CR of 12:1 [49,58]. This is the reason for using producer gas as a fuel for operating diesel engines or natural gas engines.

It may be noted that during the power stroke, the combustion pressure remains around the value of the peak pressure for a maximum duration of 0.006 s [49,58]. This corresponds to 8.4% of the total duration of the power stroke. So in practice the cylinder is at the peak pressure for only 8.4% of the power stroke. The pressure in the cylinder remains low in the rest of 91.6% duration of the power stroke. The energy generated is a function of power and time. The short duration of the peak pressure is one of the reasons for engines to operate at low efficiency in the range of 25–30%.

### 5.6. The fuel intake and part load operation

Fuel intake manifold (the carburetor) is an important component which is also responsible for the engine's efficiency. The volumetric filling efficiency of the engine can be affected by the pressure drop across the fuel intake manifold. The manifold controls the air fuel ratio as inappropriate stoichiometric air supply and non-homogeneous fuel mixture will affect the combustion pressure. The manifold also enhances the uniformity in the gas and air mixture for increased combustion efficiency. All these factors influence the efficiency of an engine. Experimental results reports that a well-designed manifold can increase the maximum power output of the engine by 25% [39]. The manifold designed for the supply of air and gas mixture to the IC engine was tested at variable load conditions. The air flow and gas flow were controlled by a single throttle valve during the variable load condition. The throttle valve was linked to a hydraulic governor which maintains the engine speed at a constant RPM irrespective of the operating load conditions.

The amount of the fuel mixture actually entering the cylinder of an engine is determined by the inlet pressure of the gas and cylinder volume [39]. However this statement stands true for the operating condition of the engine at the maximum load only. In part load operating conditions of the engine the intake of the fuel mixture is controlled by creating a resistance through a throttle valve. During part load operation the fuel intake was decreased and

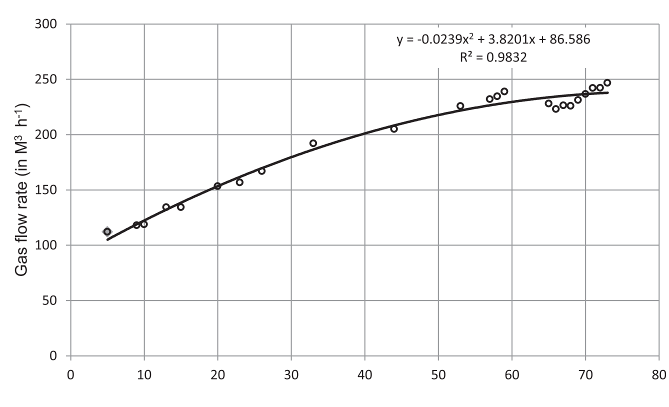


**Fig. 3.** Gas flow rate at variable load conditions.

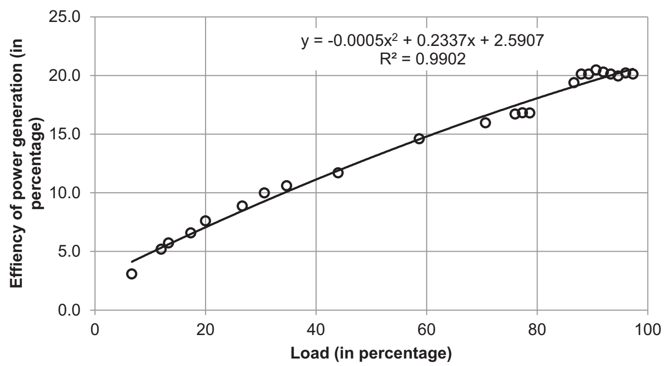


**Fig. 4.** Power generation efficiency of the producer gas engine at variable load conditions.

subsequently the expansion ratio and peak pressure were reduced. In practical scenarios, the majority of the engines are operating at part load only. When the cylinder volume is constant, the volume of the fuel mixture varies depending on the operating load. There is a variation in the volumetric filling efficiency according to the variation in the operating load of the engine. Volumetric filling efficiency is the ratio of the swept volume and the volume of the fuel intake. The volumetric efficiency can be calculated using Eq. (3).

The swept volume remains constant irrespective of the operating load. But the intake volume of the fuel mixture varies according to the variation in the operating load. The constant swept volume of the cylinder and variations in the intake volume of the fuel mixture results in variation in the expansion ratio. The expansion ratio can be estimated by using Eq. (4). Variation in the expansion ratio results in variation in the efficiency of the engine. Reduction in the fuel intake will result in a reduction in expansion ratio and consecutively on the engine's efficiency. The gas flow rate into the engine at variable load conditions is shown in Fig. 3. The power generation efficiency of the producer gas engine at variable load conditions is shown in Fig. 4. The reduction in efficiency along with the reduction in operating load can be seen in Fig. 4. A reduction in efficiency results in increase of specific fuel consumption.

According to the profile of the power generation efficiency (as shown in Fig. 4) it may be concluded that the engine should be operated at least above 30% of the design capacity to have a better efficiency. When operating the producer gas engine at a lower capacity, the engine operates at a lower efficiency. The gas flow rate of the engine reduces when it operates at a lower capacity and the producer gas produced from the gasification reactor has high impurities (due to a reduction in the fuel burning rate/gas flow rate). Operating load of the engine is proportional to the gas flow rate influencing the reactor's performance. A minimum turn down ratio of 1:3 is reported for obtaining a good quality gas which can be used to operate the IC engines [19]. A turn down ratio of 1: 3 means that the gasifier should be operated at least above one third of its design capacity for obtaining good quality gas. Apart from reduction in gas quality, the efficiency of the engine also reduces to a large extent when it is operated below 30% of the design load. So, it is better to operate the engine above 80% of the design load for production of a good quality gas and obtaining better efficiency.

*5.6.1. Efficiency of the engine based on $V_e$, $E_r$ and $C_r$ at variable load conditions*

The volumetric efficiency ($V_e$), expansion ratio ($E_r$) and the compression ratio ($C_r$) are the factors influencing the efficiency of an internal combustion engine. The efficiency of the IC engine by $V_e$, $E_r$ and $C_r$ can be estimated using the Eq. (7). Here, $V_e$, $E_r$ and $C_r$ are functions of the engine design and fuel characteristics. The efficiency of the engine estimated based on $V_e$, $E_r$ and $C_r$ is named as efficiency by method I. The efficiency estimated by energy input and output is named as efficiency by method II. The detailed comparison of the engine's performance along with the efficiency estimated based on both the methods are presented in Table 7. It may be noted from Table 7, the producer gas engine was operated at a variable load condition in the range of 5–73 $kW_e$. The overall efficiency of power generation was varying from 3.1 to 20.7% when varying the load from minimum to maximum capacity. The

**Table 7**
Power generation efficiency of the producer gas engine obtained by two different methods.

| S. No. | Load ($kW_e$) | Load (%) | Gas flow rate ($Nm^3\ h^{-1}$) | Air flow rate ($Nm^3\ h^{-1}$) | Maximum expansion of the fuel mixture ($m^3$) | Volumetric efficiency ($V_e$) (%) | Expansion ratio ($E_r$) (volume fraction) | Efficiency obtained by $V_e$, $E_r$, $C_r$ (%) | Efficiency obtained by energy input and output (%) |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 5 | 7 | 112 | 135 | 1465 | 0.45 | 2.69 | 4.5 | 3.1 |
| 2 | 9 | 12 | 118 | 142 | 1542 | 0.48 | 2.83 | 5.0 | 5.2 |
| 3 | 10 | 13 | 119 | 143 | 1553 | 0.48 | 2.85 | 5.1 | 5.8 |
| 4 | 13 | 17 | 134 | 161 | 1756 | 0.54 | 3.22 | 6.5 | 6.7 |
| 5 | 15 | 20 | 134 | 161 | 1756 | 0.54 | 3.22 | 6.5 | 7.7 |
| 6 | 20 | 27 | 153 | 184 | 2004 | 0.62 | 3.68 | 8.5 | 9.0 |
| 7 | 23 | 31 | 157 | 188 | 2049 | 0.63 | 3.76 | 8.9 | 10.1 |
| 8 | 26 | 35 | 167 | 200 | 2183 | 0.67 | 4.00 | 10.1 | 10.7 |
| 9 | 33 | 44 | 192 | 230 | 2509 | 0.78 | 4.60 | 13.3 | 11.8 |
| 10 | 44 | 59 | 205 | 246 | 2679 | 0.83 | 4.92 | 15.2 | 14.8 |
| 11 | 53 | 71 | 226 | 271 | 2952 | 0.91 | 5.42 | 18.4 | 16.1 |
| 12 | 57 | 76 | 232 | 278 | 3032 | 0.94 | 5.56 | 19.4 | 16.9 |
| 13 | 58 | 77 | 235 | 282 | 3066 | 0.95 | 5.63 | 19.8 | 17.0 |
| 14 | 59 | 79 | 237 | 284 | 3096 | 0.96 | 5.68 | 20.2 | 17.1 |
| 15 | 65 | 87 | 228 | 274 | 2980 | 0.92 | 5.47 | 18.7 | 19.6 |
| 16 | 66 | 88 | 223 | 268 | 2915 | 0.90 | 5.35 | 17.9 | 20.4 |
| 17 | 67 | 89 | 224 | 269 | 2928 | 0.90 | 5.37 | 18.1 | 20.6 |
| 18 | 68 | 91 | 226 | 271 | 2953 | 0.91 | 5.42 | 18.4 | 20.7 |
| 19 | 69 | 92 | 231 | 278 | 3023 | 0.93 | 5.55 | 19.3 | 20.5 |
| 20 | 70 | 93 | 237 | 284 | 3093 | 0.96 | 5.67 | 20.2 | 20.4 |
| 21 | 71 | 95 | 242 | 291 | 3164 | 0.98 | 5.81 | 21.1 | 20.2 |
| 22 | 72 | 96 | 242 | 291 | 3167 | 0.98 | 5.81 | 21.2 | 20.4 |
| 23 | 73 | 97 | 247 | 296 | 3224 | 1.00 | 5.92 | 21.9 | 20.4 |

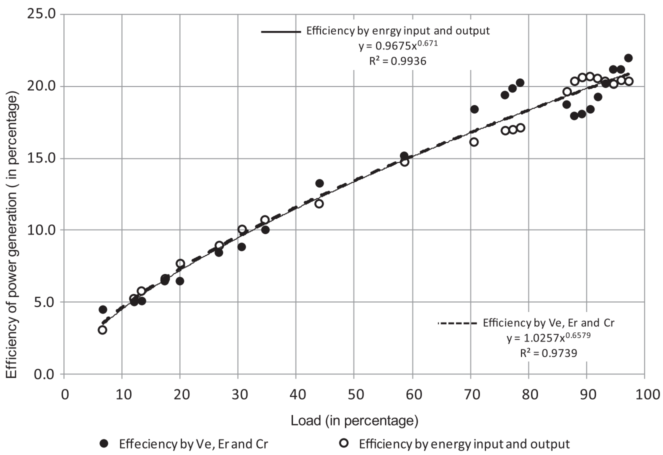


**Fig. 5.** Power generation efficiency of the engine obtained by two different methods.

volumetric filling efficiency of the engine goes as low as 45% during the minimum load conditions. The volumetric filling efficiency of the engine reaches 100% during maximum load conditions. The results indicate that the engine works at poor efficiency in part load conditions. Appropriate selection of the system capacity is essential to prevent the engine from running at part load conditions which results in low efficiency. Reduction in filling efficiency and low calorific value of producer gas results in reduction in expansion ratio of the fuel mixture. Reduction in expansion ratio results in combustion pressure and hence, decreases the engine efficiency. It may be noted from Table 6, the efficiency of the engine increases along with the increase in expansion ratio.

The power generation efficiency estimated by method I (using the volumetric efficiency, expansion ratio and the compression ratio) and the efficiency estimated by method II (using energy input and output) are presented in Fig. 5. The power generation efficiency of the engine was estimated based on 85% conversion efficiency of the shaft power into electrical power. In Fig. 5, it may be noted that there is not much variation between the trend lines representing the efficiency of the engine at different load conditions estimated by both the methods. From Fig. 5, it may be noted that the correlation coefficient of the efficiency estimated by method I using $V_e$, $E_r$ and $C_r$ and the efficiency estimated by method II using energy input and output is 0.99. The variation of 1% could be due to variation in air fuel ratio during part load conditions. The efficiency estimated by method I using $V_e$, $E_r$ and $C_r$ takes into consideration of all key parameters responsible for engine's performance and can be used to predict the energy efficiency at any given load condition. The relation between the engine efficiency estimated based on $V_e$, $E_r$ and $C_r$ (as given in Eq. (7)) is verified by comparing the efficiency estimated by energy input and output. It may be noted from Fig. 5, the trend lines formed by the efficiency values estimated by both the methods have no significant variation.

### 5.7. SFC (specific fuel consumption)

The gasification rate from fuel wood to producer gas was 2.8 $Nm^3\ kg^{-1}$. Fuel wood to gas conversion rate was arrived by measuring the gas flow rate and fuel consumption rate. The specific fuel consumption rate (fuel wood) for power generation was 1.2 $kg\ kWh^{-1}$, when the engine was operated at above 85% load. The specific fuel consumption rate (fuel wood) was 7.9 $kg\ kWh^{-1}$ when the operating load was at 7%. Generally, the specific fuel consumption rate at variable load conditions was not available for comparison. Most of the research papers and reports provide the SFC at the maximum load. SFC of 1.3 $kg\ kWh^{-1}$ is reported in Ref. [41]. An average SFC of 1.98 $kg\ kWh^{-1}$ was observed with an average power generation efficiency of the system at 11.15% [44]. The overall efficiency of 18% was obtained with a SFC of 1.38 $kg\ kWh^{-1}$ [24]. The engine operating at full load works with an efficiency in the range of 16–19% [19]. The engine operating at less than the designed capacity will have higher SFC and lower efficiency [19]. This report also highlights that when using fuel wood the SFC is 1.4 $kg\ kWh^{-1}$ [24]. The profile of SFC (fuel wood) at variable load conditions is shown in Fig. 6. The profile of SFC (producer gas) at variable load conditions is shown in Fig. 7. The producer gas consumption rate at variable load condition was estimated based on the fuel wood consumption and specific gas production rate ($Nm^3\ kg^{-1}$).

#### 5.7.1. Performance analysis of diesel engine and producer gas engine

Diesel engines operate at a higher compression ratio and have higher energy density in relation to fuel mixture. Producer gas engines operate at a low CR and at a low energy density of the fuel mixture. The efficiency of the internal combustion engine operating on diesel and producer gas was analyzed at variable load operating conditions. The power generation efficiency of a diesel engine and a producer gas engine at variable load conditions is shown in Fig. 8. The difference in the efficiency of the producer gas engine and diesel engines can be seen from Fig. 8. It may be noted that the power generation efficiency of the diesel engine reaches a maximum of 28% at 80% operating load. However, the power generation efficiency of the producer gas engine reaches to a maximum

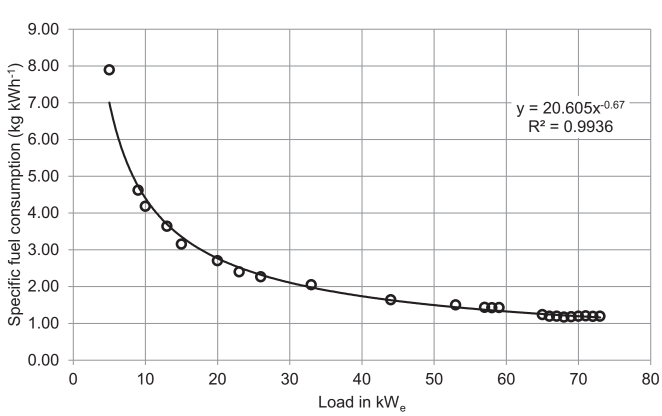


**Fig. 6.** Specific fuel consumption rate (fuel wood) at variable load conditions.

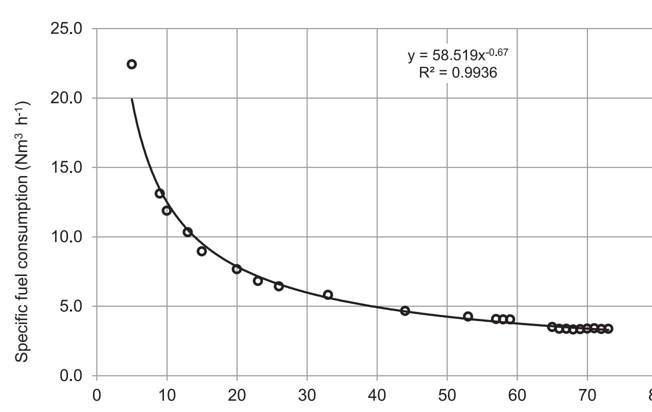


**Fig. 7.** Specific fuel consumption rate (producer gas) at variable load conditions.

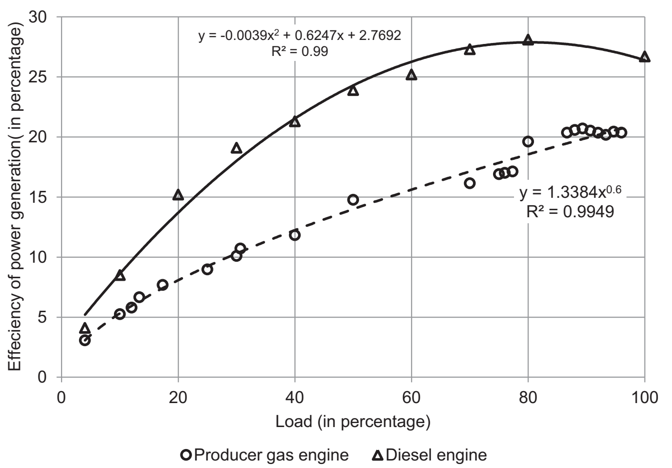


**Fig. 8.** Power generation efficiency of the producer gas engine and diesel engine at variable load conditions.

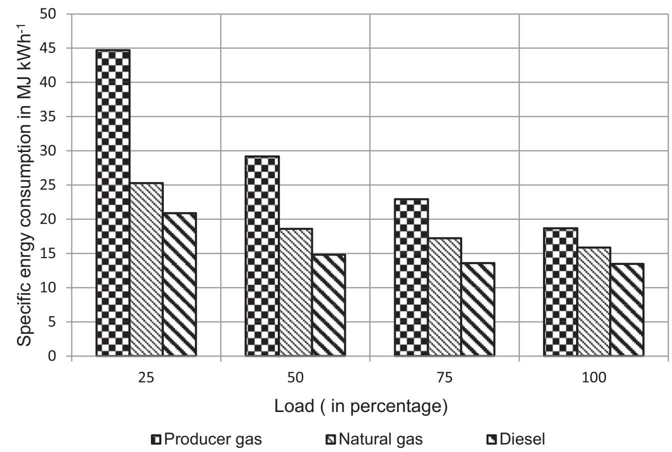


**Fig. 9.** Specific energy consumption rate of the producer gas engine in comparison with diesel engine and natural gas engine at variable load conditions.

of 21% at above 85% operating load. One of the reasons for achieving high efficiency in diesel engine is the high CR. The diesel engine works at a CR of 17:1 while the producer gas engine works at a CR of 12:1. In diesel engine spontaneous and simultaneous ignition of the fuel mixture takes place, whereas in a producer gas engine after the ignition of the fuel mixture at one location the flame propagates in all directions. In addition to the reduction in CR, the flame velocity and the profile of combustion pressure has also resulted in reduction of efficiency of producer gas engines.

*5.7.2. Comparison of the producer gas engine's performance with natural gas engine and diesel engine*

The energy consumption rate of producer gas engines, diesel engines and natural gas engines was analyzed at different operating load conditions. The combustion characteristics of the fuels and its properties relating to energy density influence the performance efficiency of IC engines. A comparison of the fuel properties of producer gas, natural gas and diesel is presented in Table 8. From Table 8, it may be noted that the energy density and the expansion ratio of the fuel mixture formed using producer gas is the least among the fuels compared. Comparison of the energy consumption rate of the producer gas engines, diesel engines and natural gas engines at variable load conditions is presented in Table 8. The specific energy consumption rate (MJ $kWh^{-1}$) of the producer gas engines, diesel engines and natural gas engines was compared at variable load conditions and is shown in Fig. 9. From Fig. 9, it may be noted, that the SEC (specific energy consumption) of the producer gas engine was higher than the natural gas engines and diesel engines. The power generation efficiency of a producer gas engine, diesel engine and natural gas engine at variable load conditions is shown in Fig. 10. From Fig. 10, it may be noted that the maximum power generation efficiency of the diesel engines is 28% and that of the natural gas engines is 24%. Thermal efficiency of a natural gas engine was reported as 28% at 1500 RPM which works out to a power generation efficiency of 24% [70]. A natural gas engine operates at 1600 RPM and works with an efficiency of 33% at 100% load [71]. Most of the engines used for power generation work at 1500 RPM to produce power at 50 Hz.

From Fig. 10, it may be noted that, at 20% load the diesel engine and natural gas engines operate at an efficiency of 17% and 14% respectively. At 20% load the producer gas engine operates at 9% which is about 50% less than the efficiency of diesel engines and 40% less than the efficiency of natural gas engines. At the maximum load, the diesel engine and natural gas engines operate at an efficiency of 28% and 24% respectively. At the maximum load the producer gas engine operates at 21% which is about 20% less than the efficiency of diesel engines and 10% less than the efficiency of natural gas engines.

A 190 $kW_e$ producer gas engine was operated at its maximum load with an efficiency of 26% [72]. An overall efficiency of 25% was obtained at a 20 $kW_e$ power output by a 100% producer gas engine [73]. This system is supported by a two-stage gasifier with steam injection. From the reported values, it may be noted that the maximum power generation efficiency of the producer gas engines varies from 21% to 26%. In the present study, the maximum power generation efficiency achieved by the 75 $kW_e$ producer gas engine was 21%. The variation in the efficiency is due to variation in CR, volumetric filling efficiency, expansion ratio of the fuel mixture, stoichiometric ratio and the quality of the fuel mixture supplied by the intake manifold.

When the producer gas engine and natural gas engines are operated at the same CR of 12:1, the efficiency of the producer gas engine is 12.5% lower than the natural gas engine. This is due to the low energy density of the fuel mixture. The energy density of the fuel mixture is directly proportional to the efficiency of the engine. So it is very important to increase the energy content of producer gas. This could be achieved by reducing the nitrogen content in producer gas.

**Table 8**
Comparison of the fuel properties of producer gas with natural gas and Diesel.

| S. No. | Fuel | Energy content (MJ $kg^{-1}$) | Air–fuel ratio | Energy density (MJ $Nm^{-3}$) | Flame velocity (m $s^{-1}$) | Adiabatic flame temperature (K) | Expansion ratio of the fuel mixture (volume fraction) | Derating (percentage) |
|---|---|---|---|---|---|---|---|---|
| 1 | Producer gas | 5.0 | 1.2 | 2.59 | 50 | 1800 | 6.35 | 21.4 |
| 2 | Natural gas | 45.0 | 18 | 3.00 | 35 | 2210 | 7.34 | 3.5 |
| 3 | Diesel | 42.5 | 18 | 2.83 | – | 2290 | 7.56 | 0.0 |

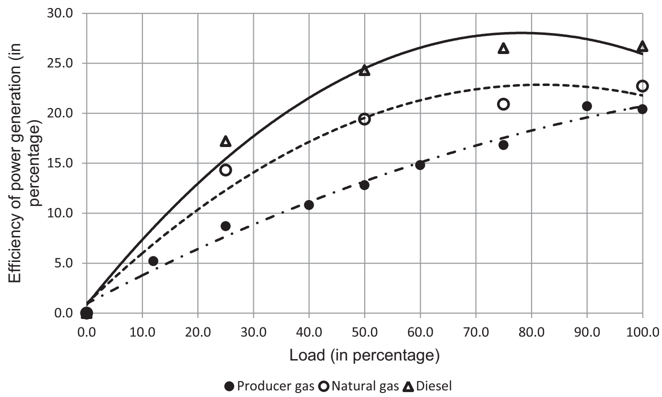


**Fig. 10.** Power generation efficiency of the producer gas engine in comparison with diesel engine and natural gas engine at variable load conditions.

The diesel consumption rate of the engine for power generation at various capacities and at different load conditions is referred from Ref. [74]. The natural gas consumption rate of the engine for power generation at various capacities and at variable load is referred from Ref. [75]. The producer gas consumption rate is obtained from Table 9. The specific fuel consumption rate is influenced by energy content, adiabatic flame temperature and expansion ratio of the fuel mixture.

Comparison of the fuel properties of producer gas with natural gas and diesel.

Comparison of the energy consumption rate of producer gas engines, natural gas engines and diesel engines at variable load conditions.

## 5.8. Energy balance

The energy flow in a producer gas engine was analyzed. The useful energy of the shaft power and the energy flow from flue gas, radiator cooling water and other losses were estimated. The results obtained during the energy balance analysis are presented in Table 10. The energy balance analysis of the producer gas engine was carried out at 97% and 31% load of the design capacity. The energy balance analysis was carried out for understanding the energy flow in detail. The results of the energy balance analysis and its results can be used for further improvement of the efficiency of the producer gas engine.

### 5.8.1. Energy balance analysis of natural gas engines and producer gas engines

Energy balance analysis in a standard natural gas engine was studied at variable load conditions [76]. When the engine was at 20% load the heat loss in the coolant was 49% and in the flue gas exhaust was 24%. When the engine was operated at 50% load the heat loss in the coolant was 36% and in the flue gas exhaust was 23%. When the engine was operated at 100% load the heat loss in the coolant was 27% and in the flue gas exhaust was 29%. From these values, it may be noted that in natural gas engines the heat loss factor in the coolant kept on reducing when there is an increase in load and the efficiency of the engine increases along with an increase in load. The present study shows that the producer gas engine also performs in the same way as the natural gas engine. In partial load conditions the shaft power is much less with a very large heat loss factors through the radiator and exhaust. The heat loss factor in the coolant reduces as the operating load increases. It may be noted from Table 10, when the efficiency of the engine increases along with an increase in operating load, the heat loss factor through the coolant reduces proportionally. This is an indication that the radiator fluid temperature needs to be kept at an optimum temperature to avoid energy loss in the cooling process and increase the efficiency of the engine.

### 5.8.2. Efficiency of the producer gas engine

The shaft power of the engine was estimated based on the power output from the alternator. A power generation efficiency of 85% was considered for estimation of the shaft power. The producer gas engine in the present study was operating with a power generation efficiency of 21%. This works out to be the efficiency of the engine as 24%, referring to the shaft power. At the maximum load, an overall efficiency of power generation of 21% is reported with an efficiency of the engine at 31% [58]. This works out to be a power generation efficiency of 67.7% of the engine's shaft power.

In the present study, the electric power output was measured directly using a digital power meter installed in the control panel of the generator set. Input power was estimated by the gas flow rate and the heating value of the producer gas. The efficiency of the engine was estimated from the energy input, power output and the efficiency of the alternator. The producer gas engine tested during the study was performing at an efficiency of 25%. The balance of the 75% energy goes un-utilized. Co-generation concepts have come up for utilizing the waste heat from the exhaust and radiator. According to the nature of applications, additional equipment is required to be installed to use the waste heat effectively. However, primarily efforts should be made to increase the efficiency of the engine for power generation. In this paper the characteristics of the producer gas and the producer gas engine were studied in detail. All the parameters discussed in Eqs. (1)–(9) need to be optimized for improving the efficiency of the engine.

### 5.8.3. Energy loss through flue gas

A large portion of the energy supplied to the engine was carried away in flue gas. The flow rate of the fuel mixture and the temperature of the flue gas were monitored for estimation of the energy flow (or heat loss) through the flue gas. The flow rate of the flue gas was estimated by monitoring the gas flow rate and estimation of air supplied for combustion of the producer gas. The ratio of air supplied for combustion of flue gas was obtained by measuring the flow rate of gas and air at the inlet of the engine's fuel intake manifold. The rise in temperature with reference to the ambient temperature was monitored to estimate the energy carried

**Table 9**
Comparison of the energy consumption rate of producer gas, natural gas and diesel engines at variable load conditions.

| S. No. | Load (%) | Energy consumption MJ $h^{-1}$ | | | Energy consumption MJ $kWh^{-1}$ | | |
|---|---|---|---|---|---|---|---|
| | | PG | NG | Diesel | PG | NG | Diesel |
| 1 | 25 | 849 | 480 | 397 | 45 | 25 | 21 |
| 2 | 50 | 1108 | 706 | 563 | 29 | 19 | 15 |
| 3 | 75 | 1283 | 964 | 761 | 23 | 17 | 14 |
| 4 | 100 | 1400 | 1188 | 1010 | 19 | 16 | 13 |

**Table 10**
Results of the energy balance analysis at maximum load and part load conditions.

| S. No. | Component | Unit | At 97% load | | At 31% load | |
|---|---|---|---|---|---|---|
| | | | Share | % | Share | % |
| 1 | Energy input | kW | 351 | 100 | 272 | 100 |
| 2 | Flue gas | kW | 98 | 29 | 114 | 42 |
| 3 | Radiator | kW | 115 | 34 | 104 | 38 |
| 4 | Shaft power | kW | 86 | 26 | 27 | 10 |
| 5 | Unaccounted | kW | 52 | 11 | 28 | 10 |

away by the flue gas. When operating the producer gas engine at 97% load the energy loss through the flue gas was 29%. When operating the producer gas engine at 31% load the energy loss through the flue gas was 42%. It may be noted the energy loss in the flue gas increases when there is a reduction in the operating load. This is the reason for the reduction in efficiency when operating an engine at lower rating of the design capacity.

### 5.8.4. Energy loss through radiator cooling water

The heat energy flow through the radiator cooling water was estimated by monitoring the water flow rate and temperature loss at the cooling water loop. Water flow was measured using a non-contact water flow meter and the water temperature was measured at the inlet and outlet of the radiator using k-type thermocouple connected to a digital display unit. During the study it was estimated that 34% of energy is lost through the radiator at 97% load. The heat loss through the radiator was 38% when operating the engine at 31% load. The heat loss increases simultaneously through the radiator and exhaust when the engine operates at part load conditions. It may be noted that, when reducing the operating load of the engine, the heat loss through the flue gas increases at a higher rate, than the heat loss through the radiator. Preventing of excessive cooling of the engine was controlled by a magnetic driven coolant pump [77]. By avoiding the excessive cooling of the engine the heat loss through the coolant was reduced and the fuel consumption rate was improved by 1.7–4.0%. This indicates that by optimizing the cooling fluid temperature, the heat loss of the engine can be minimized. The efficiency of the IC engine can be increased by minimizing the heat loss through the radiator and exhaust.

### 5.8.5. Energy losses in the engine itself

The balance of the energy loss which is not accounted through the flue gas and radiator is lost directly from the surface of the engine body and through lube oil circuit. The energy lost directly from the engine and its component accounts for 11% of the energy input when the engine was operating at 97% load. The energy lost from the engine and its component accounts for 10% at 31% load.

### 5.9. Efficiency optimization

It may be noted that only 24% of the energy supplied to the engine was converted into useful energy. This indicates there is scope for further improvement. The efficiency of the engine can be improved by increasing the energy content of the fuel mixture and increasing the expansion ratio of the fuel mixture. For achieving a better efficiency the expansion ratio ($E_r$) of the fuel mixture should be greater than or equal to the compression ratio ($C_r$). For achieving a better performance of the engine the relation between $E_r$ and $C_r$ can be written as shown in Eq. (10).

$$\left( E_r / C_r \right) \geq 1 \tag{10}$$

An engine having a compression ratio of 10:1 can achieve an efficiency of 34% and 43% when the expansion ratio is 10 and 30 respectively [58]. This indicates that corresponding to an $E_r/C_r$ of 1 and 3 the efficiency of the engine works out to be 34% and 43% respectively. In case of producer gas engine $C_r/E_r$ was closer to 0.5 when the engine operated above 90% of the design load. This indicates, $C_r/E_r$ is at its maximum value when the engine is operated at maximum load. At part load conditions the ratio of $E_r$ and $C_r$ comes much lower due to low filling efficiency. At part load conditions, the volumetric filling efficiency is controlled by increasing the resistance to the fuel flow by the partial closing of the throttle valve. At an $E_r/C_r$ of 0.5 the efficiency of power generation works out to be 21%. A profile of the power generation efficiency based on $E_r/C_r$ is presented in Fig. 11. From Fig. 11, it may be noted that the power generation efficiency is maximum when the value of ($E_r/C_r$) > 1.5. In Fig. 11, the trend line indicates that IC engines can be operated with a power generation efficiency in the range of 35%–43%, when 1 < ($E_r/C_r$) < 2.

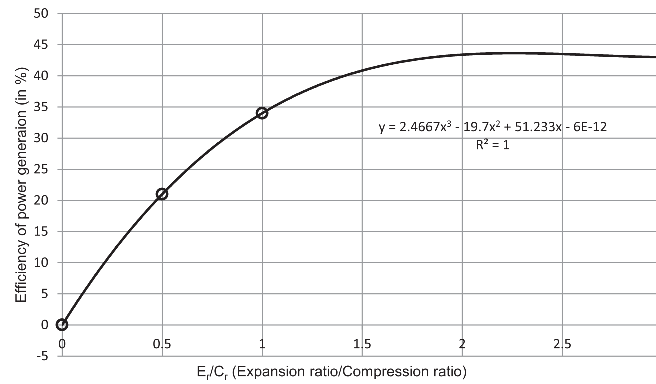


**Fig. 11.** Power generation efficiency of the IC engine with variation in $E_r/C_r$.

In order to improve the efficiency of a diesel engine, the pressure wave supercharger technology was investigated by Lei et al. [78]. This process basically raises the intake pressure by supercharging which leads to increase the expansion ratio. Increase in expansion ratio provides improvement in performance efficiency of the engine. The expansion ratio can be maximized by increasing the energy density of the fuel mixture. Nitrogen content is one of the influencing factors of the energy content of the producer gas. It is important to reduce the nitrogen content in the producer gas to increase the efficiency of the engine. Higher the nitrogen content will lower the energy content and hence, lowers the adiabatic flame temperature. Reduction in adiabatic flame temperature results in a reduction in the expansion ratio. Irrespective of the CR reduction in expansion ratio will result in poor efficiency.

## 6. Conclusions

Performance of a producer gas engine used for power generation was studied in detail. The components of producer gas were analyzed. The heating value of the producer gas was 5.6 MJ $Nm^{-3}$ based on the combustible gas components in the producer gas. The increase in heating value was obtained on supplying hot air for gasification. The cold gas efficiency of the system was 88% when the ash and charcoal return rate in the ash-pit was less than one percent. The tar content in the producer gas before and after cleaning was estimated at 300 mg $Nm^{-3}$ and 45 mg $Nm^{-3}$ respectively. The dust content in the producer gas before and after cleaning was estimated at 600 mg $Nm^{-3}$ and 25 mg $Nm^{-3}$ respectively. The performance of the producer gas engine was studied in detail at variable operating load conditions. The performance of the producer gas engine was compared with the performance of diesel engines and natural gas engines. The power generation efficiency using a 100% producer gas engine was 21% at a maximum load of 91%. With conversion efficiency of shaft power to electric power at 85%, the maximum efficiency of the producer gas engine works out to be 24%. At the maximum load, the specific fuel consumption (fuel wood) for power generation using the producer gas engine works out to be 1.2 kg $kWh^{-1}$ and the specific fuel consumption (producer gas) for power generation using the producer gas engine works out to be 3.4 $Nm^3$ $kWh^{-1}$. The specific fuel consumption rate was estimated when operating the producer gas engine at variable load conditions. The efficiency of the engine reduced to 3% when the operating load was at 7%. This is four times less than the overall efficiency of power generation at maximum load. The producer gas engine was working at a maximum efficiency, when the engine was operated at above 85% load. The specific fuel consumption rate increases up to six times, when operating the engine at 7% load. The importance of compression ratio, combustion pressure and expansion ratio was discussed in detail. A method for estimation of the engine efficiency was worked out by using the key parameters like volumetric efficiency, expansion ratio and compression ratio. The efficiency estimated using volumetric efficiency, expansion ratio and compression ratio was compared with the efficiency estimated using the common method of energy input and energy output. The relationship between the engine efficiency and parameters like volumetric efficiency, expansion ratio and compression ratio was established. The internal combustion engines can be operated with a power generation efficiency in the range of 35–43%, when $1 < (E_r/C_r) < 2$. This relationship between the efficiency and the expansion ratio of the fuel mixture can be used for optimizing an engine's design parameters and hence, maximize the power generation efficiency of the system.

## Acknowledgment

We are grateful to Dr. R K Pachauri, Director General, TERI for his continuous encouragement and support. We would also like to thank Mr. Amit Kumar, Director, Energy Environment Technology Development Division of TERI for providing valuable support to conduct this study.

## Nomenclature

| | |
|---|---|
| $E_d$ | energy density of the fuel mixture MJ Nm$^{-3}$ |
| $C_{vf}$ | energy content of the fuel mixture in MJ |
| $V_f$ | volume of the fuel in Nm$^3$ |
| $V_a$ | volume of air used for combustion of the fuel, in Nm$^3$ |
| $C_r$ | compression ratio of the fuel mixture |
| $S_v$ | swept volume of the piston between the TDC (top dead center) and the BDC (bottom dead center) in Nm$^3$ |
| $D_s$ | volume of the dead space above the piston at TDC, in Nm$^3$ |
| $V_e$ | volumetric filling efficiency, in fraction |
| $V_s$ | swept volume in m$^3$ |
| $E_r$ | expansion ratio of the fuel mixture |
| $T_1$ | initial temperature of the fuel mixture, in K |
| $A_t$ | adiabatic flame temperature of the fuel mixture, in K |
| $\eta_{\underline{e}}$ | efficiency by energy input and output, in percentage |
| $E_i$ | energy input in MJ |
| $E_o$ | energy output in MJ |
| $\eta_{\underline{c}}$ | design efficiency of the engine with respect to the compression ratio, volumetric efficiency and friction loss |
| $\varepsilon$ | compression ratio |
| $k$ | a constant equal to 1.3 in case of producer gas [39] |
| $V_e$ | volumetric filling efficiency in fraction |
| $P_\theta$ | pressure in the cylinder, when the crank angle is at $\theta$ |
| $\theta$ | crank angle, ($\theta = 0$ when the piston is at TDC during the power stroke) |
| $\eta_{E_r}$ | actual working efficiency of the engine based on the expansion ratio, volumetric efficiency and compression ratio. |
| $S_{v-\theta}$ | swept volume of the piston when the crank angle is at $\theta$ and the volume of the dead space in m$^3$ |
| $W_n$ | Wobbe number |
| $C_v$ | calorific value of the gas |
| $\rho_g$ | specific gravity of the fuel |

*Abbreviations*

| | |
|---|---|
| DPS | distributed power system |
| DDG | decentralized and distributed generation |
| VESP | village energy security program |
| ICE | internal combustion engines |
| ICT | induction coil transformers |
| SGR | specific gasification rate |
| TDC | top dead center |
| RPM | revolutions per minute |
| CV | calorific value |
| ER | equivalence ratio |
| CNG | compressed natural gas |
| SI | spark ignition |
| SFC | specific fuel consumption |

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