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Effect of nozzle and combustion chamber geometry on the performance of diesel engine operated on dual fuel mode using renewable

V. S. Yaliwal[a], R. Banapurmath[b], N. M. Gireesh[b] R. S .Hosmath[b], T. Donateo[c] , P. G. Tewari[b]

a) Department of Mechanical Engineering, S.D.M. College of Engineering and Technology, Dharwad, 580 002, Karnataka, India

b) B.V.B. College of Engineering and Technology, Hubli, 580 031, Karnataka, India

c) Department of Engineering for Innovation, Via per Arnesano, 73100, Lecce, University of Salento, Italy

Abstract

Renewable and alternative fuels have numerous advantages compared to fossil fuels as they are biodegradable, providing energy security and foreign exchange saving and addressing environmental concerns, and socio-economic issues as well. Therefore renewable fuels can be predominantly used asmfuel for transportation and power generation applications. In view of this background, effect of nozzle and combustion chamber geometry on the performance, combustion and emission characteristics have been investigated in a single cylinder, four stroke water cooled direct injection (DI) compression ignition (CI) engine operated on dual fuel mode using Honge methyl ester (HOME) and producer gas induction. In the present experimental investigation, an effort has been made to enhance the performance of a dual fuel engine utilizing different nozzle orifice and combustion chamber configurations. In the first phase of the work, injector nozzle (3, 4 and 5 hole injector nozzle, each having 0.2, 0.25 and 0.3 mm hole diameter and injection pressure (varied from 210 to 240 bar in steps of 10 bar) was optimized. Subsequently in the next phase of the work, combustion chamber for optimum performance was investigated. In order to match proper combustion chamber for optimum nozzle geometry, two types of combustion chambers such as hemispherical and re-entrant configurations were used. Re-entrant type combustion chamber and 230 bar injection pressure, 4 hole and 0.25 mm nozzle orifice have shown maximum performance. Results of investigation on HOME-producer gas operation showed 4e5% increased brake thermal efficiency with reduced emission levels. However, more research and development of technology should be devoted to this field to further enhance the performance and feasibility of these fuels for dual fuel operation and future exploitations.

Keywords: Producer gas, Dual- fuel engine, Gasifier- engine system, combustion chamber, performance, combustion and emissions.

1. Introduction

Power production from diesel engines are getting more popular because of their higher brake thermal efficiency, power output, reliability, less fuel consumption, lower emissions and durability as well. Hence diesel engine technology plays a vital role in transportation, agricultural and power generation applications. In the present energy scenario, life of conventional fossil fuels has become limited, while the demand for energy is growing at a faster rate. Due to rapid depletion of conventional fuels, increasing prices of crude petroleum and stringent environmental legislations, use of environment friendly fuels (biofuels) in partial or complete replacement for diesel engine applications is the need of the hour. Major emissions from diesel engines include nitric oxide (NOx) and smoke. These pollutants can be overcome by dual fuel concept. However, a diesel engine using biodiesel-producer gas combination operating on dual fuel mode results in a higher hydrocarbon (HC) and carbon monoxide (CO) emissions [1], [2]. Energy conservation with high efficiency and low emissions are important topics for research in engine design and development. For enhancing thermal efficiency of producer gas fueled dual fuel engine and controlling emissions, various biomass feed stock, increased compression ratio, addition of hydrogen, blends with ethanol and intake air pressure boosting are being applied [1], [3], [4], [5]. Biofuels such as biodiesel and producer gas derived from biomass are being considered as better alternative fuel in order to ensure both food and energy security in the prevailing situation of scarcity of fossil fuels. However, in both transport and power generation applications, newer emission legislations have been started to enforce limits on the emission levels [6]. For achieving better thermal efficiency with lower emission levels, many investigators have focused their interest on the domain of fuel related and engine modification techniques. Therefore an effort has been made to curtail negative effects and it is now important to investigate the effects of combustion chamber designs and injection nozzle geometry on the performance and emission characteristics of diesel engine.

Utilization of gaseous fuel along with injected fuel in a dual fuel engine leads to combustion with more complexity because it involves two fuels with different properties and is burnt simultaneously inside the engine cylinder. Therefore, heat release rate of dual fuel combustion is the result of three combustion stages [7], [8], [9]. In case of dual fuel engine, injection of liquid fuel is performed with an in-cylinder injection system. These engines can operate un-throttling, with load regulated by admitting gaseous fuel along with air through induction manifold. Substantial research on producer gas fueled dual fuel engine and its effect on performance and emission levels has been reported in the literature [4], [10], [11], [12], [13], [14], [15], [16], [17]. Dual fuel engines are known for good liquid fuel saving with decreased smoke and nitric oxide (NOx) emissions. Some of the investigators have reported lower performance, increased HC/CO and lower NOx/smoke levels under producer gas dual-fueling [2], [11], [12], [13], [14], [15], [16], [17], [18]. Some researchers have reported decreased power output of engine, whereas others have not mentioned. Loss of thermal efficiency compared to diesel operation has been reported in the literature. In view of this, several investigators have made an effort for achieving comparable efficiencies [8], [11], [13]. Biomass energy conversion technologies to achieve prominence for developing energy for rural as well as industrial sectors are required to improve the quality of life [4], [12], [13], [15], [18]. Different types of gasifier and their advantages and disadvantages, applications, current status, challenges, potential scope and economic analysis of the gasifier-engine system have been investigated [1], [2], [19], [20]. Performance of an engine depends on the two factors such as, properties of fuel used and basic engine design. Majority of the research work is focused on the utilization of compressed natural gas (CNG) and liquefied petroleum gas (LPG) in engines operated on both single and dual fuel mode and developed many new technologies. Many published literatures have reported with reference to fuel properties and their influence on dual fuel operation. However, very few literatures have been reported on basic engine modifications. Therefore, use of producer gas in engines still needs more detailed studies, as this area is less investigated. Improvement in performance with decreased exhaust emissions of producer gas dual-fuel engine needs more detailed investigation with respect to both fuel properties and basic engine design modification.

The performance of a diesel engine is significantly affected by injector type. The performance of a diesel engine is largely influenced by fuel spray characteristics that are produced using relatively high injection pressures. Highest injection pressure that can be used in the diesel engine is 205 bar, whereas, modern CI engines are equipped with common rail direct injection (CRDI) technology. This will employ very high injection pressures (2000 bar). In this injection system, fuel injection pressure can be regulated by controlling the fuel rail pressure [21], [22]. Fuel injection is an important operating parameter which affects the fuel vaporization; distribution and mixing of fuel within the combustion chamber which is in turn responsible for the overall performance of a diesel engine. Appropriate droplet size, fuel distribution, and penetration lead to more efficient combustion and thus lower emissions [24]. Some of modern diesel engines use micro-orifices having various orifice designs and affect engine performance to a great extent. Several investigators have investigated the effect of dynamic factors on injector flow spray, combustion and emission levels from a diesel engine [23], [25]. Experimental studies involving the effects of nozzle orifice geometry on global injection and spray behavior has been reported [23], [26], [27], [28]. Smaller injector nozzle hole diameter produces smaller droplet size and results into reduced spray tip penetration due to the low spray momentum [29]. Air and fuel mixing depends on the number of nozzle holes and diameter. Adverse effect on combustion and emissions has been reported when number of holes exceeds a certain threshold value. This could be due to lack of the air entrainment required for the achievement of a stoichiometric mixture [30].

Effective air and fuel mixing is significantly affected by mainly spray characteristics and air flow inside the engine cylinder. Modification of combustion chamber by suitable piston bowl can significantly affect the different phases of heat release i.e., it affects the shape and magnitude of the heat release rate profile, by affecting bulk airflow and turbulence, thus affecting air-fuel mixing rates. Effective control and manipulation of the heat release rate is important to limit peak cylinder pressure, combustion noise and emission levels. A good combustion chamber provides better squish, forcing the air to the centre of the combustion chamber [8]. This causes turbulence even when the fuel is injected into the cylinder. Present diesel engines use hemi-spherical combustion chamber and results into better performance for the diesel fuel. However, it may not be good for alternative liquid and gaseous fuels; hence it is necessary to design different combustion chambers for alternative fuels [31], [32], [33], [34]. The effect of combustion chamber configuration on the engine performance is very complex to analyze due to its influence on the flow field and air-spray interaction [35], [36]. The shape of piston bowl controls the movement of air and fuel as piston moves up during compression stroke. Suitable changes in the in-cylinder flow field or swirl results into vortex inside the piston bowl before combustion takes place, creating a better mixture formation. Swirl is used to promote rapid mixing of inducted air and fuel at the end of the compression stroke. However, fuel-air mixing is predominantly governed by fuel injection characteristics and air-swirl. Rapid mixing is essential, because small DI diesel engines operating at high speed have a very short time-window over which combustion must occur. This is necessary in order to limit formation of soot in the expansion phase, and minimize specific fuel consumption [31]. This fact results into better and more efficient combustion, leading to enhanced power output. Therefore the behavior of fuel injected in the combustion chamber and its interaction with air is important as far as combustion and emissions are concerned. It is well known that nozzle geometry and cavitations strongly affect evaporation and atomization processes of fuel. Hence, the combustion chamber of an engine plays a major role during combustion of wide variety of fuels. At fixed compression ratio and newly developed piston, researchers have observed increased swirl at TDC, less smoke, comparable HC and NOx levels. In this context, many researchers have performed both experimental and numerical studies on the use of various combustion chambers and analyzed its effects on the engine performance [8], [9], [35], [37], [38], [39], [40], [41], [42], [43]. Improvement in air entrainment with increasing swirl and injection pressure has been reported in the literature [44], [45]. Optimum combustion chamber geometry of engine must be considered to have a better engine operation, performance and emission levels. Suitable combustion chamber geometry helps to increase squish area and proper mixing of gaseous fuel with air [35], [46]. Designing the combustion chamber with narrow, deep, shallow reentrance and low protuberance on the cylinder axis while the spray should be oriented towards the bowl entrance reduces the NOx emission levels to the maximum extent [12]. Several investigators have studied that, the effect of piston bowl, number of nozzle holes and swirl ratio on the performance of CI engine operating on single and dual fuel mode (diesel/biodiesel-CNG combination). Soot emission decreases with increasing piston center depth and when larger diameter piston bowl is used. This allows more space and spray traveling throughout the combustion chamber leading to better utilization of oxygen and reducing the wall impingement [32], [33].

For a fixed compression ratio, the swirl levels at TDC increases if the bowl diameter is reduced, leading to less smoke, higher NOx levels and HC emissions [47]. The squish–swirl interaction, are changed when offset of the bowl with respect to the cylinder axis is changed. The ratio of the throat diameter to the cylinder bore size is a percentage squish area. A small throat diameter causes a larger fraction of the piston face to approach the cylinder head closely, and therefore produces a higher squish velocity [36]. The total volume of piston bowl, and compression ratio, is largely controlled by the maximum bowl diameter. This is one of the parameter to be set when designing a new piston bowl shape. High velocity airflow into the bowl, and combustion gas out of the bowl, creates large temperature gradients and high heat transfer rates to the piston bowl's internal surface [31]. Reducing throat diameter can enhance the squish flow and improve the fuel air mixing, therefore reducing the particulates and specific fuel consumption (SFC) but increasing the NOx emissions. Bowl with reduced throat and retarded injection could reduce not only particulates but also NOx [31]. Increasing bowl diameter will reduce the swirl velocity in the bowl. This reduces the rate of air/fuel mixing, but can reduce heat transfer and other energy losses. The maximum bowl diameter is important because it affects the bulk swirl velocity at the end of compression [48]. Piston pip design is constrained to prevent injector tip-to-piston contact when the engine is running, and to prevent wall wetting of the piston pip by closely approaching fuel sprays. Pip shape strongly affects the engine performance and emissions. Deep piston bowl depth tends to increase wall-wetting of the lower bowl surfaces, by preventing re-entrainment of fuel into the air-stream [33]. The inclined pip bowl (less steep gradient to the top of the pip) has better air fuel mixing and combustion than the vertical pip [32]. Large torroidal radius bowl produces better combustion than the small torroidal radius bowl. The soot emissions and indicated specific fuel consumption (ISFC) are reduced with increasing piston bowl torroidal radius.

2. Present work

Effects of nozzle and combustion chamber geometry on the performance of single fuel operation have been reported in the literature. The review of literature suggests that the effect of nozzle and combustion chamber geometry on the performance of producer gas-diesel/biodiesel dual fuel engine has been less investigated. Hence, this area still needs detailed investigation. In this context, major attention and interest is given to enhance the thermal efficiency of producer gas operated dual fuel engine with decreased emission levels. In this present work, experiments have been conducted on a water-cooled single-cylinder direct injection (DI) CI engine operated on dual fuel mode using HOME and producer gas induction. In the initial stage of the present work, the nozzle geometry was optimized in terms of injection pressure (varied from 210 to 240 bar in steps of 10 bar), number of nozzle holes and nozzle size (three different nozzles were selected having 3, 4 and 5-holes and each one is having an orifice size of 0.2, 0.25 and 0.3 mm in diameter). Further, effect of combustion chamber geometry was optimized (two combustion chamber configurations such as hemispherical and re-entrant type combustion chamber were selected). The optimum parameter in terms of injection timing has been reported in earlier studies by same authors [2], [43]. Finally, the results were compared with baseline data and analyzed.

3. Characterization of fuels used

Honge oil methyl ester (HOME) was derived from Honge seeds and producer gas by partial combustion of woody (Babul) biomass. The properties of the fuels were determined in the laboratory. Table 1 shows the properties of liquid fuels and proximate and ultimate analysis of biomass feed stock used in the present study. The composition of producer gas derived from babul wood is shown in Table 2.

Table 1 Properties of liquid fuels and Proximate and ultimate analysis of biomass feed stocks

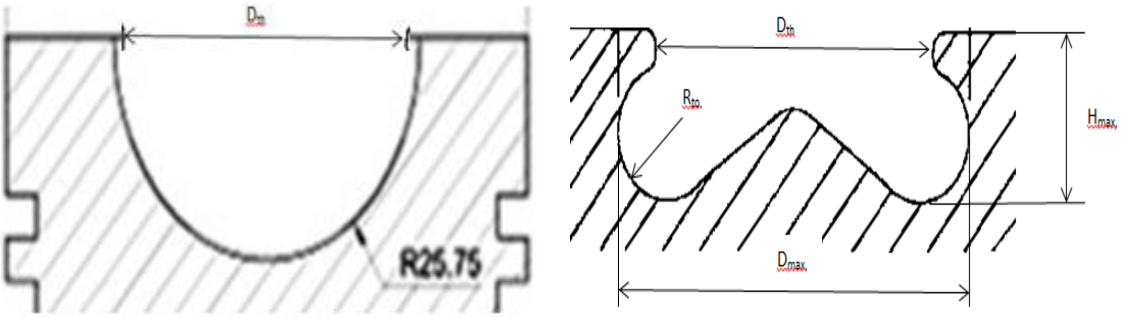
Sl.No	Properties	Diesel	HOME	Description	Babul wood
1	Viscosity @ 40 0 C (cst)	4.59 (Low)	5.6	Moisture Content, % wlw	10.3
2	Flash point 0 C	56	163	Ash Content, % wlw	0.79
3	Calorific Value in kJ / kg	45000	36,010	Volatile Matter, % wlw	85.8
4	Specific gravity	0.830	0.870	Fixed Carbon % wlw	13.4
5	Density Kg / m3	830	890	Sulphur, % wlw	0.05
6	Type of oil	Fossil	Non edible	Nitrogen, as N % wlw	0.30
7	------	----	------	Gross Calorific value, Cal/g	5631.O
	------	-----	------	Gross Calorific value, kJ/ kg	23575.8
8	------	----	------	Density, kg/ m3	380
9	------	-----	------	Phosphorus % w/w	---

Table 2. Composition of producer gas

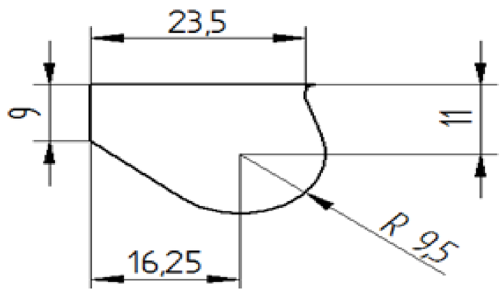
Type of wood	CO %	H2 %	Methane %	HC %	N2 %	Water Vapour %	CO2 %	Calorific value MJ/Nm3	Density kg/m3
Babul wood	18-22%	15-19%	1-5 %	0.2-0.4%	4.5-5.5%	4	8 - 10%	5.6	360

4. Development of re-entrant combustion chamber for dual fuel operation

Combustion chamber has been modified without altering compression ratio. In the present work, Re-entrant type combustion chamber (RCC) has been developed from the baseline of hemi-spherical combustion chamber (HCC).



(a) Line diagram of HCC and RCC



(b) Dimensions of RCC

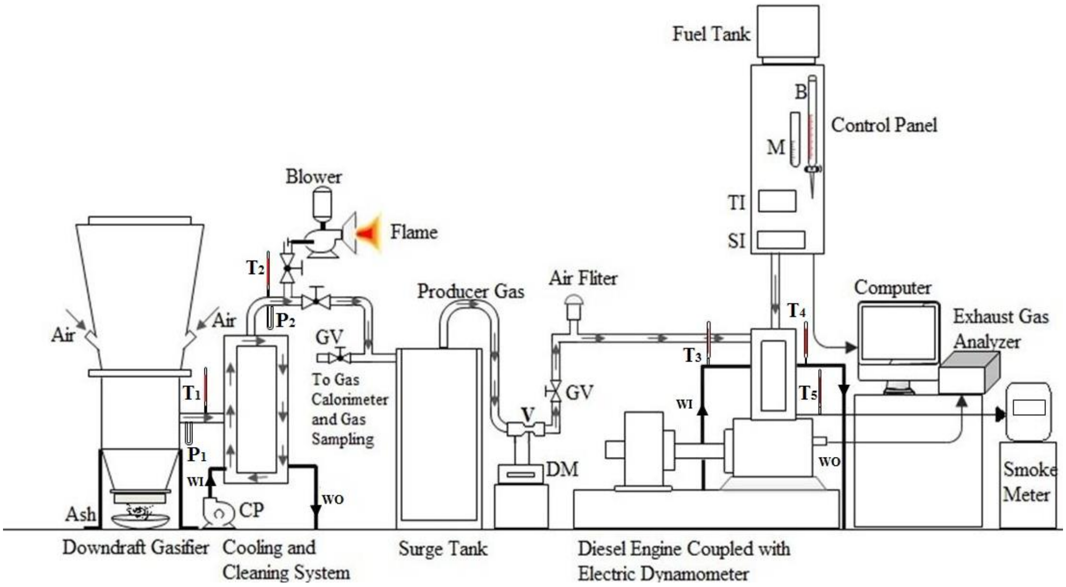
(a) c) Photographic view of HCC and RCC

Fig. 1. Combustion chamber shapes

Based on the literature available, piston depth to diameter has been varied proportionately to obtain RCC. Fig. 1(a),(b) and (c) provide sketches of hemispherical (HCC) and re-entrant combustion chamber (RCC) geometries. The dimensions have been found for RCC is shown in Fig. 1(b). During the development of re-entrant combustion chamber, the bowl volume has been kept same as that of hemispherical combustion chamber (35759.4 mm3). The other dimensions of re-entrant combustion chamber are 47.0, 20.5, 9.2 and 51.5 mm for throat diameter (Dth), Maximum bowl depth (Hmax), Torroidal radius (Rth) and maximum bowl diameter (Dmax) respectively. Re-entrant type combustion chamber uses bowl to redirect fuel down into the bowl and then towards center of the bowl. This feature helps to utilize the air in the bowl very well. Spray targeting is such that small portion of fuel is directed into squish region utilizing the air in the region properly [8], [35], [46].

5. Experimental setup

Experiments were conducted on a Kirloskar TV1 type, four stroke, single cylinder, water-cooled diesel engine test rig. Fig. 2(a) and (b) shows the schematic experimental set up. Eddy current dynamometer was used for loading the engine.



(a) Overall view of Experimental Setup

(b) Photographic view of experimental set up

Fig. 2. Experimental set up

The engine was operated at the rate of constant speed of 1500 rev/min. The down draft gasifier was suitably connected to the engine with filter and cooling and cleaning system. Producer gas was generated using a downdraft gasifier and is taken inside the combustion chamber by the suction of engine. The gas flow was measured using a calibrated venturimeter provided with digital gas flow meter. Cooling of the engine was accomplished by circulating water through the jackets of the engine block and cylinder head. The cylinder pressure was measured using Piezo electric transducer fitted in the cylinder head as shown in Fig. 3. Fig. 4 shows the parallel flow gas entry carburetor for producer gas induction fitted to the inlet manifold of the engine. In the present work, the amount/quantity of both injected fuels of Diesel and HOME have been measured on volumetric basis. At fixed brake power, more amount of HOME is injected as its calorific value is comparatively lower and also its kinematic viscosity is higher (nearly twice diesel). This is done by adjusting the governor speed so that constant speed is maintained in both the versions of the injected fuels. At 80% load the specific fuel consumption for diesel operation is 260 g/kWh while for HOME it is 320 g/kWh.

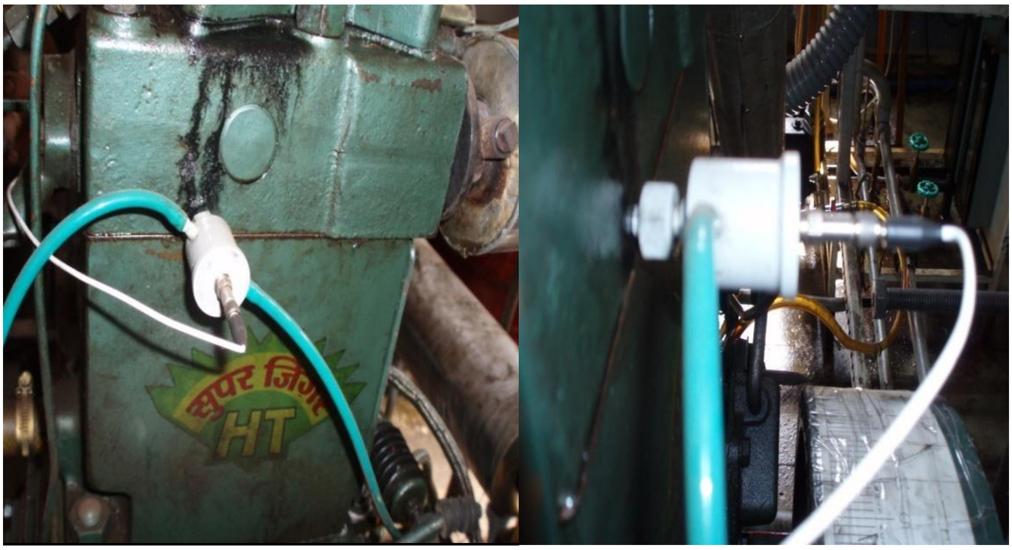


Fig. 3. Views of Pressure Sensor fitted to engine cylinder

Fig. 4 . Parallel gas entry carburetor for producer gas induction fitted to the engine

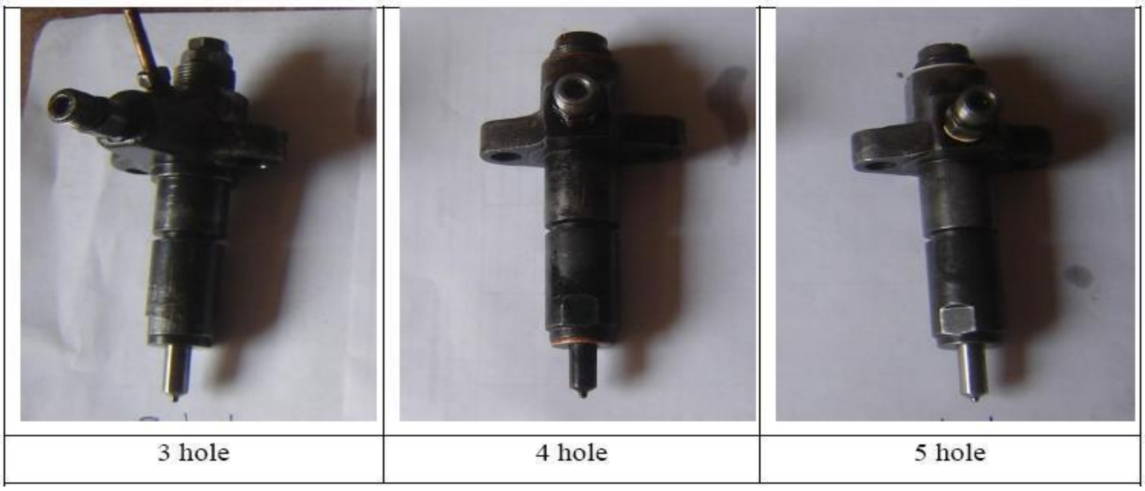


Fig. 5. Different types of injectors.

The emission characteristics were measured by using HARTRIDGE smoke meter and five gas analyzer during the steady state operation. The smoke meter works on the principle of comparative basis. A DELTA 1600 S Exhaust Gas Analyzer was used to measure the regulated emission levels such as HC, CO and NOx. It uses a non-dispersive infrared technology for measuring various emissions levels. It has different sensors for measuring an individual gas and pyroelectric detectors. These detectors collect the light transmitted and produces a corresponding voltage proportional to light intensity. Then, the detector output is sent to an analogue or digital converter (ADC) and microprocessor samples the ADC and thereby provides the actual data. The exhaust gas analyzer and smoke meter were switched on and allowed to stabilize before the measurements, and these instruments were periodically calibrated. The temperature of cooling water at exit was maintained at 70 °C.

Table 3 .Specification of downdraft gasifier.

Type	Down draft gasifier
Rated capacity	15000kcal/hr
Rated gas flow	$15Nm^3/hr$
Average gas calorific value	$1000kcal/m^3$
Rated woody biomass consumption	5-6kg/hr
Hopper storage capacity	40kg
Biomass size	10mm (Minimum) 50mm (Maximum)
Moisture content (DB)	5 to 20%
Typical conversion efficiency	70-75%

Table 4. Specification of experimental test rig

Sl No	Parameters	Specification
1	Machine Supplier	Apex Innovations Pvt Ltd, Sangli. Maharastra State.
2	Engine Type	Single cylinder four stroke water cooled direct injection TV1 CI engine with a displacement volume of 662 cc, compression ratio of 17:1, developing 5.2 kW at 1500 rev/min TV1 (Kirolsker make)
3	Software used	Engine Soft
4	Nozzle opening pressure	200 – 225 bar
5	Governor type	Mechanical centrifugal type
6	Cylinder diameter (Bore)	0.0875 m
7	Stroke length	0.11 m
8	Piston bowl dimension	52 mm diameter
9	Clearance/length	40.1 cc at CR 17.5.
10	Connecting rod length	234 mm
11	Combustion chamber	Open chamber (Direct Injection) with hemispherical cavity
12	Eddy current dynamometer:	Model:AG – 10, 7.5 kW at 1500 to 3000 RPM and water flows through dynamometer during the use

In this present work, mechanical fuel injection system was used. During the complete experimentation, the gas flow rate and engine speed were maintained constant. For the present work, the injection timing was kept constant at 27° bTDC and compression ratio at 17.5. The injection pressure for diesel/HOME – producer gas operation was varied in the range of 210–240 bar in steps of 10 bar. However, initial tests were carried out at an injection pressure of 205 bar (manufacturer setting). Further, experiments were conducted by using HOME-producer gas combination using two different combustion chamber shapes (Hemispherical (HCC) and re-entrant combustion chamber (RCC) shapes). The various dimensions of RCC are shown in Fig. 1(a) and (b). Different types of injector nozzles used in the present work are shown in Fig. 5. Finally the results obtained with HOME – producer gas operation was compared with Diesel–producer gas operation. The specification of the down draft gasifier and compression ignition (CI) engine is given in Table 3, Table 4. For each load, five readings were generated to ensure the accuracy of the data recorded and averaged out data was considered for analysis. During the study careful experimental arrangements were made to obtain consistent and repeatable measurements.

5.1 Fuel supply system

Ensuring uniform supply of quality gas to the dual fueled engine is quite difficult as this would depend on the flow conditions occurring through the gasifier system, pressure drop across the gasifier system and gas temperature at the gasifier outlet, which varies with both engine design and operating conditions. Suitable carburetor was used for mixing air and gaseous fuel. It was developed in such a way that it must have an ability to maintain the required air-fuel ratio (1.2–1.5:1) with varying load and pulsating gas flow conditions, besides providing smooth operation with minimal pressure losses and on-line provision is provided for air/fuel tuning [16].

The downstream of the venturimeter was connected to the carburetor and gas accelerates with the air flow as it is introduced into the carburetor. During the engine operation, appropriate mixture of both producer gas and air were supplied to the engine. The supply of producer gas was adjusted manually to obtain maximum substitution of producer gas or percentage of biodiesel displacement. In this present work, for each load, the pilot quantity of diesel/biodiesel were controlled in such a way that minimum quantity of their injection is ensured for initiating the combustion of fuel combinations and further ensuring maximum gas induction in to the intake manifold. During experiment, the producer gas valve was fully opened in order to substitute more gas and the quantity of liquid fuel was varied to maintain constant speed.

6. Results and discussions

This section presents the effect of injector nozzle and combustion chamber geometry on the performance of dual fuel engine. Comparative assessment of dual fuel engine by using suitable nozzle and combustion chamber design is a special feature of this study. In this present work, effect of injection pressure, number of nozzle holes and diameter and combustion chamber configuration on the performance, combustion and exhaust emissions of diesel engine operating on dual fuel mode using diesel/HOME-producer gas combination at variable engine load conditions were investigated. However, the effect of nozzle geometry was carried out only for 80% load. In this context, a brief overview of preliminary engine testing is summarized in the following sections.

6.1 Optimization of injector opening pressure

6.1.1 Performance characteristics

It is stated that for each engine test examined, the brake thermal efficiency (BTE) is estimated from the calculated power output, the calculated mass flow rate of liquid fuel and producer gas. Fig. 6 provide BTE for various injectors opening pressure (IOP). It is observed that, IOP has significant influence on the spray pattern, droplet size and penetration. For the same IOP, diesel-producer gas operation resulted in higher BTE compared to HOME-producer gas operation. This could be attributed to lower mixing rate caused by higher viscosity and lower volatility character of HOME. Also, lower calorific value of both HOME and producer gas, lower flame speed and auto ignition temperature of producer gas further adds to this trend. Results showed that, HOME-producer gas operation resulted in higher thermal efficiency at an IOP of 230 bar compared to the operation with 205, 210,220 and 240 bar. From the results, it is observed that 10.9% increased BTE when IOP was increased from 205 to 230 bar. This may be due to improved liquid fuel breakup and evaporation; reduced drop let size leading to better atomization and mixing at higher injection pressure (230 bar), which in turn lowers combustion duration with faster rate of heat release. Also, higher injection pressures produce fully developed sprays within a short time; this may help in improving the vaporization process as the surface area of the liquid core increases. However, reduced thermal efficiency was observed at an IOP of 240 bar. This may be due to the fact that, injected fuel penetration was reduced and maximum fuel evaporated may collect on the cylinder wall and may pass away without mixing with air and wet the cylinder wall. Further, it is seen that lower thermal efficiency was recorded for HOME-producer gas operation at lower IOP. Because, at lower IOP (<230 bar), hydraulic flow rate of liquid fuel decreases and droplets size will be larger leading to improper fuel-air mixing rates and hence ignition delay period increases. In addition, diesel-producer gas operation results in to lower BTE at higher IOP (more than 205 bar). It could be due to improper diesel penetration in to the compressed air.

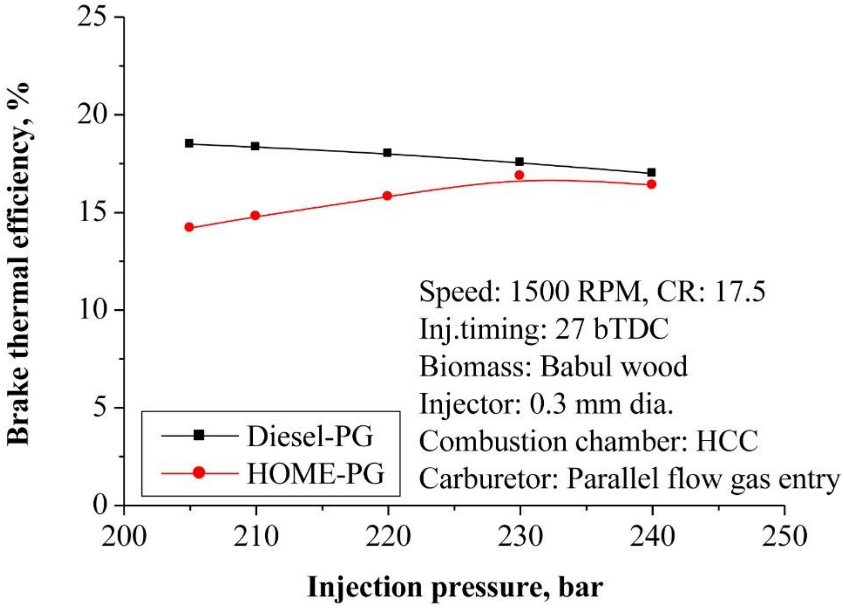


Fig. 6. Variation of BTE with injection pressure.

It is also noticed that equivalence ratio was significantly affected. It could be due to frequent variations in the flow rate of producer gas intake. For HOME-producer gas operation, experimental investigations with different injection pressures (205–240 bar) showed lower thermal efficiency for wide range of equivalence ratios. The air-fuel equivalence ratio was found to be in the range from 0.42 to 0.54 for HOME-producer gas operation compared to 0.59 to 0.63 for diesel-producer gas combination. The BTE obtained for diesel-producer gas and HOME-producer gas operation at the optimized IOPs (205 and 230 bar) were 18.51% and 16.86% respectively. For HOME–producer gas operation, the BTE values at the IOPs of 205, 210, 220, 230 bar and 240 bar were found to be 14.2, 14.8, 15.78, 16.86% and 16.4% respectively.

Effect of IOP on the exhaust gas temperature (EGT) was presented in Fig. 7. For the same IOP, HOME-producer gas operation resulted in higher EGT compared to diesel-producer gas operation. It could be attributed to late burning of HOME-producer gas combination. Also producer gas cannot burnt early as it is a slow burning gas i.e., it requires more time to burn or requires better vaporization, atomization of liquid fuel and mixing. However, HOME-producer gas operation with IOP of 230 bar resulted in lower EGT compared to the operation with low and high IOP. Increased IOP from 205 to 230 bar, 19.5% decreased EGT was observed. At lower IOP, the fuel was injected comparatively with larger droplets. They cannot burn completely during premixed combustion phase leading to burn during diffusion combustion phase. However, at higher IOP (more than 230 bar), the droplet size reduces to smaller size and will have lesser relative velocity. Once its initial velocity is lost they will travel in air and resulting partial combustion. Therefore, HOME-producer gas operation with low and high IOP, the EGT was found to be high. In addition, diesel-producer gas operation results in to higher EGT at higher IOP (more than 205 bar). It could be due to improper diesel penetration in to the compressed air and producer gas mixture. The EGT obtained for diesel-producer gas and HOME-producer gas operation at the optimized IOPs (205 and 230 bar) were 330 °C and 425 °C respectively. For HOME–producer gas operation, the BTE values at the IOPs of 205, 210, 220, 230 bar and 240 bar were found to be 495, 474, 450, 425 °C and 415° respectively.

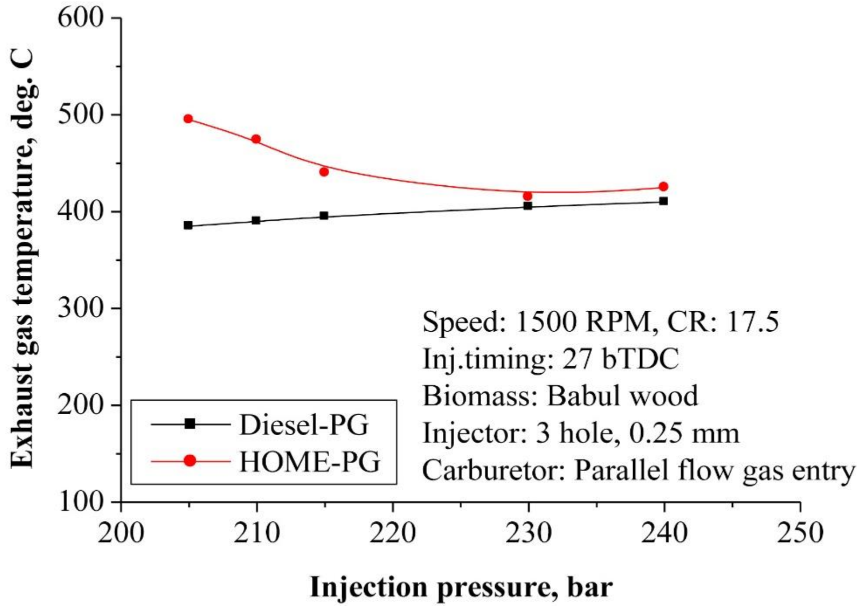


Fig. 7. Variation of exhaust gas pressure with injection pressure.

6.1.2 Emission characteristics

Emissions are formed during combustion of heterogeneous mixture of diesel/HOME-producer gas-air and it depends on the engine design and operating conditions. The emission characteristics of the engine are important from environmental perspective. The emission levels from the engine indicate quality of combustion taking place inside the engine. The exhaust emission levels for the producer gas fueled dual-fuel engine were measured under steady-state conditions. Following section summarizes emission levels observed during diesel/HOME-producer gas combustion.

Fig. 8 presents variations in smoke levels with respect to various injection pressures. As expected that, lowest oxygen and combustion temperature to be a problem for smoke emission levels. The effect of lower temperature is probably one of the main origins to this phenomenon. Being operating conditions are same; diesel-producer gas combination resulted in lower smoke emission levels compared to HOME-producer gas operation. Lower oxidation process, incomplete combustion due to higher viscosity of HOME and lower flame velocity of producer gas may be responsible for this trend. However, at an IOP of 230 bar, HOME-producer gas operation resulted in lower smoke levels compared to the dual fuel operation with IOP of 205, 210, 220 and 240 bar. From the results it is observed that 8.42% decreased smoke levels when IOP was increased from 205 bar to 230 bar. It could be due to the fact that increasing the injection pressure lowers the droplet size leading to better mixing with air and producer gas mixture resulting in improved combustion and flame produced by burning of fuel may reach the entire area of the cylinder. Better burning of fuel combinations during premixed combustion phase due proper utilization of air could be the reason for lower smoke levels at an IOP of 230 bar. Smaller fuel droplets and better mixture quality leading to lesser soot particles, and improved oxidation are also responsible for such observed trend. Another possible reason is that, at higher IOP (230 bar) soot particles may be in the center of cylinder and is advantageous since interactions with the cylinder wall would be lowered. However, if the injection pressure is too high (240 bar) ignition delay become shorter. So, combustion efficiency falls down.

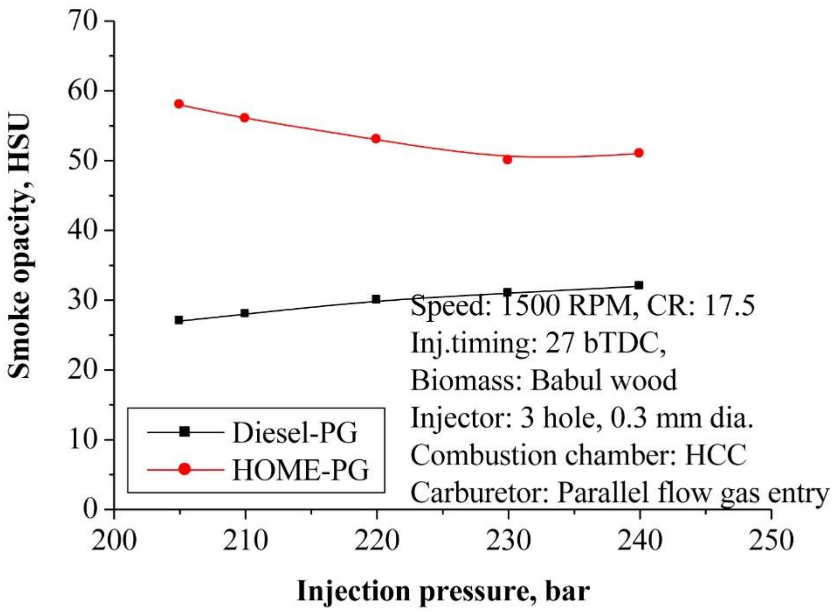


Fig. 8. Variation of smoke opacity with injection pressure.

Therefore, comparatively higher smoke levels were formed due to poor combustion. Further, at an injection pressure less than 230 bar, comparatively larger fuel droplets were formed and became difficult to mix with air leading to partial burning of the fuel along with slow burning producer gas resulting higher smoke levels compared to dual fuel operation with IOP of 230 bar. However, lower soot oxidation is also responsible. In addition, diesel-producer gas operation at higher IOP (>205 bar) leads to increased smoke levels due to reduced delay period, lower mixing rates and lower penetration of diesel into compressed high density air-producer gas mixture due to lower relative velocity of fuel droplets. The smoke opacity obtained for diesel-producer gas and HOME-producer gas operation at the optimized IOPs (205 and 230 bar) were found to be 27 HSU and 48 HSU respectively. For

HOME–producer gas operation, the smoke levels at the IOPs of 205, 210, 220, 230 bar and 240 bar were found to be 58, 54, 52, 47 HSU and 51 HSU respectively.

Fig. 9, Fig. 10 shows the variations of hydrocarbon (HC) and carbon monoxide (CO) emissions for diesel – producer gas and HOME – producer gas operation with respect to various injection pressures. Experimental investigation suggests that HC and CO emissions resulted during combustion because of incomplete combustion of under mixed fuel. For the same IOP, it is observed that HOME-producer gas operation resulted in higher HC and CO levels compared to diesel-producer gas operation over entire pressure range. Experimental investigation for HOME-producer gas operation showed that HC and CO emission levels are decreased by 14.8% and 32.45% respectively when IOP was increased 205–230 bar. It could be attributed to incomplete combustion caused by reduced air induction and lower mixing rates during HOME-producer gas dual fuel operation. For HOME-producer gas operation, increase in injection pressure (IOP of 230 bar) slightly decreases both HC and CO emission levels. This is because, increasing the injection pressure increases the combustion temperature and pressures due to proper utilization of available air and mixing of the fuel combination leading to better combustion. Also, smaller droplets due to the use of high injection pressure (230 bar) results into faster flame propagation due to more complete burning of the fuel combination, helps to burn the entire fuel mixture, hence lower HC and CO emissions were obtained at the IOP of 230 bar compared to 205, 210, 220 and 240 bar. Already presence of CO in the producer gas leads to higher CO levels in the exhaust. Further, the operation with low injection pressure (<230 bar), results into lower combustion temperature due to improper mixing of fuel and air leading to the freezing of the oxidation process. Also, decreased air induction due to dual fuel operation and reduced mixing rates due to decreased hydraulic flow rate of liquid fuel is responsible for this trend. However at 240 bar injection pressure, rapid vaporization of the liquid fuel lowers the penetration leading to improper mixing of fuel combination with air resulting higher emission levels. In addition, diesel-producer gas operation has resulted in higher HC and CO levels at higher IOP of more than 205 bar. This could be due to improper fuel –air mixing due to reduced penetration of diesel fuel. The HC obtained for diesel-producer gas and HOME-producer gas operation at the optimized IOPs (205 and 230 bar) were 33 ppm and 48 ppm respectively. For HOME–producer gas operation, the HC values at the IOPs of 205, 210, 220, 230 bar and 240 bar were found to be 54, 52, 49, 47 ppm and 51 ppm respectively. Similarly, CO obtained for diesel-producer gas and HOME-producer gas operation at the optimized IOPs (205 and 230 bar) were 0.31% and 0.36% respectively. For HOME–producer gas operation, the BTE values at the IOPs of 205, 210, 220, 230 bar and 240 bar were found to be 0.48, 0.44, 0.39, 0.35% and 0.36% respectively.

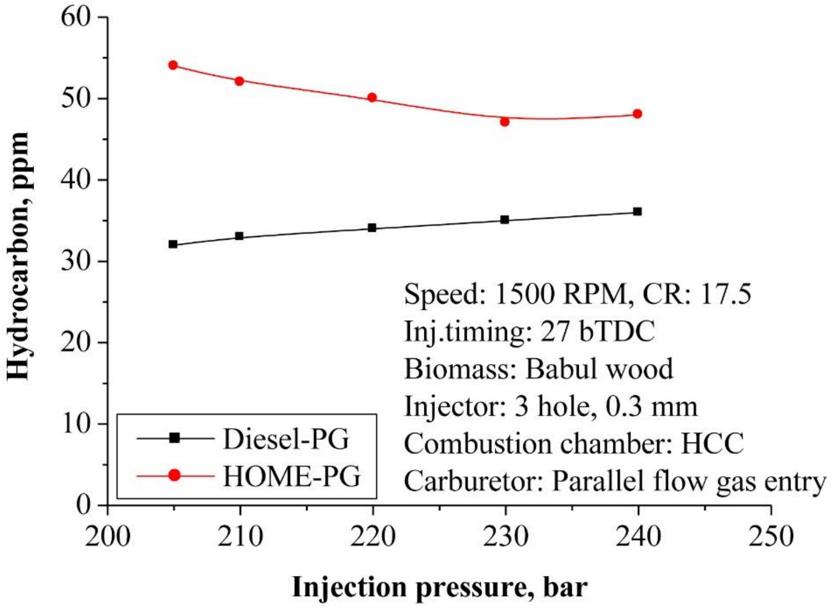


Fig. 9. Variation of hydrocarbon with injection pressure.

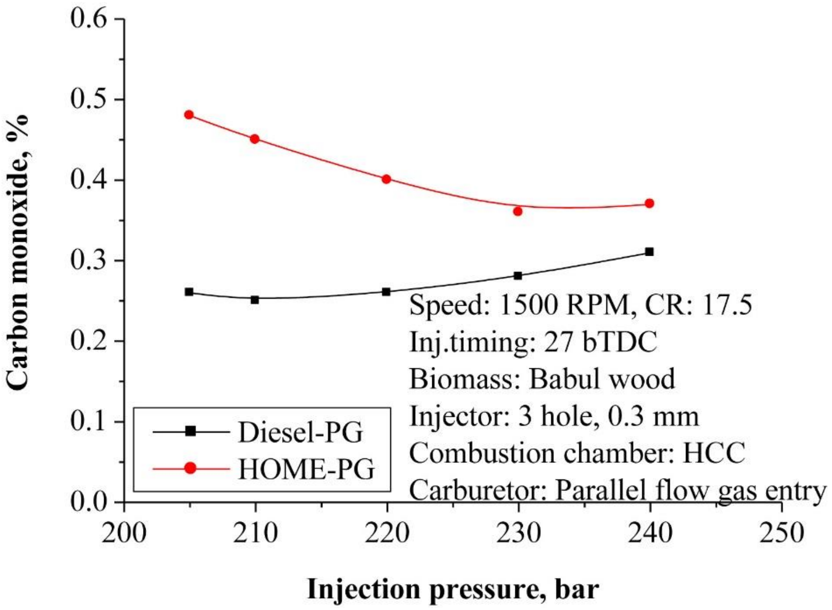


Fig. 10. Variation of carbon monoxide with injection pressure.

The variation of NOx emission levels with brake power is shown in Fig. 11. The NOx emission levels with HOME-producer gas dual fuel operation are found to be lower compared to diesel-producer gas operation. This is because; lower air induction and combustion temperature inside the engine cylinder with HOME-producer gas combination. The lower calorific value of both HOME and producer gas, slow burning nature of producer gas and reduced flame propagation due to incomplete combustion is responsible for this trend. HOME-producer gas operation with IOP of 230 bar showed that increased NOx levels by 18.9% compared to the dual fuel operation with other IOPs tested. This could be attributed to the fact that, improved combustion caused by the use of proper injection pressure (230 bar). Injection pressure of 230 bar determines reduction of HOME droplet's average size leading to better penetration and good diffusion of the HOME and comparatively faster combustion and higher temperatures in the cycle leading to both higher in-cylinder pressure and heat release rate peak at the expenses of higher NOx levels. In addition, diesel-producer gas operation resulted in lower NOx emission levels at higher IOP (>205 bar) due to reduced combustion rate during premixed combustion phase. Reduced relative velocity and penetration of diesel are also responsible for this trend. The NOx emission levels obtained for diesel-producer gas and HOME-producer gas operation at the optimized IOPs (205 and 230 bar) were 110 HSU and 88 HSU respectively. For HOME–producer gas operation, the BTE values at the IOPs of 205, 210, 220, 230 bar and 240 bar were found to be 74, 82, 85, 88 HSU and 86 HSU respectively.

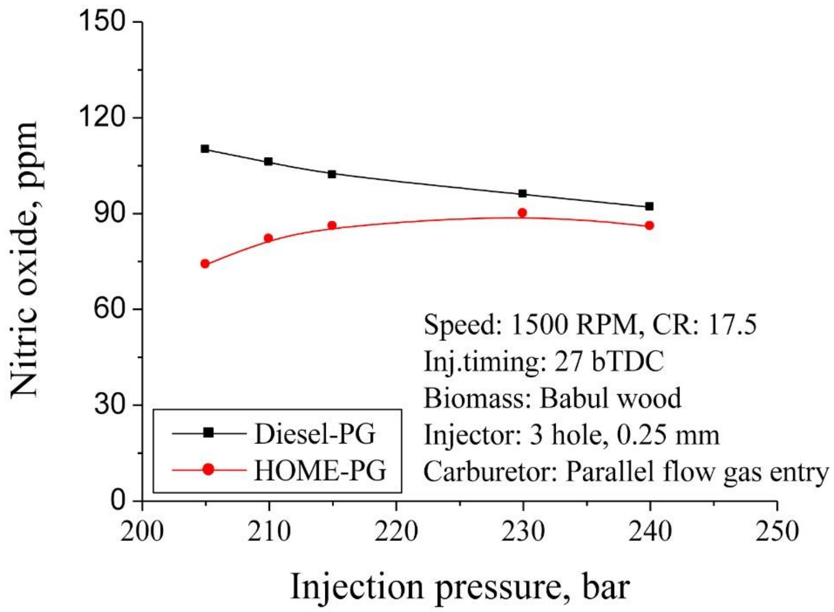


Fig. 11. Variation of nitric oxide with injection pressure.

6.2 Optimization of number of nozzle holes

In this section, effect of number of nozzle holes on the performance of dual fuel engine has been discussed.

6.2.1 Performance characteristics

Fig. 12 shows the effect of number of nozzle holes on the brake thermal efficiency. Higher BTE was observed with 3 hole nozzle for diesel-producer gas and 4 hole nozzle for HOME-producer gas operation. HOME-producer gas fueled dual fuel engine with 4 hole injector resulted in 6.8% and 7.6% increased BTE compared to 3 and 5 hole injector operation respectively. It could be attributed to increased atomization and improved spray dispersion with proper fuel penetration for the 4 hole nozzle compared to 3 and 5 hole nozzles. Also, enhanced liquid fuel breakup leading to smaller fuel droplets with higher dispersion is responsible for this trend. Better relative velocity and good break-up of the droplet with good penetration and better distribution of droplets at delay period leads to better mixing of liquid fuel with air-producer gas mixture. Therefore, better premixed combustion is possible with a four hole nozzle. However, with 5 hole injector, decreased BTE was observed because of incomplete combustion due to higher mass flow rate of liquid fuel inside the combustion chamber leading to larger droplets, low momentum of liquid droplets and lower injection velocity. However, for the same fuel combination, similar result have been observed with 3 hole nozzle. This could be attributed to improper and insufficient fuel injection, which may lead to lower liquid fuel concentration in the air-producer gas mixture resulting lower heat release rate. In general, diesel-producer gas operation results into higher BTE with 3 hole nozzle compared to 4 and 5 hole nozzles. It could be due to increased mass flow rate of fuel inside the combustion chamber leading to improper burning of fuel combination. The BTE obtained for HOME-producer gas operation at nozzle hole 3, 4 and 5 were 16.4, 17.52 and 16.24% respectively compared to 19.05% for diesel–producer gas operation with 3 hole nozzle.

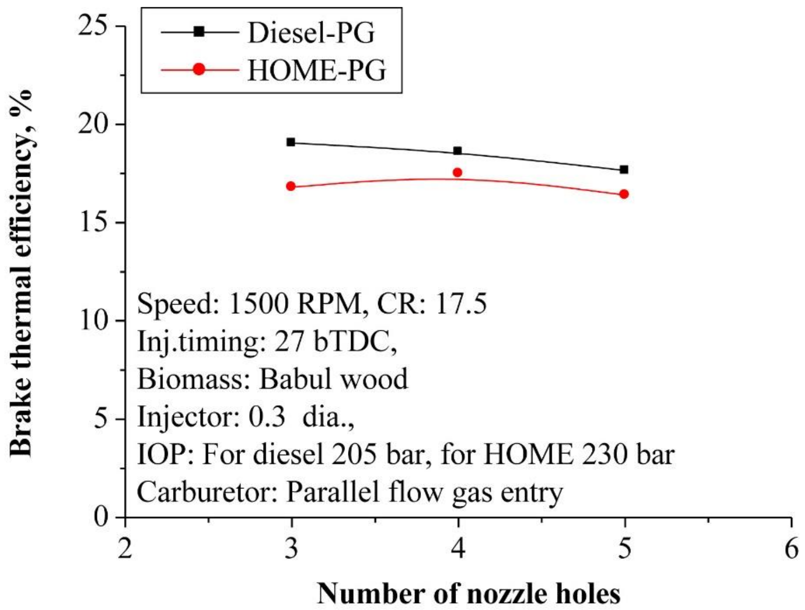


Fig. 12. Variation of brake thermal efficiency with number of holes.

Effect of number of nozzle holes on the exhaust gas temperature (EGT) is presented in Fig. 13. For the same operating conditions, it is observed that diesel producer gas operation resulted in lower EGT compared to HOME-producer gas operation. Compared with 3 and 5 hole nozzle operation, 4 hole nozzle resulted in 6.45% and 11.23% decreased EGT. It could be attributed to the fact that, higher momentum of liquid droplets leads to increased velocity. Therefore, better spray pattern and good penetration of HOME in a compressed air-producer gas mixture leads to better combustion during premixed and rapid combustion phase with 4 hole nozzle during HOME-producer gas operation. Increased air entrainment, higher injection velocity with 4 hole, results in smaller fuel droplet size at delay period and better mixing rates before impinging of liquid fuel on the piston bowl may also be responsible for this trend. However, for the same fuel combination with 5 hole nozzle operation, it resulted in higher EGT due to increased hydraulic flow rate of HOME and improper spray penetration leading to improper air fuel mixing during ignition delay which resulted into lower peak value of premixed combustion and increased diffusion combustion phase. Further, with 3 hole nozzle operation, similar result was observed because of under penetration and burning of fuel during diffusion combustion phase. In general, diesel-producer gas operation, 3 hole nozzle resulted in lower EGT compared to 4 and 5 hole nozzle operation due to better air utilization during premixed combustion phase. The EGT obtained for HOME-producer gas operation at nozzle hole 3, 4 and 5 were 460, 425 and 490 HSU respectively compared to 390 HSU for diesel–producer gas operation with 3 hole nozzle.

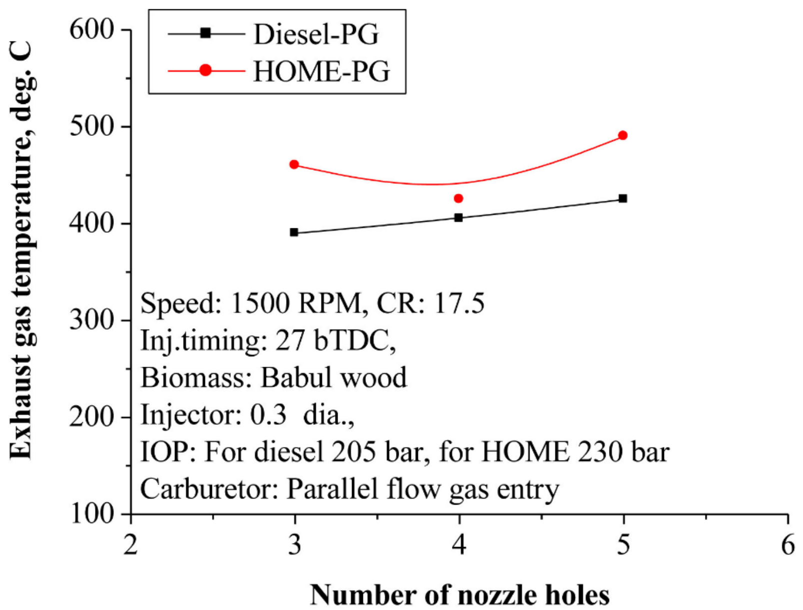


Fig. 13. Variation of exhaust gas temperature with number of holes.

6.2.2 Emission characteristics

Emission levels have been measured during dual-fuel mode of operation with various injector nozzles and are discussed below.

In Fig. 14, variation of smoke opacity for diesel-producer gas and HOME-producer gas operation with different injector nozzle designs are presented. For same nozzle hole operation, smoke opacity with HOME-producer gas combination under dual fuel mode was found to be higher compared to diesel-producer gas combination. This could be attributed to the presence of free fatty acids in HOME and burning of high viscosity fuel (HOME) in presence of slow burning producer gas compared to diesel resulting in higher smoke levels. Lower combustion pressure and temperature during the combustion of HOME-producer gas combination is also responsible for this trend. However, for the same fuel combination, lower smoke levels are observed with 4-hole injector nozzle operation. It could be due to improvement in fuel air mixing. Also, lower wall wetting leads to lesser chance for air-fuel mixture to be locally rich particularly near piston bowl. Furthermore, with 4 hole injector, the spray will disperse the fuel over entire combustion chamber. Such a condition is indeed favorable for suppressing smoke levels. Further, the dual fuel operation with 5 hole injector, the spray temperature where fuel vapor meets with surrounding air is to be lowest among all regions of spray. Therefore, regions with highest temperatures are accordingly diminished with increased hole numbers. Increased amount of injected fuel during the combustion due to increased hydraulic flow rate is also responsible. This is likely to produce considerable amount of smoke at the outset of combustion. Lower combustion temperature during combustion of HOME-producer gas combination reduces the cylinder temperature which could promote soot formation. Also, in case of 3 hole injector operation, the smoke levels were found to be higher due to poor entrainment of surrounding air. In addition, diesel-producer gas operation with 3 hole injector resulted in lower smoke levels compared to the dual fuel operation with 4 and 5 hole injector. It may be due to comparatively better mixing of the fuel combination and better oxidation with 3 hole injector.

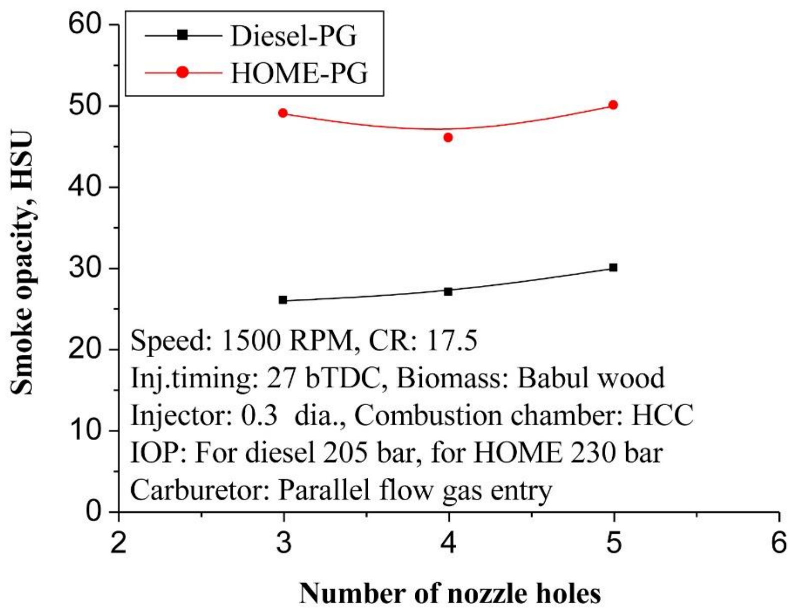


Fig. 14. Variation of smoke opacity with number of holes.

Variation of HC and CO emission levels for diesel-producer gas and HOME-producer gas operation with different injector nozzle designs are presented in Fig. 15, Fig. 16. For the same operating condition, HC and CO levels with HOME-producer gas combination under dual fuel mode of operation were found to be higher compared to diesel-producer gas combination. Results of

HOME-producer gas operation showed HC emission levels were decreased by 8.89% and 15.9% with 4 hole nozzle compared to 3 and 5 hole nozzle operation respectively and CO levels decreased by 16.6% and 24.6% with 4 hole nozzle compared to 3 and 5 hole nozzle operation respectively. This could be attributed as incomplete combustion and presence of CO already in the producer gas. Lower combustion pressure and temperature and lower oxygen during the combustion of HOME-producer gas are also responsible for this trend. During 4 hole operation the HOME disperses over entire area of combustion zone, hence slightly better entertainment of air leading to slightly better combustion and lowers both HC and CO emission levels. However, at 5 hole injector nozzle operation, HC and CO emissions are caused mainly due to reduced velocity of fuel, which is not sufficient to penetrate the HOME into compressed air-producer gas mixture leading to improper air-fuel mixing. Also, lack of fuel vaporization and atomization is responsible for this trend. Whereas at 3 hole injector operation, the hydraulic flow rate was reduced. This leads to reduced air-fuel mixture leading to incomplete combustion of the fuel combination. In addition, diesel-producer gas operation with 3 hole injector resulted in lower HC and CO levels compared to the dual fuel operation with 4 and 5 hole injector. It may be due to comparatively better spray formation leading to better mixing of the fuel combination and oxidation with 4 hole injector. The HC emission levels obtained for HOME-producer gas operation at nozzle hole 3, 4 and 5 were 48, 44 and 51 ppm respectively compared to 32 ppm for diesel–producer gas operation with 3 hole nozzle. Similarly, CO emission levels obtained for HOME-producer gas operation at nozzle hole 3, 4 and 5 were 0.42, 0.36 and 0.38% respectively compared to 0.32% for diesel–producer gas operation with 3 hole nozzle.

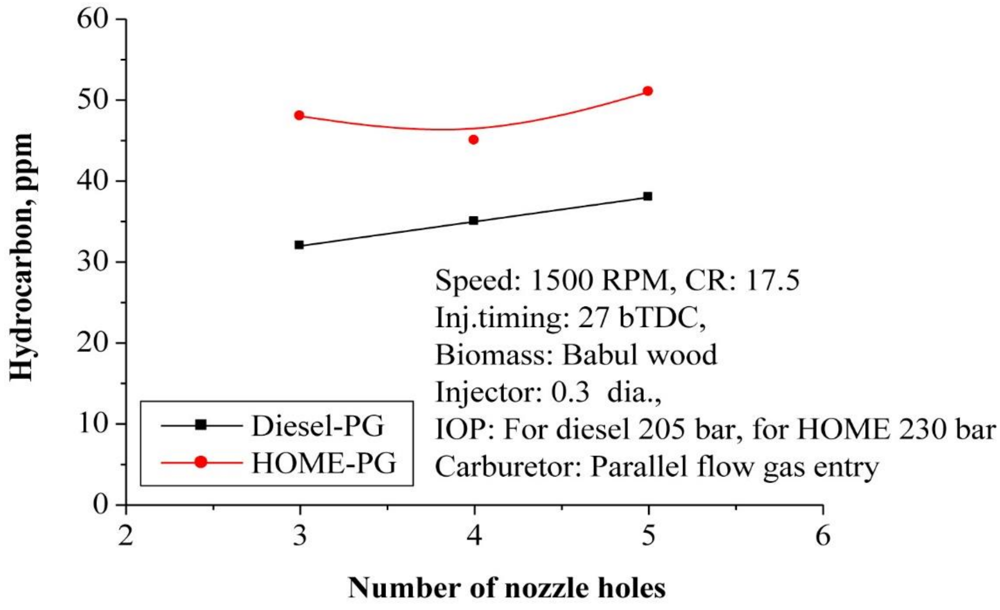


Fig. 15. Variation of hydrocarbon with number of holes.

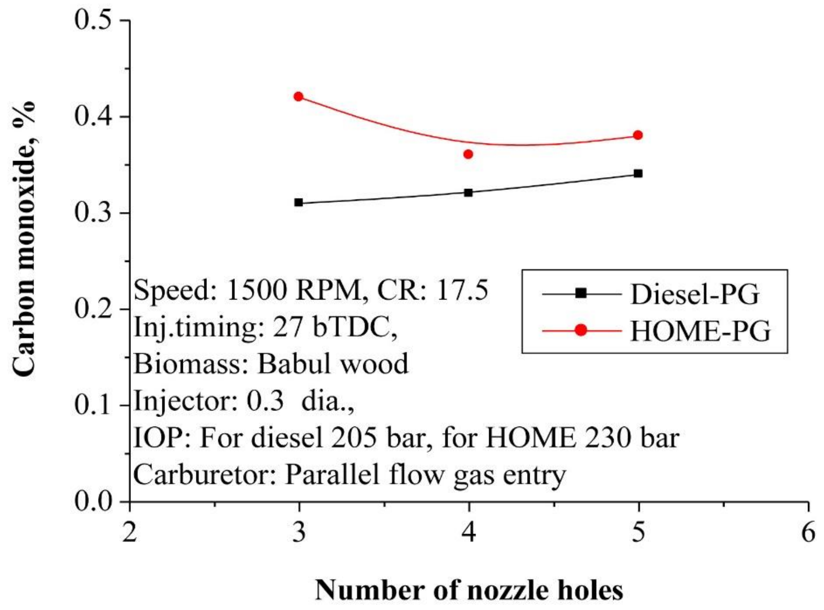


Fig. 16. Variation of carbon monoxide with number of holes.

Variation of NOx emission levels for diesel-producer gas and HOME-producer gas operation with different injector nozzle designs are presented in Fig. 17. NOx levels with HOME-producer gas combination under dual fuel mode were found to be lower compared to diesel-producer gas combination. Higher viscosity, presence of free fatty acids and lower calorific value of HOME is responsible for this trend. Lower combustion temperature and improper utilization of air during combustion may also be responsible for this observed trend. However, HOME-producer gas operation with 4 hole injector nozzle resulted in higher NOx levels in the exhaust. It could be due to comparatively better atomization and mixing of fuel with air resulting into improved heat release rate, peak combustion pressure and temperature. Results showed that increased NOx levels by 9.4% and 14.2% with 4 hole nozzle compared to 3 and 5 hole nozzle operation respectively. Further, HOME-producer gas operation with 5 hole injector nozzle, high temperature zones are diminished due to improper combustion of HOME along with slow burning producer gas. Whereas, with 3 hole injector operation, lower flow rate of HOME leads to less fuel preparation for the combustion. Therefore, from 3 to 5 hole injector nozzles, lower NOx emission level is expected as and when combustion proceeds. In addition, diesel-producer gas operation with 3 hole injector resulted in higher NOx levels compared to the dual fuel operation with 4 and 5 hole injector. Comparatively better mixing of the diesel with air-producer gas mixture and increased combustion rate with 3 hole injector is responsible. The NOx emission levels obtained for HOME-producer gas operation at nozzle hole 3, 4 and 5 were 86, 94 and 82 ppm respectively compared to 110 ppm for diesel–producer gas operation with 3 hole nozzle.

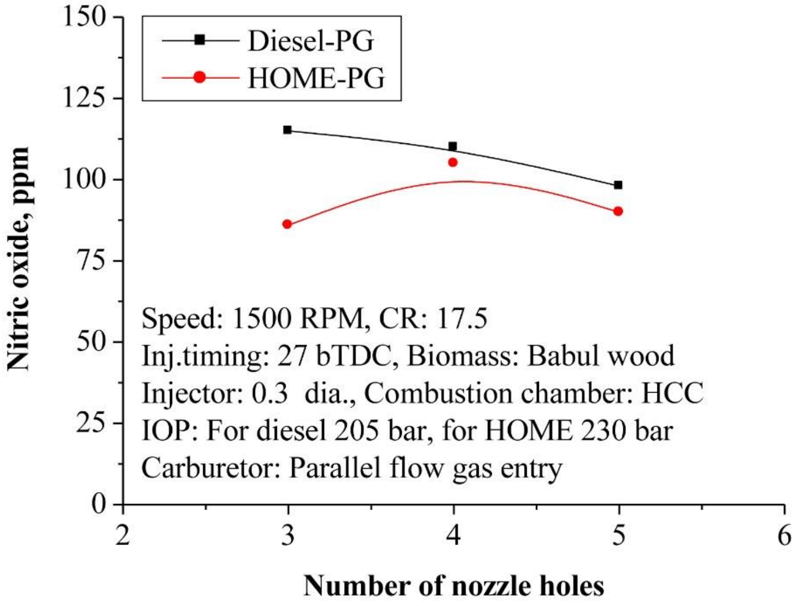


Fig. 17. Variation of nitric oxide with number of holes.

6.3 Optimization of nozzle hole size

In this section, effect of nozzle hole size on the performance of dual fuel engine has been presented.

6.3.1 <u>Performance characteristics</u>

The effect of nozzle hole size on the brake thermal efficiency is presented in Fig. 18. For the same operating conditions, results showed diesel-producer gas operation always resulted in higher thermal efficiency at all the nozzles tested compared to HOME-producer gas operation. It could be due to difference in the fuel properties for variations in the thermal efficiency. Total mass of fuel injected increases with increase in nozzle hole diameter. HOME-producer gas operation with 0.25 mm hole size resulted in 8.1 and 2.8% increased thermal efficiency compared to 0.2 and 0.3 mm nozzle operation respectively. This is an important result with a 0.25 mm hole diameter. Reduced chances of piston and wall impingements, smaller diameter droplets with proper penetration of HOME and spray dispersion in the compressed air-producer gas mixture can be achieved, thus decreasing the incomplete combustion tendency. Enhanced liquid breakup leads to smaller droplets and thus higher dispersion may be possible with 0.25 mm size nozzle. Smaller droplets and proper dispersion imply enhanced fuel–air mixing with important consequences for the flame structure and emissions. Improved interaction between fuel droplets and air and increased turbulence levels with 0.25 mm nozzle hole is also responsible for this trend. However, it is seen that thermal efficiency was decreased with 0.2 and 0.3 mm nozzle hole diameter. It could be due to decreased liquid penetration due to reduced momentum of fuel droplets and enhanced primary breakup caused by the cavitations and turbulence generated inside the nozzle for the 0.2 mm nozzle hole. The enhanced breakup leads to smaller sauter mean diameter, and thus lowers the penetration. Similar results were obtained with 0.3 mm nozzle operation. HOME-producer gas operation with 0.3 mm nozzle hole yields comparatively lower mixing rate, which is due to the higher fuel injection rate, decreased velocity of HOME and resulting into comparatively larger droplets [49]. Therefore, higher velocity of fuel is essential to penetrate into high density compressed air-producer gas mixture. Droplet size of HOME may be higher for 0.25 mm nozzle diameter and is expected since the density at the exit of nozzle was increased. From the results, it could be concluded that, increased mass flow rate of fuel during the ignition delay leads to improper mixing and chemical kinetics. Consequently, the vaporization and fuel-air mixing rate are reduced, and hence ignition occurs during downstream of the combustion process. In addition, with 0.25 mm hole diameter and at 205 bar IOP, diesel-producer gas operation resulted in comparatively better thermal efficiency. The BTE obtained for HOME-producer gas operation with a nozzle size of 0.2, 0.25 and 0.3 mm were found to be 17.2, 18.6% and 18.1% respectively compared to 19.2% for diesel–producer gas operation with 0.25 mm nozzle size.

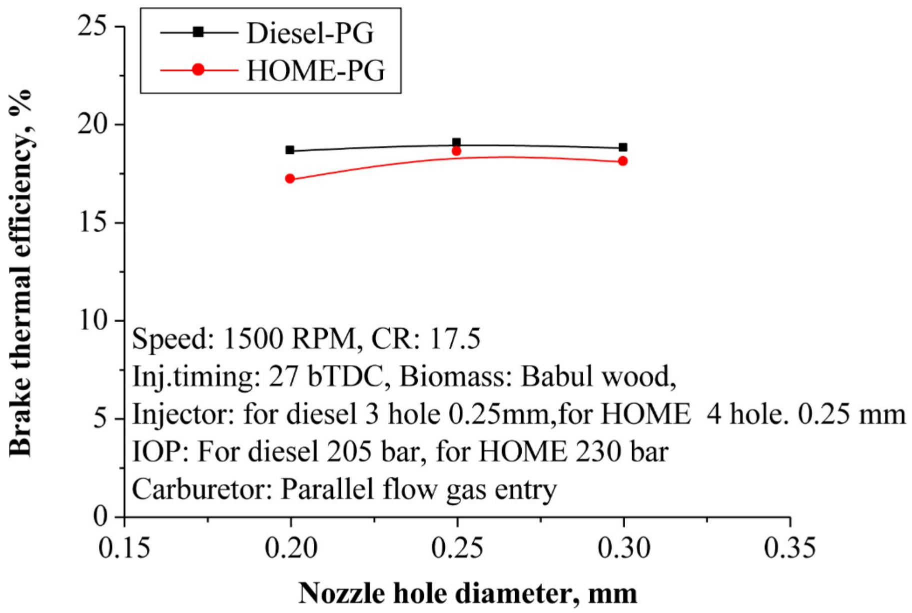


Fig. 18. Variation of brake thermal efficiency with nozzle hole diameter.

6.3.2 Emission characteristics

Various regulated emission levels are measured during dual-fuel mode of operation and are discussed below.

The effect of nozzle hole size on the smoke opacity is presented in Fig. 19. For the same operating conditions, diesel-producer gas operation always resulted in lower smoke emission levels at all the nozzles tested compared to HOME-producer gas operation. It could be due to incomplete combustion of HOME in presence of slow burning producer gas. However, smoke levels of HOME-producer gas combustion can be reduced to some extent if mixing of injected HOME with air improved. This is achieved by using suitable nozzle hole size with appropriate injection pressure. It is observed that the HOME-producer gas operation with 0.25 mm hole size resulted in 4.5% and 13% decreased smoke levels compared to 0.2 and 0.3 mm respectively. HOME-producer gas operation with a 0.25 mm hole diameter resulted in lower smoke levels due to the result of smaller droplets with proper spray penetration of HOME and spray dispersion in the compressed air-producer gas mixture leading to improved air-fuel mixing rate. For the nozzle hole size of 0.3 mm, increased hydraulic flow rate from the holes of an injector nozzle resulting lower penetration of injected HOME in the gas mixture and lowers flow velocity and increases droplet size. This leads to incomplete combustion of fuel combination used resulting heavy smoke emission levels. Fuel combination being same, with 0.2 mm nozzle hole dual fuel operation, the enhanced fuel breakup leads to smaller sauter mean diameter, and leads to lower penetration resulting improper combustion and higher smoke emission levels [49]. The smoke levels obtained for HOME-producer gas operation with a nozzle size of 0.2, 0.25 and 0.3 mm were found to be 48, 46 and 52 HSU respectively compared to 26 HSU for diesel–producer gas operation with 0.25 mm nozzle size.

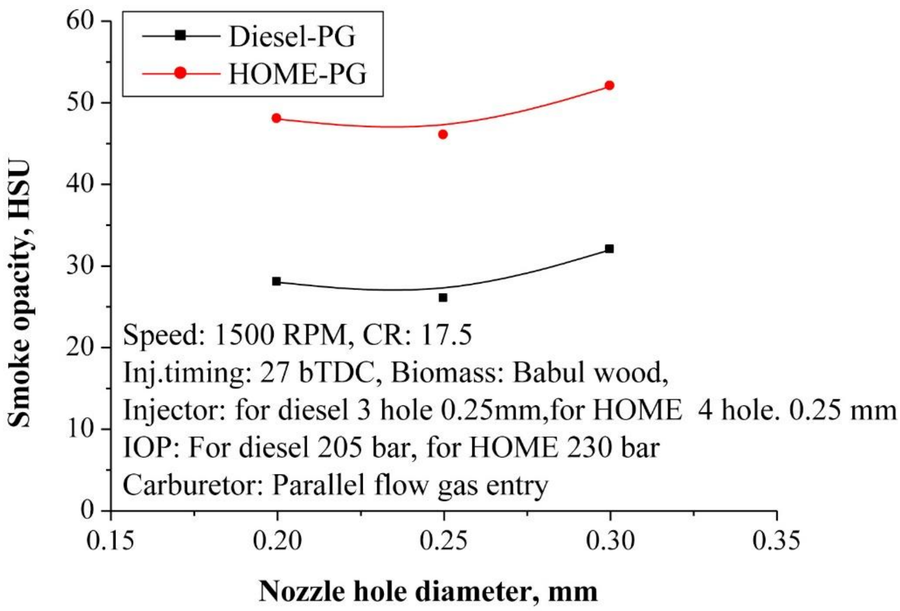


Fig. 19. Variation of smoke opacity with nozzle hole diameter.

Fig. 20, Fig. 21 presents the effect of nozzle hole size on the HC and CO emission levels. Experimental investigation on dual fuel engine with diesel-producer gas operation showed lower HC and CO emission levels compared to HOME-producer gas operation. It could be attributed to the presence of free fatty acids and higher viscosity of HOME leading to incomplete combustion of HOME in presence of slow burning producer gas. Another possible reason for the higher HC and CO emissions in the exhaust is mainly due to the presence of CO in the producer gas. It is known that combustion is characterized by a rich premixed and diffusion flame. Most of the unburned HC and CO levels are produced in the rich premixed flame in turn they strongly depend on fuel–air mixing and ignition behavior [49]. However, during HOME-producer gas operation with 0.25 mm nozzle hole operation, both HC and CO emission levels were found to be lower compared to 0.2 and 0.3 mm nozzle hole size operation. For the HOME-producer gas operation with 0.25 mm nozzle showed 9.5% and 19.01% decreased HC levels and 12.2% and 21.2% decreased CO levels compared to the dual fuel operation with 0.2 and 0.3 mm nozzle respectively. This could be attributed to the fact that, fuel may be injected closer to TDC with 0.25 mm hole nozzle in a dense air-producer gas mixture resulting enhanced premixed combustion. Higher spray penetration and better air utilization is also responsible for this trend of result. Whereas, during 0.3 nozzle hole size operation, the small portion of the fuel which is injected at low pressure at the end of injection, may cause increased HC and CO emissions, i.e., HC and CO emission is mainly caused due to low velocity of fuel due to larger diameter of nozzle hole (0.3 mm) which is not sufficient to penetrate the fuel into air and induced improper air-fuel mixing. Similarly, results were observed for 0.2 mm hole nozzle operation. This could be due to the fact that higher viscosity fuel (HOME) requires comparatively larger nozzle hole size (0.25 mm). Hence lower quantity of fuel may be injected with 0.2 mm nozzle hole size resulting improper mixing of air and fuel. The HC levels obtained for HOME-producer gas operation with a nozzle size of 0.2, 0.25 and 0.3 mm were found to be 46, 41 and 50 ppm respectively compared to 32 ppm for diesel–producer gas operation with 0.25 mm nozzle size. Similarly, CO levels obtained for HOME-producer gas operation with a nozzle size of 0.2, 0.25 and 0.3 mm were found to be 0.37, 0.33 and 0.4% respectively compared to 0.28% for diesel–producer gas operation with 0.25 mm nozzle size.

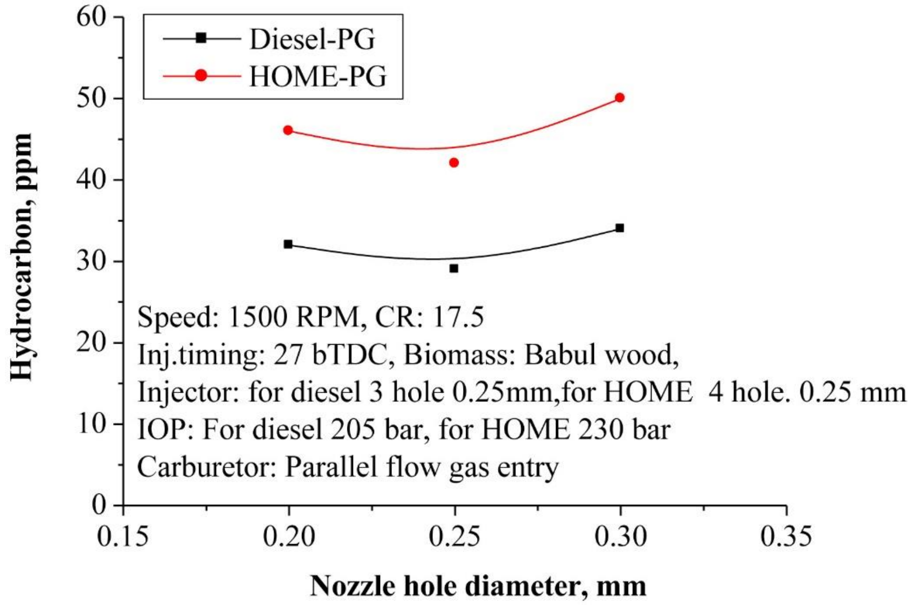


Fig. 20. Variation of hydrocarbon with nozzle hole diameter.

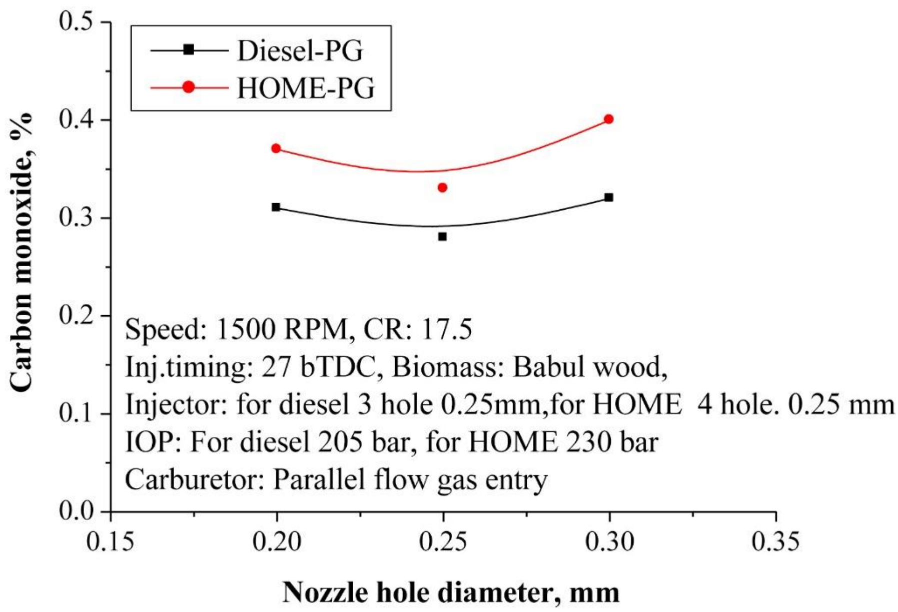


Fig. 21. Variation of carbon monoxide with nozzle hole diameter.

The effect of nozzle hole size on the NOx emission levels were presented in Fig. 22. For the same operating conditions, diesel-producer gas operation always resulted in higher NOx emission levels at all nozzles used compared to HOME-producer gas operation. It could be due to better burning of diesel along with producer gas during premixed combustion phase. During combustion of HOME-producer gas combination, 0.25 mm hole nozzle results into 9.1% and 22.5% increased NOx levels compared to 0.2 and 0.3 mm hole nozzle respectively. It could be due to better burning fuel combination during premixed combustion phase. However, quantity of HOME injected with 0.3 mm nozzle hole is comparatively higher, hence most of the fuel burns in the diffusion combustion phase rather than premixed combustion phase during HOME-producer gas operation. The NOx levels obtained for HOME-producer gas operation with a nozzle size of 0.2, 0.25 and 0.3 mm were found to be 88, 96 and 78 ppm respectively compared to 115 ppm for diesel–producer gas operation with 0.25 mm nozzle size.

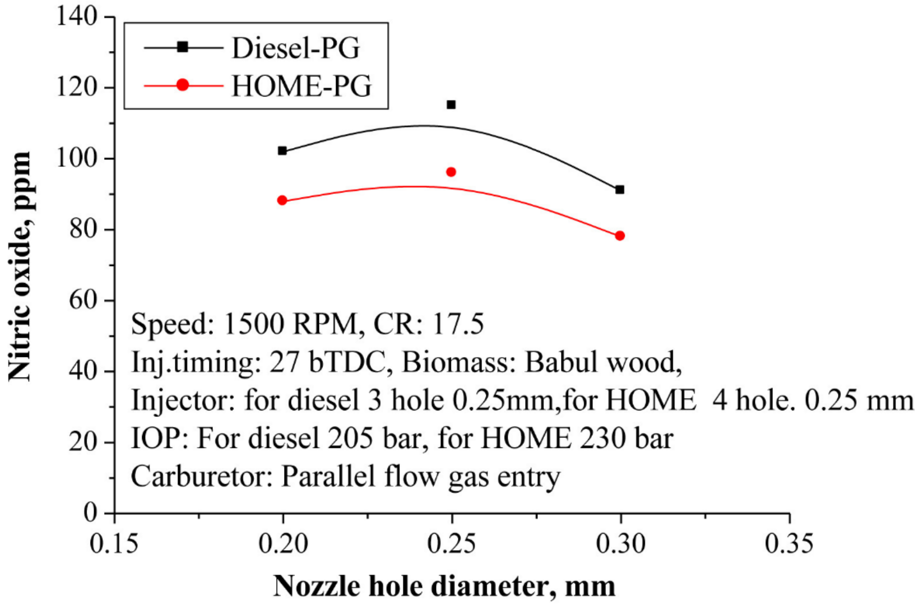


Fig. 22. Variation of nitric oxide with nozzle hole diameter.

6.4 Optimization of combustion chamber

In this section, effect of combustion chamber configuration on the performance of dual fuel engine has been presented.

6.4.1 Performance characteristics

Fig. 23 displays a variation of brake thermal efficiency (BTE) with brake power. For the same operating conditions, results showed higher BTE for diesel-producer gas operation compared to HOME-producer gas combination. Fuel properties are responsible for the observed trend. The study with different combustion chamber shapes show that HOME-producer gas operation with reentrant combustion chamber (RCC) resulted in better performance compared to dual fuel operation with HCC. RCC improves BTE by 3.01% compared to the dual fuel operation with HCC. It may be due to the fact that, the RCC prevents the flame from spreading over to the squish region resulting in better mixture formation of HOME along with producer gas – air combination. Based on the results, it is observed that the RCC has an ability to direct the flow field inside the sub volume at all engine loads and therefore substantial differences in the mixing process may not be present [9], [35], [36]. Increased flame propagation may also be responsible for this trend. Diesel engine operated on HOME-producer gas combination with HCC and under constant operating parameters, combustion is initiated much before TDC is reached. This increases the compression work and more heat loss and thus reduces BTE of dual fuel engine. HCC may also cause an increase of ignition delay during the combustion of HOME-producer gas combination that determines a reduction in BTE. Therefore the use of RCC for the combustion of HOME-producer gas combination, results into enhanced combustion during the expansion stroke, thus preventing the diffusion of the flame in the squish region and giving better performance. In addition, it is observed that the combustion at lower load was more erratic. It could be attributed to reduction in pilot fuel quantity. Hence liquid fuel injection was increased slightly to achieve better combustion of producer gas. However, at higher loads, the fuel injection was reduced in order to achieve better fuel saving and smooth engine operation. For the dual fuel operation with RCC, equivalence ratio for HOME-producer gas operation was found to be 0.59 compared to 0.65 for diesel-producer gas operation at 80% load.

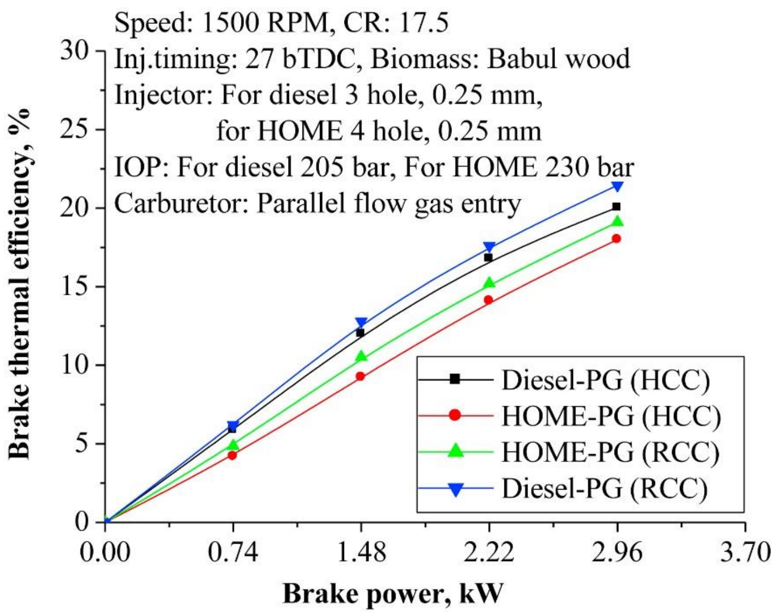


Fig. 23. Variation of brake thermal efficiency with brake power.

It is observed that reduced amount of pilot injection causes reduction in the ignition sources leading to improper combustion or misfiring especially at lower load condition. Therefore reduction in pilot quantity below certain limit decreases the path that flame needs to propagate to consume all premixed mixture. In addition, it is observed that, lower flow rate of producer gas may have no appreciable effect on combustion, but at higher flow rate of producer gas, the air-fuel equivalence ratio was significantly affected. Hence it lowers the overall performance of engine. The BTE obtained for HOME-producer gas operation with a HCC and RCC were found to be 18.02% and 19.1% respectively compared to 21.45% for diesel–producer gas operation with RCC.

6.4.2 Emission characteristics

Different emission parameter measurements during the dual-fuel mode of operation are discussed below.

Fig. 24 presents the variation of smoke opacity with brake power. Dual fuel operation with same combustion chambers, lower smoke levels were observed for diesel-producer gas combination compared to HOME-producer gas operation. It could be attributed to improper mixing rates and reduced oxidation during combustion of HOME-producer gas combination. However the lower air-fuel equivalence ratio obtained for HOME-producer gas operation may be the reason for such observed trend. For the same operating conditions, HOME-producer gas operation with RCC has lower smoke opacity compared to the operation with HCC. It may be due to the fact that, improved air –fuel mixing and better air utilization caused by the optimum turbulence in the combustion chamber. This factor results in better combustion and oxidation of the soot particles which further reduces the smoke emission levels. Increased turbulent kinetic energy during the use of RCC compared to the operation with HCC is also responsible for this trend. However, literatures suggest that narrow width have a higher unburned fuel-air mixture region, and thus would have higher smoke emissions. But with slightly wider combustion chamber, lower smoke levels can be obtained [33]. The smoke levels obtained for HOME-producer gas operation with a HCC and RCC were found to be 57 HSU and 48 HSU respectively compared to 24 HSU for diesel–producer gas operation with RCC.

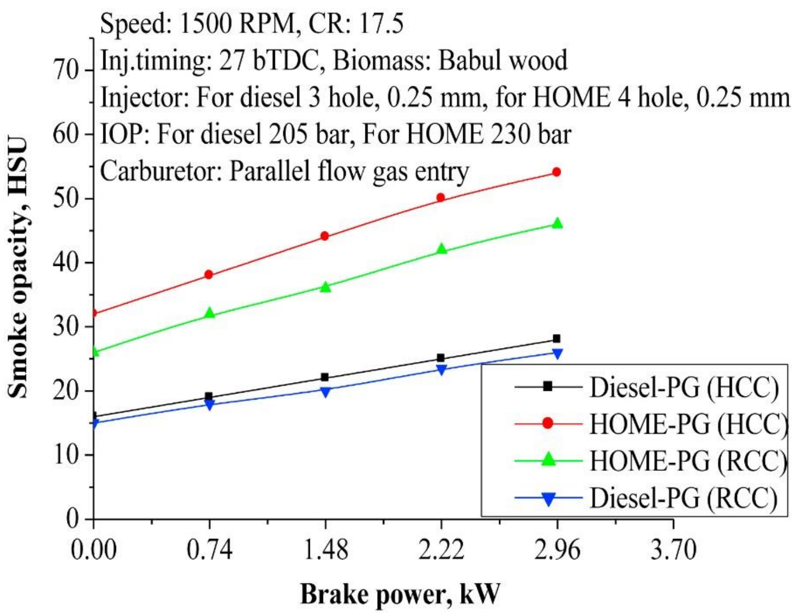


Fig. 24. Variation of smoke opacity with brake power.

Fig. 25, Fig. 26 shows the variation of hydrocarbon (HC) and carbon monoxide (CO) emission levels for diesel–producer gas operation with all loads. Dual fuel operation with similar combustion chambers, higher HC and CO emission levels are observed for HOME-producer operation compared to diesel-producer gas operation. It could be due to incomplete combustion of the HOME-producer gas combination caused by the improper air utilization during combustion. In case of dual fuel mode of operation, incomplete combustion is mainly due to the replacement of air by producer gas. This factor could affect the air-fuel equivalence ratio significantly. Hence, lower oxidation rate, combined effect of lower calorific value of HOME and producer gas, lower adiabatic flame temperature of producer gas and higher viscosity of HOME and lower mean effective pressure are responsible for higher HC and CO emission levels. From the Fig. 24, Fig. 25, it was noticed that re-entrant combustion chamber (RCC) emit lower HC and CO levels compared to the operation with HCC. This was due to better combustion of HOME-producer gas combination as a result of improved swirl and squish motion of air during dual fuel operation with RCC. Proper utilization of oxygen present in the HOME could be the reason for higher BTE during combustion. However, basic combustion chamber (HCC) may not contribute to the proper mixing of fuel combinations leading to incomplete combustion during dual fuel operation. It may be due to confinement in the inferior part of the bowl by the vortex generated with HCC configuration. Also presence of CO in the producer gas may further add to this trend. The HC levels obtained for HOME-producer gas operation with HCC and RCC were found to be 52 ppm and 46 ppm respectively compared to 32 ppm for diesel–producer gas operation with RCC. Similarly, CO levels obtained for HOME-producer gas operation with a HCC and RCC were found to be 0.34% and 0.28% respectively compared to 0.23% for diesel–producer gas operation with RCC.

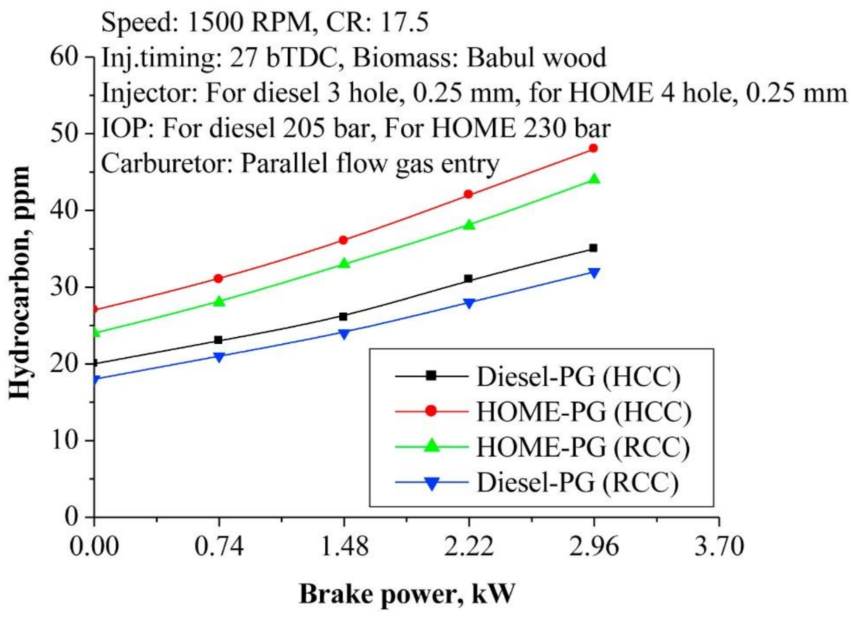


Fig. 25. Variation of hydrocarbon with brake power.

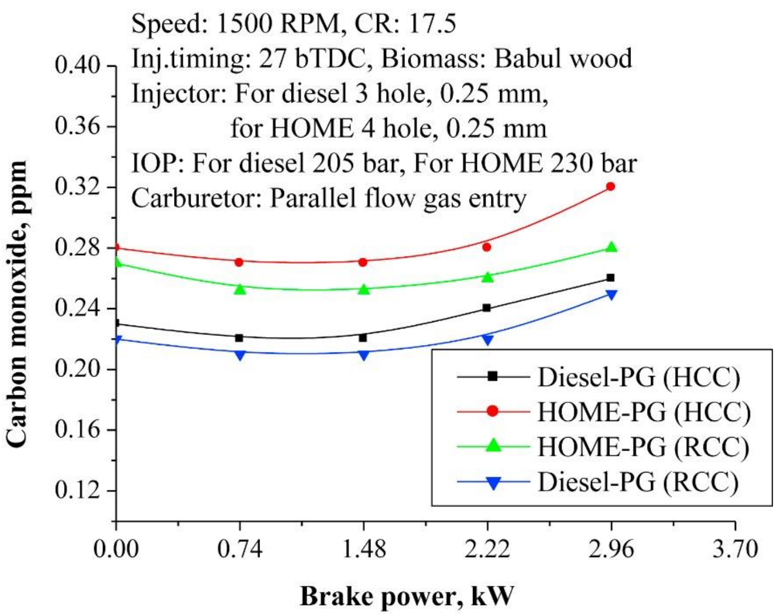


Fig. 26. Variation of carbon monoxide with brake power.

When combustion chambers are same, NOx emission levels were found to be higher in diesel-producer gas dual fuel operation compared to HOME-producer gas dual fuel operation over the entire load range (Fig. 27). This is because of higher heat release rate during premixed combustion phase which occurs with diesel-producer combination. It could also be due to the fact that during HOME-producer gas operation with HCC, the lower utilization of oxygen during combustion is also responsible for the observed trend. However, for the same HOME– producer combination, slightly higher NOx levels have been resulted from the dual fuel operation with RCC compared to the operation with HCC. This could be attributed to the slightly better combustion occurring due to homogeneous mixing and improved air utilization caused by better squish and swirl, and larger part of combustion which occurs just before top dead center. Presence of oxygen in HOME is better utilized when dual fuel operation was used with RCC. Therefore, it has resulted in higher peak cycle temperature. The NOx levels obtained at 80% load for HOME-producer gas operation with a HCC and RCC were found to be 96 ppm and 102 ppm respectively compared to 118 ppm for diesel–producer gas operation with RCC.

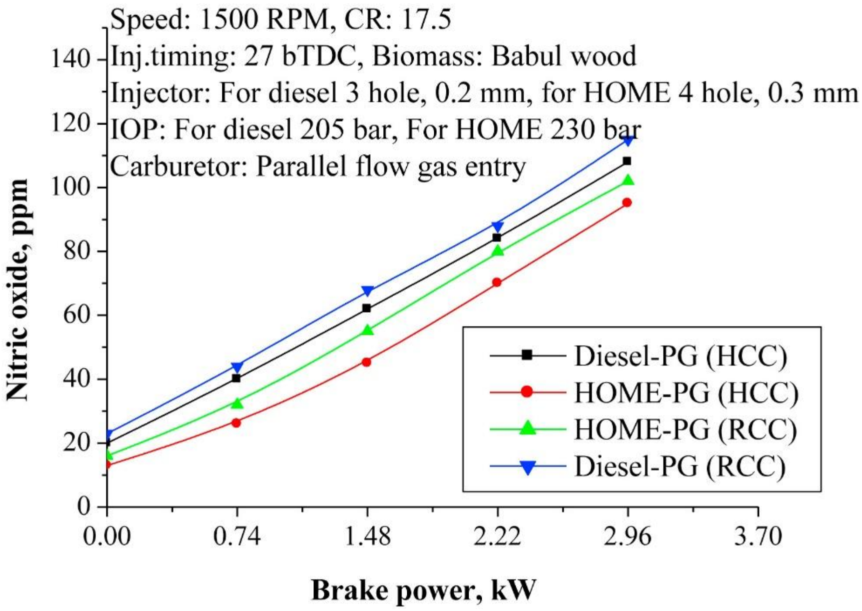


Fig. 27. Variation of nitric oxide with brake power.

Fig. 28 presents the fuel substitution for dual fuel operation at different power outputs. For the same operating conditions, Fuel substitution values were found to be higher for diesel-producer gas operation compared to HOME-producer gas combination. Injected fuel properties such as cetane number, viscosity and calorific value may be considered as responsible for the observed trend. It is observed that for HOME-producer gas combination, fuel substitution was higher for RCC operation compared to HCC. Better utilization of air because of higher squish and swirl leads to slightly better combustion of high volatile fuel with producer gas. Gaseous fuel substitution was found to be higher at higher load due to control of liquid fuel supply properly. However, producer gas substitution at lower load was lower. One possible contributing factor to improve the combustion at low load is to reduce leanness of the fuel combination. Hence liquid fuel supply was increased slightly at low load. This can lead to better combustion of producer gas. In this present work, the fuel combinations selected for the engine study like biodiesel and producer gas both being renewable fuels gives freedom in controlling fuel substitution ratio. The fuel substitution values obtained for HOME-producer gas operation with a HCC and RCC were found to be 49% and 53% respectively compared to 65% for diesel–producer gas operation with RCC.

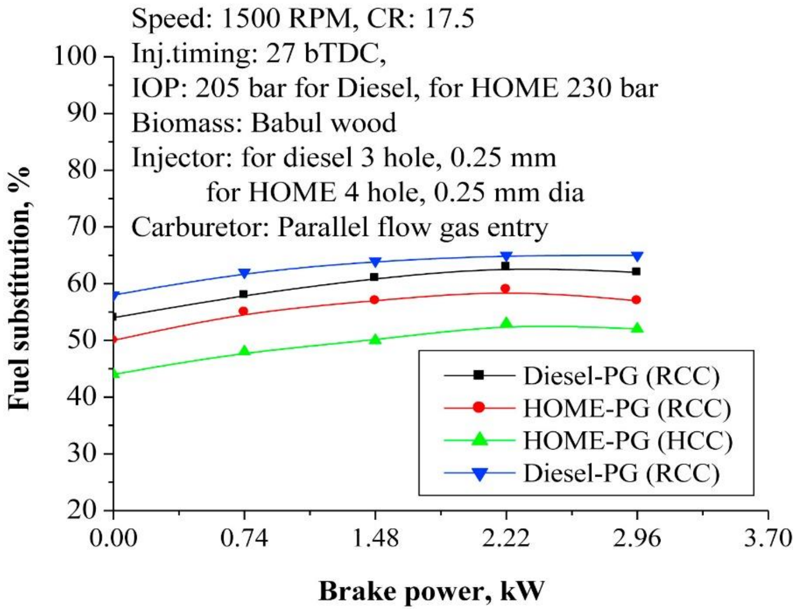


Fig. 28. Variation of fuel substitution with brake power.

6.4.3 Combustion characteristics

Combustion of liquid fuel taking place in a diesel engine differs when gaseous fuels are used, and it depends on the air–fuel mixture quality. Use of re-entrant combustion chamber (RCC) in a direct injection (DI) diesel engine strongly affects the fuel distribution and air–fuel mixing in a combustion chamber and hence affects the combustion of fuel combination. The different combustion parameters for dual-fuel mode of operation are summarized below.

The variation of ignition delay with brake power for HCC and RCC configurations is shown in Fig. 29. The delay period was determined by the period between the start of the injection and rapid pressure rise timing on the pressure curve. It is inferred that ignition delay, decreases with an increase in brake power for both configurations. This could be attributed to increase in in-cylinder gas temperature due to increased amount of fuel being burnt inside the cylinder. The ignition delay is calculated based on the static injection timing. Dual-fuel operation with HOME–producer gas combination and different combustion chamber shows variations in the ignition delay. It could be due to the large amount of producer gas taking part in the combustion phase. Variations in the air–producer gas mixture and lower mixing rates are also responsible for this trend. For the same operating conditions, it is observed that ignition delay for diesel-producer gas operation is lower than HOME-producer gas operation. For dual fuel operation, the ignition delay is defined with two parts that contains diesel ignition and producer gas ignition. The diesel with 3 hole nozzle ignited at a faster rate and then caused a shorter duration for producer gas ignition. This is because of the diesel break up has been favored with 3 hole with 0.3 mm nozzle compared to 4 and 5 hole nozzle due to higher injection velocity. Therefore, air-fuel mixing will enhance in ignition delay. This fact results into lower ignition delay for diesel-producer gas operation. However, for the same fuel combination, ignition delay for HOME-producer gas combination with RCC operation was found to be lower compared to HCC engine operation. Increased flame velocity due to better burning of fuel combination may be responsible. Improved air utilization caused by the development of better squish due to use of RCC further add to this trend. Ignition delay was decreased with RCC as a result of increased pressure and temperature due to proper utilization of air and also due to improved squish and swirl during premixed combustion phase. This could also be due to fast and complete burning of the charge because flow of high velocity flames throughout the combustion chamber. In general, temperature drop due to evaporation and mixing of the fuel combination may take place and it depends on the type and quantity of fuel used during delay period. But, it can be recovered to some extent when RCC was used. The ignition delay obtained for HOME-producer gas operation with a HCC and RCC were found to be 11.8 deg. CA and 11.4 deg. CA respectively compared to 9.8 deg. CA for diesel–producer gas operation with RCC .

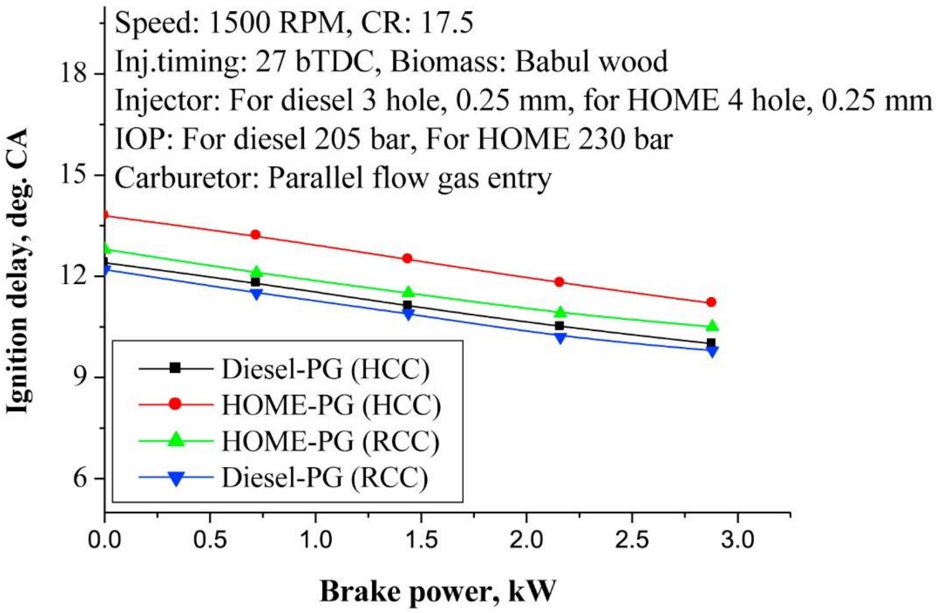


Fig. 29. Variation of ignition delay with brake power.

The combustion duration shown in Fig. 29 was calculated based on the duration between the start of combustion and 90% cumulative heat release. The combustion duration increases with increase in the power output. This is due to increase in the quantity of fuel injected. For the HOME-producer gas dual fuel combination with RCC, reduced combustion duration was observed. This could be due to proper mixing of air-fuel with improved premixed combustion. Dual fuel operation with basic combustion chamber (HCC), the second peak in the diffusion-burning phase was found to be greater for HOME-producer gas compared to diesel-producer gas dual fuel operation. This may also be due to higher viscosity of HOME and reduction of air – fuel mixing rates along with slow-burning producer gas. This leads to less fuel being prepared for rapid combustion after the ignition delay. Therefore more burning occurs in the diffusion phase rather than in the premixed phase with HOME-producer gas operation. Significantly higher combustion rates during the later stages with HOME-producer gas operation leads to higher exhaust temperatures and lower thermal efficiency. However, improvement in heat release rate can be achieved for HOME-producer gas operation with RCC compared to the operation with HCC i.e., HOME-producer gas fueled dual fuel operation with RCC shows maximum peak heat release rate compared to HCC. It can be concluded, in the case of RCC, maximum amount of evaporated fuel is accumulated resulting in better mixing with air-producer gas combination due to better squish and swirl leading to lower ignition delay and hence comparatively shorter combustion duration with RCC. The combustion duration obtained for HOME-producer gas operation with a HCC and RCC were found to be 46.0 deg. CA and 43.4 deg. CA respectively compared to 39.4 deg. CA for diesel–producer gas operation with RCC (Fig. 30).

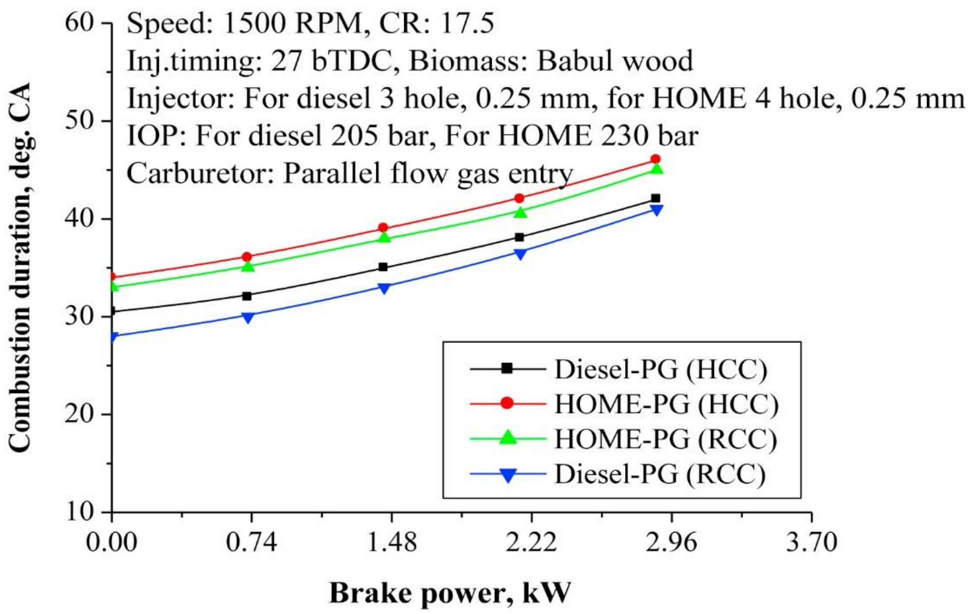


Fig. 30. Variation of combustion duration with brake power.

The variation of mean gas temperature (MGT) with crank angle can be seen in Fig. 31 at 80% load. Diesel-producer gas operation resulted in higher gas temperature than HOME-producer gas operation due to incomplete combustion caused by the burning of low calorific value of both HOME and producer gas. For the same operating condition and with RCC, maximum mean gas temperature of 1286 K is produced with diesel-producer gas (RCC) operation compared to 1115 K for HOME-producer gas (RCC) operation. Diesel-producer gas combination is an indicative of high flame propagation accompanied with better combustion which may be due to better combustion which in turn associates better fuel properties. The increase in mean gas temperature is due to the increase in peak cylinder temperature and peak pressure, which is due to instantaneous combustion that takes place in diesel-producer gas combustion. Highest temperature of 1266 K can be seen to have effected on the combustion of fuel combination used. However, it makes a significant spike in the heat release calculations. However, the effect of this on the total heat release of the dual fuel operation should be the least.

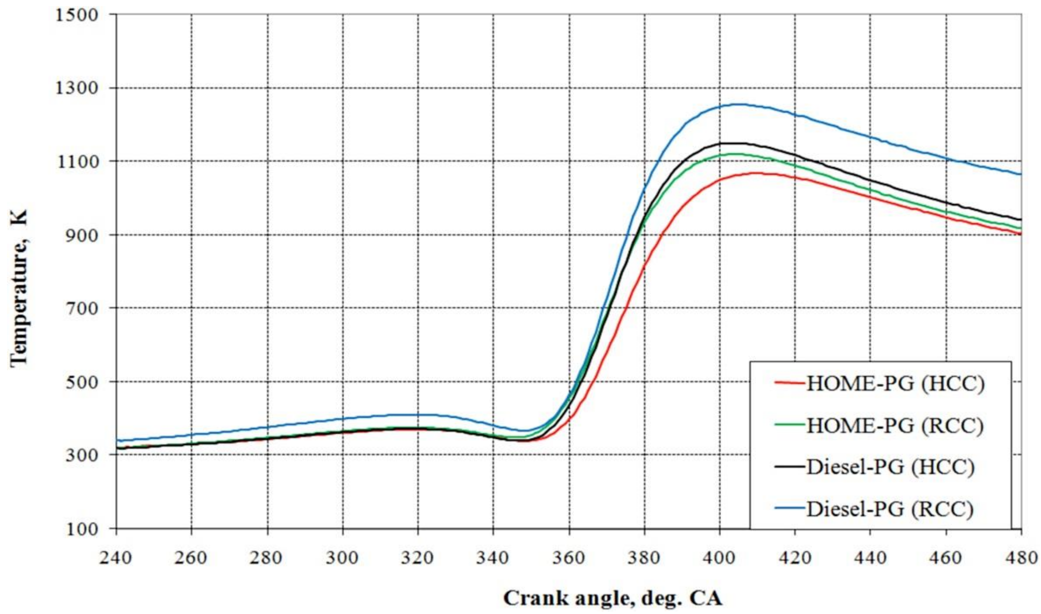


Fig. 31. Mean gas temperature versus crank angle with different combustion chamber configurations for diesel/HOME-Producer gas combinations at 80% load.

Fig. 32 shows cylinder pressure versus crank angle for the diesel/HOME–producer gas combinations with HCC and RCC configurations. The cylinder pressure–crank angle history is obtained for 100 cycles for diesel-producer gas and HOME-producer gas combination at 80% load, and the average pressure variation with crank angle is shown in Fig. 32. It is observed that the cylinder pressure is increased with an increase in brake power. The peak pressure depends on how much fuel was consumed and how combustion took place during the rapid combustion period. The uncontrolled combustion phase is governed by mixture preparation during the delay period. Therefore, mixture preparation and the slow-burning nature of the producer gas during the ignition delay period are responsible for peak pressure and the maximum rate of pressure rise. Results have shown that HOME-producer gas operation with RCC resulted in higher cylinder pressure compared to dual fuel operation with HCC. It could be attributed to the fact that during the dual fuel operation with RCC, the charge that would be accumulated in the bowl and swirl induces the high flame velocities by directing the flame propagation throughout the combustion chamber and the entire fuel-air got mixed properly due to better squish formed and hence the mixture got ignited and burnt simultaneously. This might have led to highest in-cylinder pressure during the dual fuel operation with RCC configuration compared to the operation with HCC. HOME-producer gas operation with RCC, in-cylinder flow turbulence was found to have affected the combustion and generated higher turbulence levels resulting air-producer gas mixing along with injected fuel, and leading to shorter combustion durations in the present engine. With use of RCC, both intensity of swirl and mixing of fuel combination increases, thereby the pre-flame combustion reaction speeds up quickly. The physical delay and pre-flame combustion reaction may be completed early and lead to rapid combustion in the bowl behaving like a sharp front propagating throughout the charge. This suggests that the charge in front of this reaction zone reaching quick auto ignition and heat from the reaction zone heats up the gas mixture in close proximity. Therefore the combustion rate is faster till completion of the unburnt mixture and thus consuming the charge completely. Based on the results obtained, second peak during the diffusion burning phase was reduced due to the use of RCC configuration. The sharp increase in combustion acceleration shows increase in cylinder pressure during the piston's descent and that the combustion energy is efficiently converted into work. The cylinder pressure obtained for HOME-producer gas operation with a HCC and RCC were found to be 54.8 bar and 58.5 bar respectively compared to 66 bar for diesel–producer gas operation with RCC.

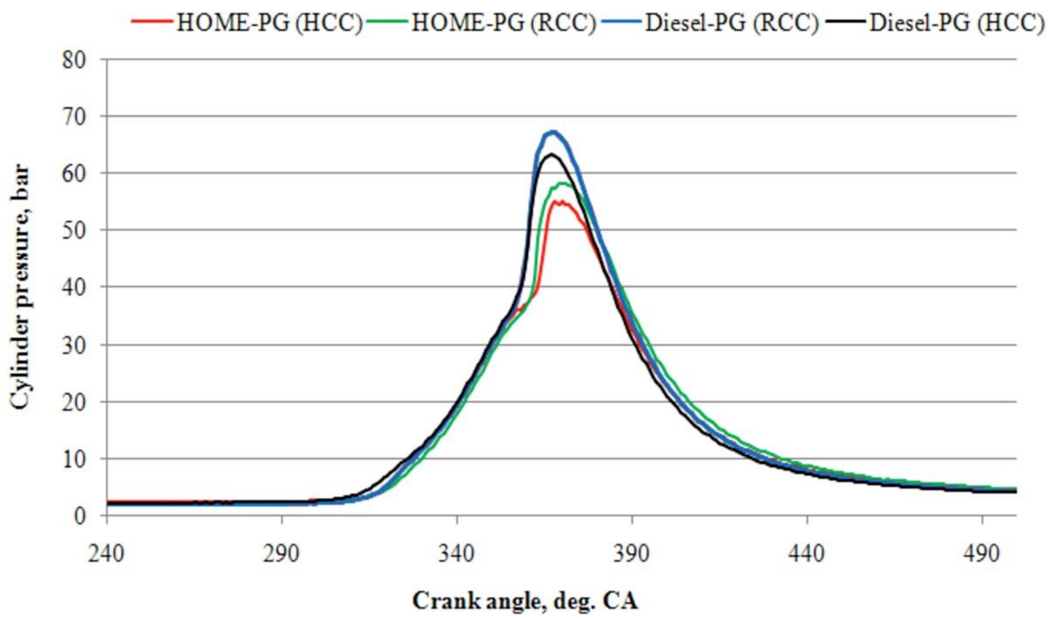


Fig. 32. In-cylinder pressure versus crank angle with different combustion chamber configurations for diesel/HOME-Producer gas combinations at 80% load.

Fig. 33 shows heat release rate versus crank angle for HOME – producer gas combinations at 80% load with different combustion chamber configurations. Heat release rate is dependent on the complex turbulent mixing of fuel and air at high temperature after compression. The variety of combustion chambers and types of fuel injection equipments influence the heat release rate characteristically. It can be seen from the Fig. 33 that, for the same fuel combination (HOME-producer gas operation), maximum heat release rate with HCC was found to be lower than operation with RCC. This was due to longer ignition delay and combustion duration for dual fuel operation with HCC compared with that of RCC operation. The main reason for this is due to the fact that the center of gravity of heat release rate diagram was shifted from TDC and reducing the available expansion ratio. Lower heat release rate resulted in lower pressure-rise rate, which benefits noise reduction. This is due to the result of higher second peak obtained with HCC operation in the diffusion combustion phase compared to RCC operation. It was also found that the HOME-producer gas fueled dual fuel operation with HCC has a higher unburned fuel-air mixture region, and thus would have lower heat release rate. However, for the same fuel combination with RCC, optimum operating point could be obtained and has better heat release rate. In addition, the dual fuel operation with HCC, a potential has been found to improve the NOx emission compared to the operation with RCC. In addition, the poor spray atomization characteristics of HOME due to higher viscosity and surface tension and slow burning nature of producer gas leads to improper mixing when dual fuel operation is with HCC. Further it was noticed that the heat release rate during the diffusion combustion phase of HOME-producer gas operation with HCC was slightly higher compared to the dual fuel operation with RCC. However, the heat release rate for RCC fueled with HOME-producer gas combination demonstrated similar trend, but slightly better than HCC.

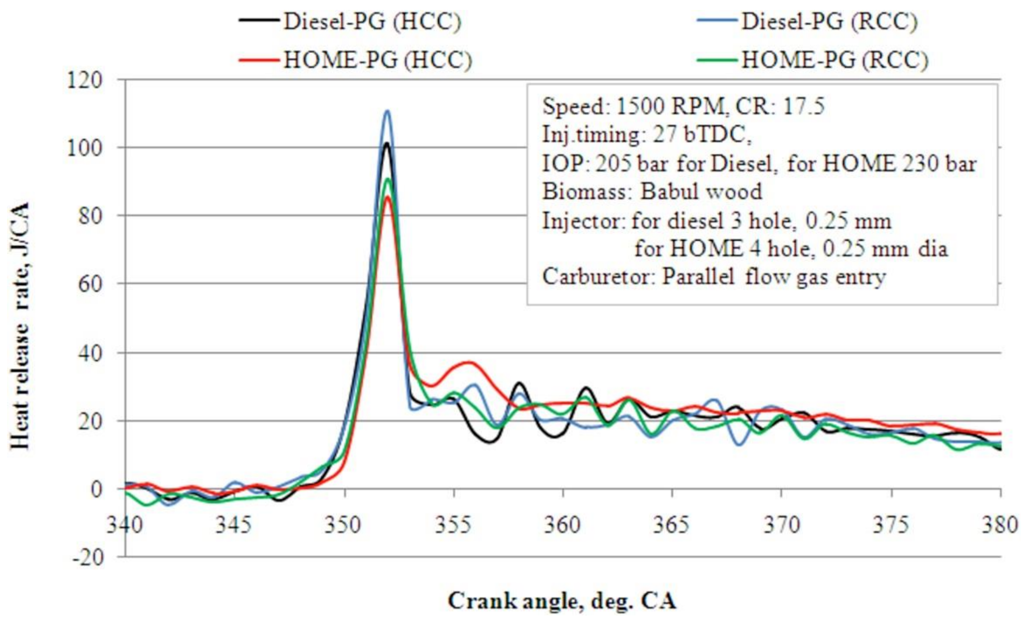


Fig. 33. Rate of heat release versus crank angle with different combustion chamber configurations for diesel/HOME-Producer gas combinations at 80% load.

From the results, it is observed that diesel-producer gas combination has higher combustion rate compared to HOME-producer gas combination. Moreover, HOME has lower calorific value and hence higher mass is required to achieve the energy release rate [50], [51]. Mass fraction burned represents the process of transforming chemical energy of fuel into heat and is a function of crank angle. As it is evident from the Fig. 31, it is observed that gas temperature distributions are different for different fuel combinations. Hence heat release rate variations are obvious, and differences in the heat release rate curves are bound to occur. Increase in the burning rate of fuel combination is mainly due to fast combustion of the injected fuel (diesel). However, at full load condition, the mass fraction burnt of HOME-producer gas combination is slightly improved with dual fuel operation using RCC compared to HCC. It is concluded that, fuel properties have major effect on the combustion.

7. Conclusions

The application to a HOME-producer gas dual fuel engine showed that combustion process is strongly influenced by the nozzle and combustion chamber type. However, the procedure is successful in defining the best combustion chamber configuration with perfectly matched nozzle geometry in order to achieve the goals of the conversion process and a reduction of emissions.

- From the experimental investigations, for high viscous fuel HOME – slow burning producer gas operation, it is observed that HOME-producer gas operation resulted in improved performance with a nozzle geometry of fuel injection pressure (230 bar), number of nozzle hole (4 hole) and size (0.25 mm). The increase in injection pressure, hole number with smaller hole size could lead to efficient mixture preparation resulting in lower emissions. Significant improvements in power output and the trade-off between smoke and NOx emissions can be obtained for dual fuel operation if mentioned nozzle geometry is used along with re-entrant combustion chamber configuration.
- With all optimized operating conditions, the engine was operated for 6 h long duration. On an average, for HOME–producer gas operation with optimized nozzle geometry and RCC resulted in 5.65% increased BTE, 19% decreased smoke, 11.2% decreased HC, 17.64% decreased CO, and 15.68% increased NOx levels, and lower ignition delay and combustion duration and slightly increased cylinder pressure and improved heat release rates were observed compared to HOME–producer gas operation with optimized nozzle geometry and HCC. It has also been observed that, HOME-producer gas combination resulted in smoother engine operation. Renewable and alternative fuels like biodiesel and producer gas derived from various bio-mass are suitable for dual fuel operation and their extensive use will pave way for the energy security of the country.

- Future developments concern about mixing of hydrogen with producer gas and optimization of producer gas fueled dual fuel engine by developing a turbocharger for diesel engine along with increased compression ratio in order to obtain better results in terms of performance and emissions.

On the whole it is concluded that HOME and producer gas induction could be used as an alternative and renewable fuels in diesel engines. Running the engine in dual fuel mode requires no major modifications in the existing diesel engine. HOME-producer gas operation with optimum parameters resulted in overall better engine performance with reduced emission levels.

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