Research Paper

EFFECT OF CARBURETOR TYPE AND INJECTION STRATEGIES ON THE PERFORMANCE OF ROME AND COCONUT SHELL DERIVED PRODUCER GAS FUELED DUAL FUEL ENGINE

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In the present work, effect of Carburetor type and injector nozzle geometry variation is studied on the performance of dual fuel engine operated on Rice bran oil methyl ester (ROME) and coconut shell derived producer gas fuel combinations. For this different carburetors providing stoichiometric gas-air mixture were designed and fabricated in conjunction with different injector nozzles provided with varied hole size as well as varied number of nozzle holes as well. The performance of the dual fuel engine was optimized by comparing the results with the base line data generated. ROME-producer gas fueled dual fuel operation with 45° gas entry carburetor, and a mechanical injection system with 4 hole injector each having 0.2mm hole diameter resulted in overall acceptable performance with reduced emission levels.

Keywords: Dual-fuel engine, Producer gas, Gasifier-engine system, Injector nozzle geometry, Carburetor, Combustion and Emissions

INTRODUCTION

Due to their high thermal efficiency and reliability diesel engines are widely used in transportation and power generation applications. Advances in stationary power generation applications using diesel engine technology are becoming more sustainable with use of renewable energy recourses. In the present energy scenario, demand for energy is on a continual increase. Environmental concerns and depletion of fossil fuel reserves have led to the extensive search for alternative fuels. Using biomass as a source of energy can not only reduce the dependency on imported oil, but may also benefit the environment by reducing emissions of greenhouse gases and pollutants that affect the air quality (Rajiv Varshney et al., 2010; Murugesan A et al., 2009; Banapurmath N R et al., 2005). Biofuels such as biodiesels and
renewable gaseous fuels derived from biomass present a very promising alternative to diesel oil since biofuels have numerous advantages compared to fossil fuels (Varshney et al., 2010; Murugesan et al., 2009; Banapurmath et al., 2005; Banapurmath et al., 2007; Parikh et al., 1987; Cruz I 1985; Shrinivasa and Mukunda 1984; Ernest 1980; Breag and Chittenden 1979; Mohod et al., 2003). Reduction of harmful gases and improvement of brake thermal efficiency of diesel engine have been in the progress. It is observed that type of fuel and engine operating conditions influence the formation of pollutants and their control can help to decrease the tailpipe emissions. Blends of alcohols with base fuel are used to decrease CO, NOx and smoke emission levels, but they can increase unregulated emission levels (Likos et al., 1982; Hansen et al., 2005; Park et al., 2008, Banapurmath et al., 2010).

**Dual Fuel Concept**

Limited resources of fossil fuels, stringent emission norms and rapid hike in the fossil fuel prices have triggered many alternative solutions. In view of this many investigators have studied dual-fuel engines using different fuel combinations. Renewable and alternative fuels have numerous advantages compared to fossil fuels as they are renewable, biodegradable, providing energy security and foreign exchange savings besides addressing environmental concerns, and socio-economic issues as well (Yaliwal et al., 2013). The dual-fuel concept uses both injected liquid and inducted gaseous fuel. The advantages of dual-fuel engine have attracted many investigators to use this for power generation applications (Sahoo et al., 2009).

Biomass can be converted into gaseous fuel called as producer gas with the help of a chemical conversion processes using down draft gasifier. Such biomass energy conversion technologies are gaining more prominence for providing energy for rural as well as industrial sectors. Use of biodiesel-producer gas fuel combinations can result in a significant reduction in the oil import bill and simultaneously reduce air pollution (Ramachandran, 1992; Singh et al., 2007; Ramadhhas et al., 2008; Banapurmath et al., 2009, 2012; Yaliwal et al., 2013).

Numerous experimental investigations concerning the dual fuel engines using combination of diesel/biodiesel-producer gas have been reported in the literature. Literature on gasification technology, pyrolysis parameters, potential scope and economic analysis of the gasifier system have been reported (Bridgwater, 2003; Ouadiet et al., 2002; Ravindranath and Balachandra, 2009; Rathore et al., 2009, 2012; Dasappa, et al., 2011).

Study on dual fuel engine operation in high-speed direct injection (DI) and low-speed indirect injection engines have been investigated (Parikh et al., 1989). It is reported that dual fuel engine fueled with diesel/biodiesel-producer gas combination always resulted into lower performance with increased HC and CO emission levels. (Parikh et al., 1989; Singh Ramadas, Jayaraj and Muraleedharan, 2006, 2008; Banapurmath and Tewari, 2009; Banapurmath, et al. 2009, 2011; Shrivastava et al., 2013; Yaliwal et al., 2013). Performance of dual-fuel engine for both rural and urban applications have been reported (Banapurmath, Tewari and Hosmath 2008; Dasappa et al., 2011; Banapurmath et al., 2009, 2011; Parikh et al., 1989; Ravindranath and Balachandra 2009; Sridhar et al., 2005).
To have stoichiometric mixture of air-producer gas, a suitable carburetor is required. In order to achieve proper dual fuel combustion in diesel engines, air-fuel mixing plays a crucial role and is one of the most important operating variables in a dual fuel engine. However, other parameters such as injection timing, injector opening pressure, compression ratio and water injection affect dual fuel engine performance with reduced emissions (Banapurmath et al., 2008, 2009; Sahoo et al., 2009; Roy et al., 2009). Effect of carburetor type on the air-fuel mixing and engine performance has been investigated (Anil et al., 2006; Banapurmath et al., 2011). Fuel substitution of about 60-90% has been reported in dual fuel engines providing conservation of injected liquid fuels (Parikh et al., 1989; Banapurmath and Tewari, 2009; Banapurmath et al., 2009, 2011; Ramadas, Jayaraj, and Muraleedharan, 2006).

Fuel injection systems in diesel engines and their injection characteristics affect fuel atomization and formation of proper fuel air mixture, and hence decide engine performance and pollutant formation from diesel engines (Tuan et al., 2007). Proper fuel evaporation, its penetration and dispersion is important to achieve improved performance with reduced emission levels. Injection parameters such as injection timing (IT), injector opening pressure (IOP), and injector nozzle geometry plays a significant role in dictating the dual fuel engine performance.

Advancing the IT, increasing the IOP improves the engine performance (Banapurmath et al., 2008; 2011). Varied injector opening pressure and rate profiles have been reported for different injected liquid fuels under conditions of different speed and load during engine operation. In general, use of higher injection pressure and desired injection rate provides improved mixing of fuel combinations and heat release rates resulting in reduced hydrocarbon (HC) and carbon monoxide (CO) and soot emission levels (Lee et al., 2010; Ramirez et al., 2012).

Effect of compression ratio, hydrogen addition and properties of various biomass feed stock used on the performance of dual fuel engine have been reported in the literature (Sridher et al., 2005; Roy et al., 2009; Banapurmath et al., 2011; Yaliwal et al., 2013).

Thus fuel injection is an important operating parameter which affects the fuel vaporization; distribution and mixing of fuel within the combustion chamber which in turn is responsible for improved overall performance of a diesel engine. Appropriate droplet size, fuel distribution, and penetration leads to more efficient combustion and lowered emissions (Ramirez et al., 2012). The break-up model of droplets has been discussed in detail in the literature (Reitz and Diwaker, 1986). The behavior of fuel once it is injected in the combustion chamber and its interaction with air is important. It is well known that nozzle geometry and cavitations strongly affect evaporation and atomization processes of fuel. Suitable changes in the in-cylinder flow field resulted in differing combustion. The performance and emission characteristics of compression ignition engines are largely governed by fuel atomization and spray processes which in turn are strongly influenced by the flow dynamics inside injector nozzle. Modern diesel engines use micro-orifices having various orifice designs and affect engine performance to a great extent. Effects of dynamic factors on injector flow spray combustion and emissions have been investigated by various researchers (Mulemane et al., 2004; Som et al., 2010). Experimental studies involving the effects of nozzle orifice...
geometry on global injection and spray behavior has been reported (Pyari et al., 2008; Benajes et al., 2004; Hans et al., 2002). (Cenk et al., 2013) studied the effect of number of nozzle holes on the performance of diesel engines. Increased nozzle holes reduced BSFC (Heywood 1988; Cenk et al., 2013). Smaller size injector nozzle hole resulted into reduced spray tip penetration due to the low spray momentum (Sharma et al., 2013).

In this present study, effect of mixing of air and producer gas, injector opening pressure, number of holes and nozzle size on the performance, combustion and emission characteristics of single cylinder four stroke diesel engine operating on dual fuel mode using rice bran oil methyl ester (ROME) and producer gas derived from coconut shell biomass has been investigated.

CHARACTERIZATION OF FUELS USED

Properties of fuels influence the performance, combustion and emission levels of dual fuel engines. Characterization of fuels forms the first step for efficient utilization of their energy potential. As mentioned earlier rice bran oil methyl ester [ROME] and producer gas derived from coconut shell biomass feed stock were used. Table 1 shows the fatty acid composition of rice bran oil used. Table 2 shows the properties of rice bran oil. Table 3 shows properties of coconut shell derived biomass used for the study.

<table>
<thead>
<tr>
<th>Fatty acid</th>
<th>Percentage</th>
<th>Type</th>
<th>C:N</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lauric</td>
<td>0.11%</td>
<td>Saturated</td>
<td>12.0</td>
</tr>
<tr>
<td>Myristic acid</td>
<td>0.24%</td>
<td>Saturated</td>
<td>14.0</td>
</tr>
<tr>
<td>Palmitic acid</td>
<td>12.46%</td>
<td>Saturated</td>
<td>16.0</td>
</tr>
<tr>
<td>Stearic acid</td>
<td>8.32%</td>
<td>Saturated</td>
<td>18.0</td>
</tr>
<tr>
<td>Oleic acid</td>
<td>27.78%</td>
<td>Unsaturated</td>
<td>18.1</td>
</tr>
<tr>
<td>Linoleic acid</td>
<td>37.65%</td>
<td>Unsaturated</td>
<td>18.2</td>
</tr>
<tr>
<td>α-Linolenic acid</td>
<td>13.44%</td>
<td>Unsaturated</td>
<td>18.3</td>
</tr>
</tbody>
</table>

Table 2: Properties of Rice Bran Oil

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>Properties</th>
<th>Diesel</th>
<th>Rice Bran oil</th>
<th>ROME</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Viscosity @ 40°C (cst)</td>
<td>2-5</td>
<td>928</td>
<td>4.12</td>
</tr>
<tr>
<td>2</td>
<td>Flash point °C</td>
<td>75</td>
<td>316</td>
<td>174</td>
</tr>
<tr>
<td>3</td>
<td>Calorific Value in kJ / kg</td>
<td>43,000</td>
<td>40106</td>
<td>41106</td>
</tr>
<tr>
<td>4</td>
<td>Cetane number</td>
<td>44-55</td>
<td>50.1</td>
<td>51.6</td>
</tr>
<tr>
<td>5</td>
<td>Density Kg / m³</td>
<td>840</td>
<td>880</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Type of oil</td>
<td>------</td>
<td>Non edible</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Colour</td>
<td>Light brown</td>
<td>Yellowish brown</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Cloud Point (°C)</td>
<td>6</td>
<td>13</td>
<td>9</td>
</tr>
<tr>
<td>9</td>
<td>Pour Point (°C)</td>
<td>-7</td>
<td>1</td>
<td>-2</td>
</tr>
<tr>
<td>10</td>
<td>Conardson carbon residue</td>
<td>0.1</td>
<td>0.6</td>
<td>0.35</td>
</tr>
</tbody>
</table>

Table 3: Properties of Coconut Shell Biomass Used

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>Description</th>
<th>Coco-Nut shell</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Moisture Content, % w/w</td>
<td>9.2</td>
</tr>
<tr>
<td>2</td>
<td>Ash Content, % w/w</td>
<td>2.1</td>
</tr>
<tr>
<td>3</td>
<td>Volatile Matter, % w/w</td>
<td>67.2</td>
</tr>
<tr>
<td>4</td>
<td>Fixed Carbon % w/w</td>
<td>19</td>
</tr>
<tr>
<td>5</td>
<td>Sulphur, % w/w</td>
<td>0</td>
</tr>
<tr>
<td>6</td>
<td>Nitrogen, as N % w/w</td>
<td>0.04</td>
</tr>
<tr>
<td>7</td>
<td>Gross Calorific value, kJ/ kg</td>
<td>20490</td>
</tr>
<tr>
<td>8</td>
<td>Density, kg/ m³</td>
<td>404</td>
</tr>
</tbody>
</table>

EXPERIMENTAL SETUP

Experimental investigations were conducted on a four-stroke single cylinder direct injection water cooled compression ignition engine coupled with down draft gasifier as shown in Figure 1. The specification of the engine and gasifier is given in Table 4 and 5. The engine was always operated at a rated speed of 1500 rev/min. The engine was having a conventional mechanical fuel injection system. The fuel consumption was measured with burette and stopwatch. The
injector opening pressure and the static injection timing as specified by the engine manufacturer were 205 bar and $23^\circ$ before top dead centre (BTDC), respectively. In the present work, the injection timing was kept constant ($27^\circ$ bTDC) and the injection pressure was varied from 220 – 240 bar. However, for Diesel-Producer gas operation injection pressure was kept at 205 bar. Three different injectors having 3, 4 and 5-holes each having an orifice size of 0.2, 0.25 and 0.3 mm in diameter was used for the present research work. The engine had been provided with a hemispherical combustion chamber with overhead valves operated through push rods. A piezoelectric pressure transducer was mounted flush with the cylinder head surface to measure the cylinder pressure. The cylinder pressure was measured with piezo electric transducer fitted in the cylinder head. Five gas analyzer and Hart ridge smoke meter was used to measure HC, CO, CO$_2$, NO$_x$ and smoke opacity. All the measurements were done when the engine attained a steady state. For each load, five readings were generated to ensure the accuracy of the data recorded and careful experimental arrangements were made to obtain consistent and repeatable measurements.

Table 4: Specification of Experimental Test Rig

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>Parameters</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Machine Supplier</td>
<td>Apex Innovations Pvt Ltd, Sangli, Maharashtra State.</td>
</tr>
<tr>
<td>2</td>
<td>Engine Type</td>
<td>Single cylinder four stroke water cooled direct injection TV1 compression ignition engine with a displacement volume of 662 cc, compression ratio of 17:1, developing 5.2 kW@1500 rpm TV1 (Kirolsker make)</td>
</tr>
<tr>
<td>3</td>
<td>Software used</td>
<td>Engine Soft</td>
</tr>
<tr>
<td>4</td>
<td>Nozzle opening pressure</td>
<td>200 - 225 bar</td>
</tr>
<tr>
<td>5</td>
<td>Governor type</td>
<td>Mechanical centrifugal type</td>
</tr>
<tr>
<td>6</td>
<td>Cylinder diameter (Bore)</td>
<td>0.0875 mtr</td>
</tr>
<tr>
<td>7</td>
<td>Stroke length</td>
<td>0.11 mtr</td>
</tr>
<tr>
<td>8</td>
<td>Combustion camber</td>
<td>Open Chamber (Direct Injection) with hemispherical cavity</td>
</tr>
<tr>
<td>9</td>
<td>Eddy current dynamometer:</td>
<td>Model : AG - 10, 7.5 KW at 1500 to 3000 RPM and Water flows through dynamometer during the use</td>
</tr>
</tbody>
</table>

Table 5: Specification of the Downdraft Gasifier

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

Producer Gas Carburetors

Carburetors available for gaseous fuels such as natural gas, biogas and landfill gas is unsuitable in the present situation due to widely different stoichiometric air to fuel requirements. For the present work, carburetors were developed based on area ratio (Dasappa 2013; Sridher et al., 2005,
Anil et al., 2006). The carburetor designed for producer gas must have an ability to maintain the required air-fuel ratio (1.2 to 1.5) with varying load conditions and must give smooth operation with minimal pressure loss.

Three carburetors with 30, 45 and 60° gas entries were developed and fabricated in house to enable effective mixing of air and producer gas. Preprocessing has been done in GAMBIT and solver FLUENT has been used for analysis. In this present work, the air-fuel ($\lambda$) equivalence ratio was calculated by using the relation (Papagiannakis and Hountlas, 2011). Results with CFD analysis showed that 45° shaped carburetor gives better air-fuel mixing and subsequently is checked experimentally.

$$
\lambda = \frac{m(a)}{m(F_{\text{injected fuel}}) + (m\frac{A}{F})_{\text{inducted fuel}}}
$$

Where, $m(a)$ corresponds to mass flow rate of inducted air and $(m\frac{A}{F})_{\text{injected fuel}}$ and $(m\frac{A}{F})_{\text{inducted fuel}}$ represent the product of mass flow rate and stochiometric air-fuel ratio for injected and inducted fuel respectively.

RESULTS AND DISCUSSION

Problem Definition and Mesh Generation

A specially developed producer gas carburetor was analyzed for its mixing performance. Simulations are carried out on the producer gas carburetor model as shown in the respective sub sections. Figure 2, 3, and 3 shows air-producer gas mixing with 30, 45 and 60 deg., gas entry carburetor.

The 30° Gas Entry Carburetor

The carburetor has two inlets one for producer gas of 31.75 mm and other for air of 25.4 mm and outlet has 25.4 mm diameter. In this case producer gas entry is provided with a flow at 30 degree and three-dimensional structured mesh was created and it consisted of 154643 nodes.
The 45° Gas Entry Carburetor
The Y-shape carburetor has two inlets one for producer gas and other for air having diameter of 2 inch each. The CFD analysis was carried out on 45° shaped carburetor. Three-dimensional model has been used to simulate the flow of air and producer gas mixture. The hex mesh was created in ICEM CFD with mesh size of approximately 135000 nodes. The grid is a structured mesh with a first layer mesh size of 0.1mm with expansion factor of 1.2.

The 60° Gas Entry Carburetor
The carburetor has two inlets one for producer gas of 31.75 mm and other for air of 25.4 mm and outlet has 25.4 mm diameter respectively. In this case producer gas entry is provided with a flow at 60° and three-dimensional model structured mesh is created and it has been used to simulate the flow of air and producer gas mixture in a carburetor with mesh grid density of around 156431 computational nodes is considered.

Boundary Conditions
The flow domain considered for simulation is the whole carburetor assembly with steady state flow. The inlet boundary conditions applied for air and producer gas include mass flow rate and pressure with no buoyancy steady state condition. The initial condition of flow rate through the air inlet with ideal mass fraction is taken as 0 while mass fraction of producer gas is 1. Mass flow rate of producer gas is 0.0064763676 kg/s, and mass flow rate of air is taken as 0.0064205311 kg/s at their respective inlets. The results obtained for different carburetor shapes are given in the subsequent paragraph. Preprocessing was carried out in ICEM CFD and solver FLUENT (ANSYS) has been used for analysis. Turbulent model based on k-ε theory with a RANS code has been used for the analysis of various carburetor shapes. Solver uses Navier stoke, continuity, momentum and energy equations and other related equations are not addressed in the present work. Producer gas mass fraction across a selected plane, velocity streamlines and velocity vectors were obtained and the same is explained in the subsequent paragraphs.

Optimization of Carburetor for Dual Fuel Engine
This section presents the results of investigation carried out on a single cylinder, DI engine operating on ROME and producer gas in dual fuel mode of operation with different types of carburetors namely 30°, 45° and 60° gas entry carburetors. The speed of the engine was maintained constant at 1500 rpm.

Performance Characteristics
The brake thermal efficiency (Figure 5) is found to be higher for producer gas-diesel dual fuel mode of operation compared to ROME-Producer gas operation over the entire load range. Producer gas being common, properties of the injected fuel has a major effect on the engine performance. The bio-diesel injected fuel has higher viscosity than diesel which makes atomization difficult and also has lower calorific value, resulting in lower brake thermal efficiency. Of all the carburetors, 45° gas entry carburetor gives better performance compared with 30° and 60° degree gas entry carburetors. The brake thermal efficiency values at 80 % load with ROME - Producer gas operation for 30°, 45° and 60° flow gas entry carburetors were found to be 13.56, 17.2, 15.4% and 18.68% for diesel-Producer gas operation respectively.
Exhaust gas temperature at different power outputs in dual fuel mode of operation is presented in Figure 6. The Exhaust gas temperature was found to be higher for ROME -Producer gas compared to Diesel-Producer gas operation. This could be attributed to incomplete combustion of gaseous fuel and injected biodiesel burns during diffusion combustion phase. Results showed that 45° carburetor gives lower exhaust gas temperature compared with other carburetors tested. The Exhaust gas temperatures at 80 % load with ROME - Producer gas operation for 30°, 45° and 60° carburetors were found to be 540, 525, 510 and 495°C and 410°C for Diesel-Producer gas operation respectively.

Emission Characteristics

The different emission parameter measurements during the dual fuel mode operation are discussed below.

The effect of brake power on smoke opacity is shown in Figure 7. The smoke opacity was found to be lower for diesel-Producer gas dual fuel operations compared to ROME - Producer gas over the entire load range. The presence of free fatty acid and heavier molecular structure of the injected bio-diesel fuel compared to diesel results in higher smoke levels. Of all the carburetors, 45° gives better performance compared with 30° and 60° gas entry carburetors with reduced smoke opacity. Better air and gas mixing as reported in CFD analysis results in to complete combustion of fuel combinations used. The smoke opacity values at 80 % load for ROME - Producer gas operation with 30°, 45° and 60° carburetors were found to be 69, 63, 58 and 53 HSU and 32 HSU for Diesel-Producer gas operation respectively.
The variation of HC emission levels with brake power is shown in Figure 8. The presence of free fatty acid and heavier molecular structure of the injected bio-diesel fuel compared to diesel results in higher HC emissions due to incomplete combustion. 45° carburetor ensures supply of stoichiometric mixture of air and producer gas compared to other carburetors used and this ensures better combustion as well.

At lower loads, HC emission were found to be higher and decreased at higher loads. The HC emission values at 80% load for ROME-Producer gas operation with 30°, 45° and 60° carburetors were found to be 71, 64, 58 and 52 ppm and 41ppm for diesel-Producer gas operation respectively.

The effect of brake power on CO emissions for diesel-Producer gas and ROME-Producer gas dual fuel operation is shown in Figure 9. Higher concentration of CO in the exhaust is a clear indication of incomplete combustion of the pre-mixed mixture. The CO levels were found to be higher for dual-fuel operations due to already presence of CO in the producer gas and the associated combustion inefficiencies. Further Producer gas-air mixture supplied to the engine reduces the amount of oxygen resulting in incomplete combustion and increases the CO. At lower loads, the mixture being leaner results in greater extent of incomplete combustion and hence higher CO concentration. This puts a lower load limit for the dual fuel operation. At higher loads, the CO levels in the exhaust may slightly reduce because of increased combustion temperatures prevailing in the combustion chamber. Higher emission of CO levels in the exhaust could be attributed to lower heating value of Producer gas, lower adiabatic flame temperature and lower mean effective pressures. However, the CO emission levels were found to be lower with 45° carburetor compared to other carburetors tested. This could be due to better mixing of gas and air as is evident from the CFD analysis and leads to slightly better combustion. The CO emission at 80% load with ROME-Producer gas operation for 30°, 45° and 60° carburetors were found to be 0.84, 0.712, 0.58 and 0.46% and 0.31% for diesel-Producer gas operation respectively.
The effect of brake power on nitrogen oxide emissions for both Diesel-Producer gas and ROME-Producer gas dual fuel mode of operation is shown in Figure 10. The NOx emissions were found to be higher for Diesel-Producer gas dual fuel operations compared to ROME-Producer gas over the entire load range. This is because the higher heat released during premixed combustion with Diesel-Producer gas dual fuel operation, results in higher BTE associated with higher NOx. Results indicate that 45° carburetor gives higher NOx compared with other carburetors because of higher heat release rates observed during premixed combustion. The NOx emission values at 80% load with ROME-Producer gas operation for 30°, 45° and 60° carburetors were found to be 38, 40, 43 and 48 and 56 ppm for diesel-producer gas operation respectively.

**Figure 10: Variation of NOx With Brake Power**

Optimization of Injector Opening Pressure

Performance Characteristics

The variation of brake thermal efficiency with respect to brake power for ROME-producer gas combination over entire load range. The injected liquid fuel properties are responsible for this observed trend. For ROME-producer gas operation, thermal efficiency increased up to an IOP of 230 bar compared and dropped with 240 bar. This may be due to good atomization taking place at higher injection pressure which helps in reducing the combustion duration associated with faster rate of heat release. Auto-ignition of liquid fuel was improved with higher IOP (230 bar) with improved liquid fuel breakup and evaporation. Higher injection pressures produce fully developed sprays within a short time, and thereby help in improving the vaporization process as the surface area of the liquid core increases. Lower thermal efficiency was recorded for ROME-producer gas operation at lower IOP. At lower IOP (220 bar), larger fuel droplets occur and thereby increase ignition delay period.

**Figure 11: Variation of BTE With Brake Power**

Variation in exhaust gas temperature (EGT) with respect to brake power is shown in Figure 12. ROME-producer gas operation resulted in higher EGT compared to diesel-producer gas operation. It could be attributed to lower volatility, higher viscosity of the injected ROME along with
slow burning producer gas fuel combinations. Higher IOP (230 bar) resulted in reduced EGT but increasing IOP beyond 230 bar resulted into increased EGT as is evident from reduced thermal efficiency at this pressure. Formation of larger droplets at lower IOP and their incomplete burning results into increased EGT. 

**Figure 12: Variation of Exhaust Gas Pressure With Brake Power**

![Graph showing variations of exhaust gas temperature with brake power for different fuel operations.](image)

**Figure 13: Variation of Smoke Opacity With Brake Power**

![Graph showing variations of smoke opacity with brake power for different fuel operations.](image)

**Emission Characteristics**

Figure 13 shows variations in smoke levels for dual fuel engine operation for varied injection pressures. ROME-producer gas operation at higher injection pressure (230 bar) resulted in lower smoke levels compared to 220 bar. Increased injection pressure, reduces the droplet size leading to its better mixing with air and producer gas mixture which further results in improved combustion. In such a situation flame produced by burning of fuels reaches the entire area of cylinder. At higher pressure soot particles may be very nearer to the fuel spray nozzle tip. Therefore it is speculated that formation of soot nearer to the nozzle is advantageous since interactions with the cylinder wall would be lowered. Hence lower smoke emission levels were obtained at higher injection pressure for dual fuel operation. But, if the injection pressure is too high (240 bar) ignition delay become shorter and hence the combustion efficiency lowers. Therefore, comparatively higher smoke levels were formed at 240 bar. At lower injection pressure of 220 bar, higher fuel droplets were formed and mixing with air will be non-homogeneous leading to partial burning of the fuel droplet along with producer gas resulting in higher smoke levels compared to 230 bar.

The variations of hydrocarbon (HC) and carbon monoxide (CO) emissions for diesel - producer gas and ROME - producer gas operation with respect to various injection pressures are shown in Figure 14 and 15. Both HC and CO are the products of incomplete combustion. It is observed that, increase in injection pressure decreases both HC and CO emission levels. Increasing the injection pressure increases the combustion temperature and cylinder peak pressure and provides proper mixing of the fuel combinations used leading to complete combustion and hence resulting in lower HC and CO emission levels. Increased
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 injection pressure of 230 bar compared to 220 and 240 bar. At lower injection pressure (220 bar), the combustion temperature is lower and the resulting improper mixing of fuel with air further leads to freezing of the oxidation process. Decreased hydraulic flow rate of liquid fuel results into larger droplets resulting in reduced mixing rates. 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 into incomplete combustion.

The variation of NOx emission with brake power is shown in Figure 16. The NOx emission levels with ROME-producer gas dual fuel operation are lower compared to Diesel-Producer gas operation. This is because the combustion temperature inside the engine cylinder is lower with ROME-Producer gas combination as ROME has lower calorific value, lower volatility. For ROME-Producer gas operation, an injection pressure of 230 bar results into increased NOx emission levels compared to lower injection pressure of 220. For dual operation at optimum injection pressure (230 bar) operation NOx emissions increased. This could be due to proper mixing of liquid fuel droplets with mixture of producer gas and air leading to faster combustion rates and higher temperatures in the cycle. Better atomization and improved mixing rate reduce delay period and combustion duration which may be responsible for increased cylinder pressure and heat release rates.
Optimization of Number of Injector Holes

Performance Characteristics

Figure 17 shows the variation of brake thermal efficiency (BTE) with brake power for ROME-producer gas operation with different number holes each having same orifice size of 0.3 mm used in injectors. It is observed that over the entire power range, ROME-producer gas combination operates at a lower BTE compared to diesel-producer gas mode of operation. The lower calorific value and higher viscosity of ROME in presence of slow-burning producer gas results in to lowered BTE. The results obtained are in close agreement with published literatures (Parikh et al., 1989; Sahoo et al., 2009; Banapurmath et al., 2008, 2009, 2011). Dual fuel operation with 4 hole injector resulted in higher thermal efficiency compared to 3 and 5 hole injectors. Improved spray dispersion with proper fuel penetration occurs for the 4 hole nozzle compared to 3 and 5 hole nozzles. Further enhanced liquid breakup, leading to smaller fuel droplets with higher dispersion occurs in 4 hole nozzle. However, with 5 hole injector, decreased BTE was observed because of higher mass flow rate of liquid fuel inside the combustion chamber. For 3 hole nozzle lower fuel injection rate and decreased or insufficient fuel injection may leads to lower fuel in the air-fuel mixture. The BTE for diesel-producer gas and ROME-producer gas operation with 3, 4 and 5 hole injectors were found to be 18.65, 13.82, 15.85 and 17.14% respectively for 80% load.

Figure 18 shows the variation of exhaust gas temperature (EGT) for diesel-producer gas and ROME-producer gas operation with different types of injectors. The EGT with diesel-producer gas combination under dual fuel mode was found to be lower compared to ROME-producer gas combination. This could be attributed to more burning of diesel-producer gas combination in premixed combustion phase leading to improved combustion compared to ROME-producer gas operation. (Parikh et al., 1989; Ramadas et al., 2008; Banapurmath et al., 2008; Sahoo et al., 2009). Dual fuel operation with 4 hole injector resulted into lower EGT compared to 3 and 5 hole injector. However, lower EGT with 4 hole injector could be attributed to higher brake thermal efficiency obtained from better mixing of the fuel air combinations. The EGT for diesel-producer gas and ROME-producer gas operation with 3, 4 and 5 hole injectors were found to be 385, 545, 480 and 440° respectively at 80% load.
Emission Characteristics

Combustion quality of the engine tested can be assessed with emission levels from the engine. The emissions from the dual fuel engine is mainly due to the variation of producer gas composition, type of liquid fuel used, engine design and operating parameters. The different emission parameters during the dual fuel mode of operation with different fuel combinations are discussed below.

Figure 19 shows the variation of smoke opacity for diesel-producer gas and ROME-producer gas operation with different types of injectors. The smoke opacity with ROME-producer gas combination under dual fuel mode was found to be higher compared to diesel-producer gas combination. This could be attributed to lower calorific value, improper spray pattern due to higher viscosity of ROME in presence of slow-burning producer gas compared to diesel results in higher smoke levels. The results reported were similar to the published literatures (Parikh et al., 1989; Banapurmath et al., 2008, 2009, Sahoo et al., 2009, Ramadas et al., 2008). 4-hole injector improves fuel air mixing and hence results in reduced smoke emissions. The smoke levels for diesel-producer gas and ROME-producer gas operation with 3, 4 and 5 hole injectors were found to be 32, 61, 54 and 46 HSU respectively for 80% load.

Figure 20 and 21 shows the variation of hydrocarbon (HC) and carbon monoxide (CO) emission levels for diesel-producer gas and ROME-producer gas operation with different number of holes used in injector each having 0.3 mm orifice. Both HC and CO emission levels are higher for ROME-producer gas operation compared to diesel-producer gas operation. It could be due to incomplete combustion of the ROME-producer gas operation. The incomplete combustion resulted in case of dual fuel mode of operation is due to presence of free fatty acids in ROME which has higher viscosity and lower cetane number. Also, lower calorific value of ROME and producer gas, lower adiabatic flame temperature and lower mean effective pressure are responsible for higher HC and CO emission levels (Parikh et al., 1989; Singh et al., 2007; Ramadas et al., 2008; Banapurmath et al., 2008; Sahoo et al., 2009).

Producer gas being common, the ROME-producer gas dual fuel operation with 4 hole injector resulted in lower HC and CO emission levels compared to 3 and 5 hole injector. 4 hole injector provides better mixing of the fuel combinations used with improved spray pattern and penetration. However, 5 hole injector resulted in higher HC and CO emission levels and it may be due to under-mixing of fuel injected and resulting in fuel-air ratios that are too rich for complete combustion.

The HC for diesel-producer gas and ROME-producer gas operation with 3, 4 and 5 hole injectors were found to be 38, 58, 41 and 49 ppm respectively at 80% load. The CO for diesel-producer gas operation with different types of injectors was found to be 200, 250, 230 and 210 ppm respectively.
gas and ROME-producer gas operation with 3, 4 and 5 hole injectors were found to be 0.31, 0.72, 0.42 and 0.54% respectively at 80% load.

Combined effect of incomplete combustion due to higher viscosity and lower energy density of ROME along with slow burning nature of producer gas could be responsible for this observed trend. The ignition delay observed with 4-hole injector was shorter than that of 3 and 5-hole injection, which could be correlated with NOx formation behavior. This is also responsible for lower NOx. However, it is observed that the NOx levels were found to be lower for 3 and 5 hole injector operation compared to 4 hole. It could be attributed to lower heat release rate due to incomplete combustion caused by improper spray pattern. The NOx levels for 3, 4 and 5 hole injector operation were found to be 56, 85 and 82 ppm respectively for ROME-Producer gas at 80% load while it was 110 ppm for diesel-Producer gas operation. The increased number of injector holes with reduced or smaller hole size may lead to efficient mixture preparation due to improved air utilization, which results in lower PM, HC and CO emissions. However, NOx emission increased due to the rise of the combustion temperature.

The NOx emission levels were found to be higher for diesel-producer gas dual fuel operation compared to ROME-producer gas operation with different number of holes in an injector over the entire load range (Figure 22). This is because of higher heat release rate during premixed combustion phase occuring with diesel-producer combination compared to ROME-producer gas combinations (Singh et al., 2007; Banapurmath et al., 2008; Ramadas et al., 2008; Sahoo et al., 2009).
Optimization of Injector Hole Size

Experiments were performed using 4 hole injector with three different nozzle hole sizes (0.2, 0.25 and 0.3 mm) and with conventional injector having 3 hole of 0.3 mm size that serve as baseline reading for comparison. The experiments were aimed to arrive at an optimum nozzle hole size that would give better engine performance.

Performance Characteristics

ROME-producer gas operation, lower BTE were found for 0.25 and 0.3 mm hole size compared to 0.2 mm hole size (Fig.23). This could be due to the fact that with decrease in nozzle hole size, fuel droplet size decreases but improves the mixing of injected fuel with the mixture of producer gas and air. Higher size of nozzle would have developed even larger droplets but with their increased penetration and could have not burnt completely leading to lower power, with increased fuel consumption. Lower calorific value and slow burning nature of producer gas further adds to this trend. Thus, at the prevailing conditions, an increased hole size of nozzle yielded lower BTE. Therefore four hole with 0.2mm diameter resulted in higher value of BTE. Spray breakup and decreased fuel droplet size with lower nozzle hole orifice diameter is responsible for this trend. Also, better vaporization, increased fuel-air mixing and pre-reactions were responsible for this trend with lower fuel nozzle hole orifice diameter. The optimal nozzle design would be one that provides the maximum number of liquid fuel burn in combustion process and minimum number of liquid fuel unburned (Heywood 1988). Besides, smaller hole diameter nozzle rises fuel pressure and it helps to break-up the liquid fuel and coalesce process. It is also affects the spray pattern, the distribution and size of fuel spray particle and further affects the ignition delay and the whole combustion process.

Figure 23: Variation of Brake Thermal Efficiency With Brake Power

Figure 24 shows the effect of nozzle hole diameter on exhaust gas temperature (EGT). Lower EGT with smaller nozzle hole (0.2 mm) compared to 0.25 and 0.3 mm diameter nozzle was observed. Decreased nozzle hole diameter enhances the mixture formation as well as combustion. It could be attributed to decreased hydraulic flow rate by means of reducing spray hole orifice diameter, thereby affecting mixture formation primarily through improved gas entrainment processes. Due to decreased hydraulic flow rate mixture formation improves and combustion as well. It was found that a lower hydraulic flow rate causes a higher mean relative air/fuel ratio within the fuel spray and thus induces less soot formation.

Figure 24: Variation of Exhaust Gas Temperature With Brake Power
Figure 25 shows variation of volumetric efficiency for ROME - producer operation with different hole size nozzles. The volumetric efficiency indicates breathing ability of an engine. A drop in volumetric efficiency with power output for all the fuel combinations was observed. This could be due to higher gas temperature and increased surface temperature of inlet valve and combustion chamber walls. This heating decreases the density of induced air, and hence, the drop in volumetric efficiency occurs. However, the part of the air replaced by producer gas further adds to this decreased volumetric efficiency trend. Diesel - producer gas operation resulted in higher volumetric efficiency compared to ROME - producer gas combinations. This could be due to improper utilization of air as more ROME is injected for the same power generation with producer gas being common in dual fuel combinations used. Also higher second peak was observed in the diffusion combustion phase for ROME - Producer gas operation for higher nozzle hole size. Whereas, improved volumetric efficiency was observed with the operation of 0.2 mm nozzle hole size. This could be attributed to better utilization of air. Values of the volumetric efficiency were 74, 73 and 72.5 % for ROME - producer gas operation at CR of 15, 16 and 17.5 respectively, compared to 76 % for diesel - producer gas operation at 80% load.

Emission Characteristics

Figure 26 shows the effect of nozzle hole diameter on smoke density under all load conditions. Reduced smoke levels for a nozzle having 0.2 mm hole diameter compared to other nozzles hole sizes were observed. It could be due to decreased hydraulic flow rate, obtained with reduced spray hole orifice diameter which further affects mixture formation primarily through improved gas entrainment processes. Decreased hydraulic flow rate benefits the mixture formation and combustion, which cause less soot emissions. It may also be due to proper mixing of fuel droplets with air and improved oxidation. Lower hydraulic flow rate causing higher mean relative air/fuel ratio within the fuel spray is also responsible for this trend.

Figure 27 represents the variation of HC emission against brake power of the engine when it was operated with different nozzle hole sizes in the nozzles. It can be observed that the unburned HC levels were found to be lower in case of ROME-producer gas operation with smaller nozzle hole diameter (0.2mm) compared to 0.25 and 0.3 mm. The interesting feature of the trends with reduced nozzle hole diameter is that the HC levels are lower with ROME-producer gas operation. It could be attributed to reduction in droplet size due to better spray pattern leading
to proper mixing of the fuel combinations. Smaller hole diameter nozzle with ROME could have undergone nearly complete combustion resulting in lower HC levels in the exhaust. Smaller diameter hole in a nozzle also lowers hydraulic flow rate and increases flow velocity. The occurrence resulted in spray break up and it can keep the nozzle holes free from deposits. Improvement in hydrodynamic flow rate inside the holes of an injector nozzle resulting changes in flow velocity and droplet size and will influence the fuel injection process positively.

Variations of CO emission with brake power are shown in Figure 28. Using smaller hole diameter (0.2 mm) nozzle lower CO emissions were observed for ROME-producer gas operation as shown in Figure 14. Relatively better spray pattern with smaller droplets promoting fuel spray atomization, air entrainment into the spray, and fuel-air mixing within the spray assisted the combustion. This altogether leads to better oxidation, hence lower CO emissions with smaller hole diameter (0.2mm) nozzle compared to other nozzles of 0.25 and 0.3 mm nozzle operation. This may be attributed to better fuel atomization and improved mixing rate of fuel with air, associated with reduced delay period, and combustion duration. This improved heat release rate, peak combustion pressure and temperature. Thus, the lower nozzle hole (0.2mm) injector provided better air and fuel mixing and hence higher premixed combustion occurring leading to higher NOx emissions.

Effect of nozzle hole geometry on NOx emission is shown in Figure 29. NOx emission is result of oxidation of nitrogen at high combustion temperature. ROME-producer gas operation with a 4 hole nozzle having decreased nozzle hole diameter, increased NOx emission due to faster and more complete combustion and increased temperatures in the cycle compared to 0.25 and 0.3 mm nozzle operation. This may be attributed to better fuel atomization and improved mixing rate of fuel with air, associated with reduced delay period, and combustion duration. This improved heat release rate, peak combustion pressure and temperature. Thus, the lower nozzle hole (0.2mm) injector provided better air and fuel mixing and hence higher premixed combustion occurring leading to higher NOx emissions.

Fuel Substitution

Figure 30 shows the effect of nozzle hole diameter on the fuel substitution for ROME-producer gas operation. ROME-producer gas dual fuel operation with 0.2 mm nozzle hole
diameter showed better utilization of gaseous fuel with more liquid fuel substitution by gas compared to the operation with 0.25 and 0.3 mm diameter nozzle. This could be due to reduced hydraulic flow rates as is evident with increased BTE.

Combustion Characteristics

The combustion in a diesel engine differs when gaseous fuels are used and it depends on the air-fuel mixture quality. Different combustion characteristics are discussed below.

The effect of brake power on ignition delay is shown in Figure 31. The ignition delay is calculated based on the static injection timing. Dual fuel operation with ROME - producer gas operation with various nozzle hole size showed varied ignition delay. This could be due to variations in the droplet size of injected fuel. Amount of producer gas taking part in the rapid combustion phase, variations in the air - producer gas mixture and changing of combustion temperature with use of different nozzle hole size is responsible for this trend. With reduced nozzle hole diameter (0.2 mm), lower ignition delay was observed compared to the operation with 0.25 and 0.3 mm nozzle. Reduced droplet size due to improved spray pattern and better air-fuel mixture lowers the ignition delay. Moreover, it is obvious that combustion depends on physical phenomena that include air-fuel mixing and vaporization, and also on chemical phenomena related to the reactions involved in the process (Payri et al., 2010).

![Figure 30: Variation of Fuel Substitution With Brake Power](image)

The combustion duration shown in Figure 32 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 with all dual fuel combinations. This is due to increase in the quantity of fuel injected. For the dual fuel combinations with 0.2 mm nozzle hole operation, reduced combustion duration was observed. This could be due to proper mixing of air-fuel with improved spray pattern. The second peak of diffusion-burning phase was greater for ROME-producer gas compared to diesel-producer gas dual fuel operation. This may also due to higher viscosity of ROME and reduction of air - fuel mixing rates along with slow-burning producer gas. This leads to less fuel being prepared for rapid combustion with ROME-producer gas operation after the ignition delay. Therefore more burning occurs in the diffusion phase rather than in the premixed phase with ROME-producer gas operation. Significantly higher combustion rates during the later stages with ROME-producer gas operation leads to higher exhaust temperatures and lower thermal efficiency. However, ROME -
producer gas with smaller nozzle hole diameter (0.2 mm) operation shows improvement in heat release rate compared to ROME-producer gas operation with 0.25 and 0.3 mm nozzle hole diameter.

Higher second peak during the diffusion burning phase was observed for ROME-producer gas with 0.25 and 0.3 mm nozzle hole diameter compared to 0.2 mm nozzle hole diameter. This could be attributed to incomplete combustion arising from improper spray pattern with larger nozzle size caused by higher viscosity of ROME leading to larger droplets of liquid fuel. Also, poor quality of producer gas and reduction of air entrainment and fuel air mixing rates along with slow burning nature of producer gas are also responsible for this trend.

Figure 32 shows in-cylinder pressure variation versus crank angle for ROME-producer gas combinations at 80% load using different nozzle hole sizes. The peak pressure depends on the combustion rate and on how much fuel is taking part in rapid combustion period. The uncontrolled combustion phase is governed by the ignition delay period and by the mixture preparation during the delay period. Therefore, mixture preparation and slow burning nature of producer gas during the ignition delay period are responsible for the variation of peak pressure and maximum rate of pressure rise. From figure 27, it is observed that, at the same brake power the IMEP is comparatively lower for ROME-producer gas combination compared to diesel-producer gas combination as its peak pressure and rate of pressure rise is lower.

ROME-producer gas with 0.2 mm nozzle hole diameter results in higher peak pressure as shown in Figure 27. Lower nozzle hole size results in faster combustion and thus rapid pressure increase due to better fuel spray atomization and mixing. It could also be attributed to combined effect of longer ignition delay due to improper spray pattern and mixing rate caused by the use of larger nozzle hole size. Higher heat release rate versus crank angle for ROME-producer gas combinations at 80% load with different nozzle hole sizes. ROME-producer gas operation with 0.25 and 0.3 mm nozzle hole size results in lower heat release rate compared to the operation with 0.2 mm nozzle. Higher heat release rate was observed for lower nozzle hole size showing a more rapid combustion and concentrated heat release process, while flatter and broader heat
release shapes were observed for higher nozzle sizes. 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 0.25 and 0.3 mm nozzle in the diffusion combustion phase compared to 0.2 mm nozzle operation.

This paper investigates the influence of carburetor and nozzle geometry on dual fuel engine combustion process under real engine conditions. Optimizing air-gas mixer with an appropriate carburetor can significantly affect the performance of the dual fuel engine performance. The liquid fuel penetration for different nozzles was dependent on the injection pressure and in-cylinder conditions. ROME-producer gas operation resulted in improved performance with optimized values of fuel injection pressure (230 bar), number of nozzle hole (4 hole) and size (0.2 mm). The increase in injection pressure, hole number with smaller hole size could lead to efficient mixture preparation resulting in low PM, HC and CO emissions. However, NOx emission increases due to the rise of the combustion temperature. Significant improvements in power output and the trade-off between PM and NOx emissions can be obtained for dual fuel operation if pilot injection (230 bar) is used in conjunction with 4 hole nozzle and 0.2 mm hole diameter.

REFERENCES


