# STUDY OF PERFORMANCE OF A DUAL FUEL ENGINE USING BIOGAS AND PRODUCER GAS

Thesis

Submitted for the partial fulfillment of the Degree

of

*Doctor of Philosophy*

By


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*Under the guidance of*


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# Declaration

I, hereby certify that the work which is being presented in this thesis, entitled "Study of performance of a dual fuel engine using biogas and producer gas" in fulfillment of the requirements for the award of the Degree of Doctor of Philosophy submitted to Department of Mechanical Engineering, Thapar Institute of Engineering & Technology (Deemed to be University), Patiala is a record of bonafide research work carried out by me during a period from January 2014 to December 2019 under the supervision of Prof. S.K. Mohapatra, Senior Professor, Mechanical Engineering Department, Thapar Institute of Engineering & Technology (Deemed to be University), Patiala.

The results presented in the thesis have not been submitted in part or full to any other University or Institute for the award of any other degree or diploma.

Sohan Lal

Dated: 15.01.2021

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# Certificate

It is certified that the work contained in the thesis entitled **"Study of performance of a dual fuel engine using biogas and producer gas"** by Sohan Lal, a student in the Department of Mechanical Engineering, Thapar Institute of Engineering & Technology (Deemed to be University), Patiala, India, for the award of the degree of **Doctor of Philosophy** has been carried out under my supervision and this work has not been submitted elsewhere for the degree.

Dated: 15/01/2021

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# Abstract

The disposal of biomass waste in India has been a challenge in urban as well as rural areas. Biomass is one of the most common forms of renewable energy. The agricultural residue generated from the agro-industry can be beneficially utilized in power generation in rural areas. Many states in India generate a large amount of crop residue during the harvesting season. At present, a large amount of crop residue is burnt in the open field that causes environmental pollution. In urban areas, solid waste generated from the kitchen vegetable also becomes an unhandled problem. This cumulative waste creates a terrible effect on the environment and causes many health issues in all developing countries. The purpose of the use of a dual fuel engine is to utilize biomass residue and kitchen waste for power generation and to reduce emissions levels. This utilization of biomass residue helps in reducing fossil fuel consumption. Thus, dual fuel technology can play a significant role in decreasing the dependency on conventional diesel fuel.

In the present research, producer gas was generated from crop residue (cotton stalk) and waste wood (sawdust) by using a downdraft gasifier. The producer gas was cooled and cleaned by using a water scrubber. Biogas was generated from the kitchen vegetable waste using an anaerobic digester. The influence of utilizing 'producer gas and biogas' was investigated to study the performance and emission characteristics of a dual fuel engine.

The dual fuel CI engine was run at different compression ratios and different brake power values ranging from 0–4.0 kW in steps of 0.8 kW. The results of the dual fuel engine using producer gas-diesel and biogas-diesel were discussed. An average reduction of 63.62% HC emission was achieved by increasing CR from 12–18 at 3.2 kW brake power as compared to diesel mode. Further, NOx and SOx emission levels were reduced by 56.05% and 69.70% in producer gas-diesel and biogas-diesel mode respectively. Brake thermal efficiency improved at higher compression ratio and injection pressure values in both the dual fuel modes. Maximum diesel fuel substitution of 58.02% and 48.25% in producer gas-diesel and biogas-diesel mode respectively was observed at a compression ratio of 18. Further, the reduction in noise level was observed under the dual fuel mode of operation.

**Keywords:** Gasification, agriculture residue, biogas, kitchen waste, dual fuel engine, injection pressure, performance, and emission.

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# Nomenclature

**Abbreviations**

| | |
|---|---|
| IC | internal combustion |
| CI | compression ignition |
| SI | spark ignition |
| NOx | oxides of nitrogen, (ppm) |
| HC | hydrocarbon, (ppm) |
| CO | carbon monoxide, (ppm) |
| LPG | liquefied petroleum gas |
| CNG | compressed natural gas |
| BSFC | brake specific fuel consumption, (kg/kWhr) |
| BMEP | brake mean effective pressure, (bar) |
| BSEC | brake specific energy consumption, (MJ/kWhr) |
| VCR | variable compression ratio |
| BTE | brake thermal efficiency (%) |
| RPM | revolution per minutes |
| TDC | top dead centre |
| WC | water column, (mm) |
| BP | brake power, (kW) |
| ppm | parts per million |
| CR | compression ratio |
| CA | crank angle (degree) |
| dB(A) | decibel |
| EGT | exhaust gas temperature, (℃) |
| MSW | municipal solid waste |
| MT | metric tonne |
| CP | cylinder pressure (bar) |
| PCP | peak cylinder pressure (bar) |
| P | Pressure, (bar) |
| ITE | indicated thermal efficiency |

**Notations**

| | |
|---|---|
| $m_d^*$ | mass flow rate of diesel fuel, (kg/s) |
| $m_g^\#$ | mass flow rate of producer gas, (kg/s) |

**Greek symbols**

| | |
|---|---|
| ° | degree |
| **%** | percentage |
| θ | theta |

# Chapter 1
# Introduction

Disposal of agricultural waste in India has been a challenge. Farmers resort to the burning of agricultural waste that increases air pollution. The annual biomass generated from agriculture waste in Punjab state alone is 24.83 metric tonne [1]. The various forms of biomass are available like forest waste (twinges, leaves, and wood), agriculture waste (rice husk and straw), and industrial waste (wood chips, carpentry shop waste, and sawdust). The government of India also runs a national programme on power generation from biomass for the optimum utilization of a variety of agro-based biomass [2]. In addition, vegetable and food waste generate a significant contribution to municipal waste in urban areas. The above said waste creates a terrible effect on the environment and causes health issues (i.e., stomach pain, vomiting and diarrhea, cholera, typhoid, malaria, and respiratory allergies). According to the food and agriculture organization of UN reports that one-third of the food created for human use per annum goes as waste [3]. The utilization of biomass for power generation is a conventional way, such as the use of biomass in rural areas for heating purposes for domestic needs. India generates a huge amount of agricultural residue per year [2]. To utilize a large amount of surplus biomass for power generation, dual fuel technology can be used.

## 1.1 Biomass potential

India, the growth of the country depends on the agriculture industry. However, the management of agricultural waste is not controlled as the municipal solid waste in the country. The owner of the farming land predominately handles agricultural waste. In developing countries, the demand for food has dramatically increased. Hence, the magnitude of agro-products increase in agriculture waste generation. The disposal of this waste has led to a rise in global environmental pollution [4]. Agricultural waste is defined as waste material (reside) left after various farming operations. Agricultural waste, according to the United Nations, mainly includes pesticide that enters the water, soil or air, slit, salt, poultry farms, and manure etc. [4]. Biomass residue generated by many states of India is shown in Table 1.2.

In addition to the above, according to the world energy council, spoiled food is also treated as agricultural waste [5]. The crop residue such as stubble and stalks, leaves and seed pods etc. According to the Indian ministry of new and renewable energy (MNRE) report, India generates an average of 500 metric tonne (MT) of crop residue every year [1]. The report shows that majority of the crop residue was used as fodder, fuel for industrial and domestic purposes [2]. However, after using a surplus of 140 MT is available for disposal out of this approximately 92 MT of biomass residue is burnt every year in India [1]. The agricultural waste generated by four Asian countries in MT/year is given in Figure 1.1 [2].

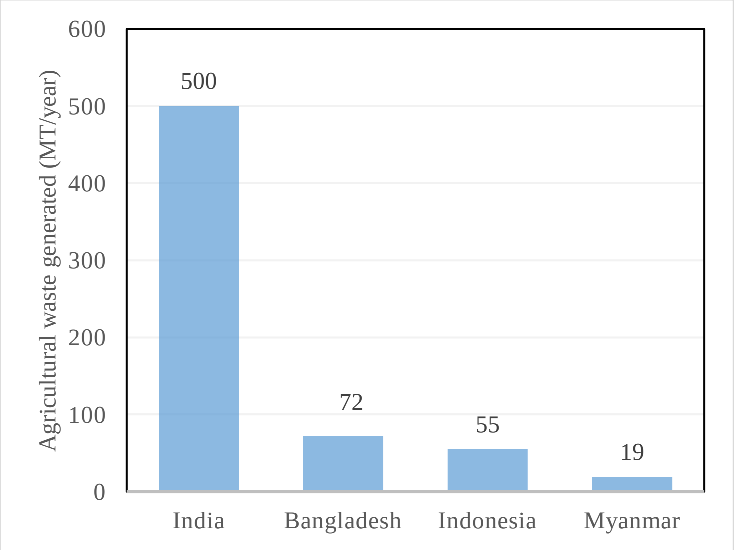



**Figure 1.1 Agricultural waste generated by four Asian countries** [2]

Agricultural waste can be utilized in many industrial processes. However, the transportation cost, handling, and collection cost can be more than its favorable use. To use the agricultural waste and food waste using two modern techniques are gasification of dry biomass and the anaerobic digestion process is used for wet biomass such as kitchen vegetable waste and food waste.

## 1.2 Energy scenario in India

With the fast depletion of fossil fuel and a large amount of surplus biomass residue in the country, the use of renewable resources is mandatory. The reduction of diesel engine emission level dual fuel diesel engine is the best approach for the utilization of producer gas and biogas. According to Punjab, state government policies on renewable energy sources

**Table 1.1 Energy status in India up to August 2020** [6]

| Year | Energy | | | | Peak | | | |
|---|---|---|---|---|---|---|---|---|
| | Requirement | Availabilit | Surplus(+)/Deficts(-) | | Peak Demand | Peak Met | Surplus(+) / Deficts(-) | |
| | × $10^5$ (MU)# | × $10^5$ (MU) | × $10^5$(MU) | (%) | × $10^5$ (MW)~ | × $10^5$ (MW) | × $10^5$ (MW) | (%) |
| 2009-10 | 8.31 | 7.47 | -0.84 | -10.11 | 1.19 | 1.04 | -0.15 | -12.72 |
| 2010-11 | 8.62 | 7.88 | -0.73 | -8.50 | 1.22 | 1.10 | -0.12 | -9.84 |
| 2011-12 | 9.37 | 8.58 | -0.79 | -8.46 | 1.30 | 1.16 | -0.14 | -10.63 |
| 2012-13 | 9.96 | 9.09 | -0.87 | -8.73 | 1.35 | 1.23 | -0.12 | -8.98 |
| 2013-14 | 10.02 | 9.60 | -0.42 | -4.23 | 1.36 | 1.30 | -0.06 | -4.49 |
| 2014-15 | 10.69 | 10.31 | -0.38 | -3.57 | 1.48 | 1.41 | -0.07 | -4.73 |
| 2015-16 | 11.14 | 10.91 | -0.24 | -2.11 | 1.53 | 1.48 | -0.05 | -3.20 |
| 2016-17 | 11.43 | 11.35 | -0.08 | -0.66 | 1.60 | 1.57 | -0.03 | -1.63 |
| 2017-18 | 12.13 | 12.05 | -0.09 | -0.71 | 1.64 | 1.61 | -0.03 | -2.02 |
| 2018-19 | 12.75 | 12.68 | -0.07 | -0.55 | 1.77 | 1.76 | -0.01 | -0.84 |
| 2019-20 | 12.91 | 12.84 | -0.07 | -0.51 | 1.84 | 1.83 | -0.01 | -0.69 |
| 2020-21* | 5.15 | 5.14 | -0.02 | -0.37 | 1.72 | 1.70 | -0.01 | -0.64 |

* Up to Aug 2020 (Provisional), Source: Central electricity authority, India
# MU represents mega units, ~ MW represents megawatts

**Table 1.2 State-wise crop residue generated, residue surplus and burned [1]**

| States | Crop residue generation (Metric tonne) | Crop residue surplus (Metric tonne) | Crop residue burned (Metric tonne) |
|---|---|---|---|
| Uttar Pradesh | 59.97 | 13.53 | 21.92 |
| Punjab | 50.75 | 24.83 | 19.65 |
| Maharashtra | 46.45 | 14.67 | 7.42 |
| Andhra Pradesh | 43.89 | 6.96 | 2.73 |
| West Bengal | 35.93 | 4.29 | 4.96 |
| Karnataka | 33.94 | 8.98 | 5.66 |
| Madhya Pradesh | 33.18 | 10.22 | 1.91 |
| Rajasthan | 29.32 | 8.52 | 1.78 |
| Gujarat | 28.73 | 8.90 | 3.81 |
| Haryana | 27.83 | 11.22 | 9.08 |
| Bihar | 25.29 | 5.08 | 3.19 |
| Orissa | 20.07 | 3.68 | 1.34 |
| Tamil Nadu | 19.93 | 7.05 | 4.08 |
| Assam | 11.43 | 2.34 | 0.73 |
| Chhattisgarh | 11.25 | 2.12 | 0.83 |
| Kerala | 9.74 | 5.07 | 0.22 |
| Jharkhand | 3.61 | 0.89 | 1.1 |
| Uttarakhand | 2.86 | 0.63 | 0.78 |
| Himachal Pradesh | 2.85 | 1.03 | 0.41 |
| Jammu &Kashmir | 1.59 | 0.28 | 0.89 |
| Manipur | 0.90 | 0.11 | 0.07 |
| Goa | 0.57 | 0.14 | 0.04 |
| Meghalaya | 0.51 | 0.09 | 0.05 |
| Nagaland | 0.49 | 0.09 | 0.08 |
| Arunachal Pradesh | 0.40 | 0.07 | 0.04 |
| Sikkim | 0.15 | 0.02 | 0.01 |
| Mizoram | 0.06 | 0.01 | 0.01 |
| Tripura | 0.04 | 0.02 | 0.02 |
| **Total** | **501.73** | **140.84** | **92.81** |

have proposed 600MW of power generation from agriculture residue and 50MW power generation from urban, municipal solid waste by 2022 [7]. The current energy status in India is shown in Table 1.1. The highest biomass generating state in India is Uttar Pradesh, according to the survey as shown in Table 1.2.

## 1.3 Existing method for utilization of agricultural residue

The agricultural residue generated in India is utilized by the following existed methods:

- Composting
- Power generation
- Open burning

### 1.3.1 Composting

In India, composting is an alternative to landfills. Mostly in the villages, compost the agricultural residue instead of a landfill. It is because organic residue becomes nutrient-rich organic fertilizer by composting.

### 1.3.2 Power generation

The combustion of biomass in presence of air to produce heat is known as direct combustion. Combustion is used to convert the chemical energy stored in biomass residue into heat.

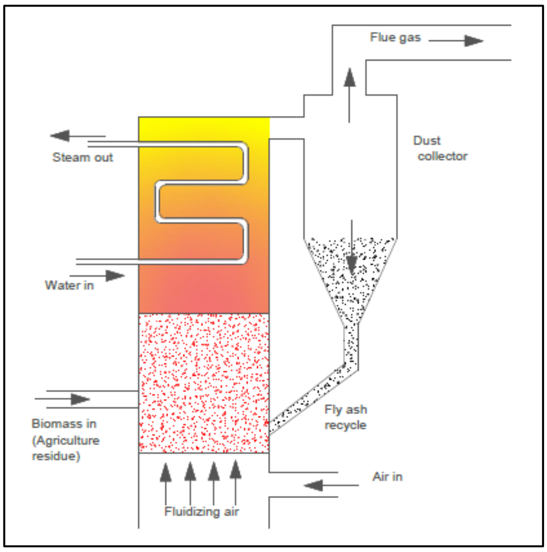



**Figure 1.2 Schematic diagram of a fluidized bed boiler**

Biomass is used for domestic heating and cooking purpose. At an industrial scale, in the last two decades in the state of Punjab, India, the power generation from biomass has been done using direct combustion in a fluidized bed boiler as shown in Figure 1.2. The details of the power generation projects commissioned by the Punjab Energy Development Agency (PEDA) are shown in Table 1.3 [7].

**Table 1.3 Biomass power projects commissioned using paddy straw** [7]

| S.No. | Name of the company | Capacity (MW*) |
|---|---|---|
| 01 | M/s Malwa Power Ltd. Vill. Gulabewala, Distt. Mukatsar | 6.0 |
| 02 | M/s Dee Development Engg. Pvt. Ltd Vill.GaddaDhob, Ferozepur | 8.0 |
| 03 | M/s Universal Biomass Energy Pvt. Ltd. Vill. ChannuTeh. Malout, Distt. Sri Mukatsar Sahib | 14.5 |
| 04 | M/s. Punjab Biomass Power Pvt. Ltd.Distt. Patiala | 12.0 |
| 05 | M/s. Green Planet Energy Pvt. Ltd.Binjon, Distt. Hoshiarpur | 6.0 |
| 06 | M/s. Green Planet Energy Pvt. Ltd.Bir Pind, Distt. Jallandhar | 6.0 |
| 07 | M/s Viaton Energy Pvt. Ltd.Khokhar Khurd Distt. Mansa | 10.0 |
| | **Total** | **62.5** |

*MW= Megawatt

### 1.3.3 Open burning

The burning of biomass in open-air generates many environmental problems. It includes global warming, greenhouse gases, particulate matter (PM), and smog [8]. The burning of crop residue mainly increases air pollution such as CO, $CO_2$, NH3, NOx, SOx, and PM [8]. On 10 November 2019, there was extensive crop burning in the area of Punjab (India). The image was recorded with the help of Visible Infrared Imaging Radiometer Suite (VIIRS) on the Suomi NPP satellite. Red outlines show the presence of crop burning, as shown in Figure 1.3 [9].

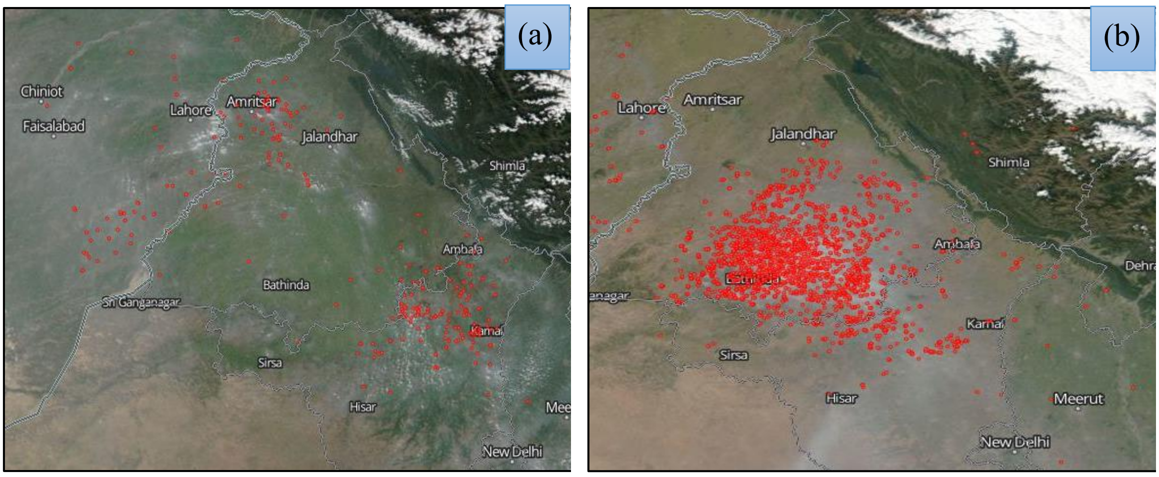



**Figure 1.3 NASA earth observatory image of Punjab state (India) on (a) 10 October 2018; (b) 10 November 2019** [9]

## 1.4 Disposal methods of kitchen waste

- Landfill
- Anaerobic digestion

### 1.4.1 Landfill method

Landfill method is used for dumping biomass waste at a landfill site. It is the oldest method of dumping waste. In the urban area, municipal solid waste is comprised of kitchen and vegetable waste, other households, and commercial reuse [10]. The lack of transportation and collection is the reason for the buildup of kitchen waste in Indian cities [11]. Because the accumulation of solid waste has become a source of pollution that creates the cause of global warming, health issues, and air pollution. Metropolitan cities (Kolkota, Chennai, Mumbai, and Delhi) have more than 42% of the Indian urban population [12].

Landfill sites harm the health and environment [13,14]. Under anaerobic conditions, open garbage release $CH_4$ gas from the decomposition of biodegradable waste. This causes the risk of fires, and it is a significant contributor to global warming. In the summer season, the odor is the main problem when the temperature is above 45℃ [15]. The major garbage collected in Landfill sites in New Delhi, India, is shown in Figure 1.4.

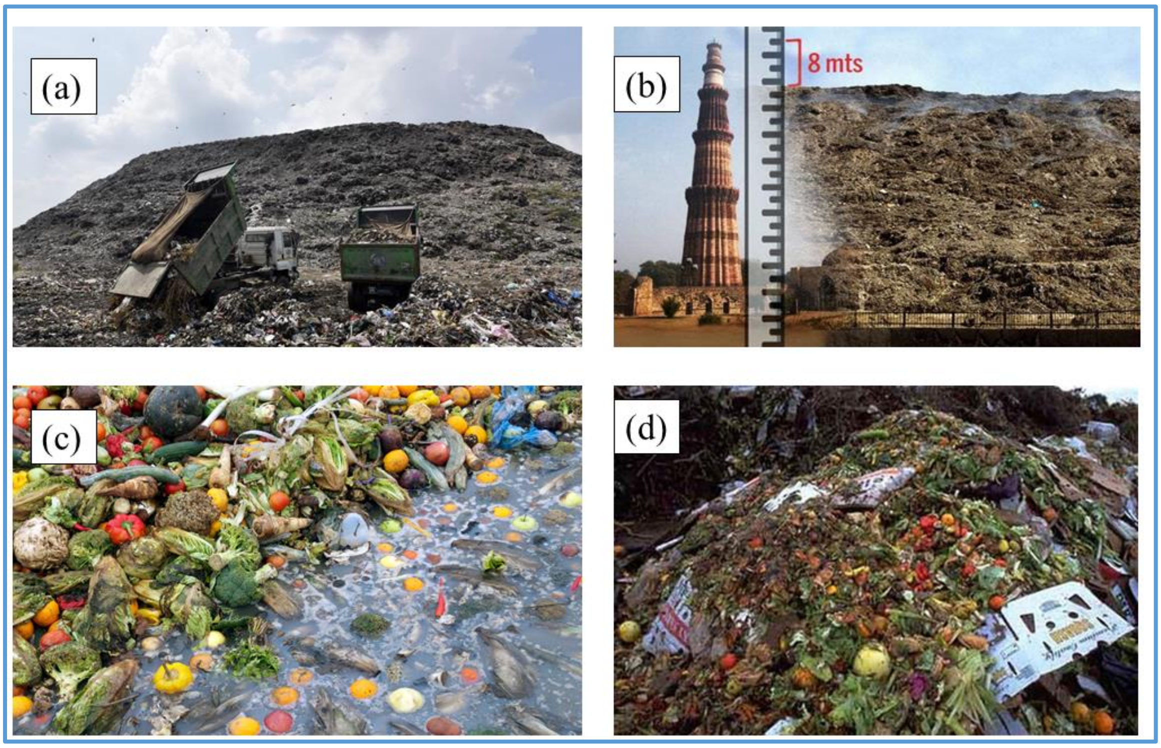



**Figure 1.4 (a) Ghazipur Landfill site, (b) Ghazipur garbage dump, (c) and (d) Food waste**

### 1.4.2 Anaerobic digestion

Anaerobic digestion is the process of converting organic material into gas, known as biogas. Biogas is mainly mixture of methane and carbon dioxide. The anaerobic digestion process is accrued in a biogas plant. It is an alternative method for composting to lower organic material landfill disposal. Biogas is generated during the process of anaerobic digestion. In this process, a biological breakdown of organic material in the oxygen-free environment. Biodegradable waste is transformed into a mixture of $CH_4$ and $CO_2$ with a minor quantity of hydrogen sulfide ($H_2S$), the trace of nitrogen, ammonia, hydrogen, and oxygen. The water vapor and dust particles are also present in it.

The biological breakdown of organic waste occurs naturally in old landfills at atmospheric temperature. Therefore, open-air dumps generate biogas. To lower the escape of biogas into the environment, biogas is produced using digester with optimizing all factors such as pH, temperature, feedstock, and time to increase the yield of methane in biogas. The energy consumed by a biogas plant is approximately 25–30% of the energy demand [16]. At present, biogas is mainly used for domestic cooking and heating purposes.

From the above existing methods, the gasification of biomass residue can be used for power generation with a high overall conversion efficiency. The producer gas generated in the gasifier is used in the CI dual fuel engine for power generation.

Besides the agricultural residue, vegetable and food waste also have the problem of disposal in the urban areas. Landfill sites have not any additional space. In the present study, biogas generation from the kitchen vegetable waste is used to overcome the problem of landfill space. For this, an anaerobic digester can be used to generate biogas.

## 1.5 Motivation and objectives of the present investigation

The goal of the present study is to minimize the dependency on fossil diesel fuel by using alternate renewable fuels. This includes utilizing the biomass waste generated from agricultural residue and the kitchen vegetable waste for power generation. This alternative source of energy (i) leads to reduced dependency on the conventional diesel fuel, and (ii) reduces the emission of harmful gases which otherwise get generated by the open burning of biomass residue. This research is also important because of the large availability of biomass/agriculture residue in the state of Punjab, India.

Various researchers have studied the performance and emission characteristics of dual fuel engine using a simulated gas (gas cylinders). However, very limited literature is available on the performance studies of dual fuel mode diesel engine used under conditions of variable compression ratio and varying injection pressure. There is limited reporting on the emission characteristics of dual fuel diesel engines with regards to sulfur dioxide (SOx) levels.

The objective of this study is to evaluate the performance of a dual fuel engine using biogas and producer gas by varying the compression ratio and injection pressure. In addition emission and noise levels in dual fuel mode have been investigated.

## 1.6 Organization of the thesis

The present **chapter** of the thesis work sets the background study for the utilization of agricultural crop residue and kitchen waste (vegetables and food ) are emphasized. The adverse impact of waste on the environment is also discussed. **Chapter 2** presents the literature concerning the operation under dual fuel mode fueled using producer gas-diesel and biogas-diesel. The experimental work conceded by many researchers on the performance, emission, engine parameters, and its impact on the engine performance and emission parameters are reviewed. **Chapter 3** represents the details of the experimental set-up, including the working principle of different types of equipment used for experimentation work. The data calculations, experimental procedure, and uncertainties in experimentation are described. **Chapter 4** represents the results of producer gas-diesel and biogas-diesel dual fuel engine investigations of performance and emissions parameters. **Chapter 5** describes the conclusions of the present work and future work for the utilization of biomass residue in a dual fuel CI engine.

## 1.7 Summary

It is observed from the literature survey that many authors have work done along with diesel to find out the effect of load on the performance and emissions parameters. However, it is observed that very little literature has been reported on injection timing and pressure parameters using biogas-diesel and producer gas-diesel in dual fuel engine. More emphasis is required to find out agricultural waste utilization in agricultural base states like Punjab, Haryana, Uttar Pradesh in (India).

The gasification process for the utilization of agricultural waste could be an alternate method for the solution of biomass utilization. Anaerobic digestion based (biogas plants) could be used for utilization of kitchen waste as community practice.

# Chapter 2
# Literature review

This chapter represents a review of literature on dual fuel diesel engine running with various kinds of gaseous fuels. Emphasis is given to the dual fuel engine using biogas and producer gas. It also includes diesel fuel saving, brake thermal efficiency, the effect of compression ratio, injection pressure on the performance, and emission characteristics of a dual fuel engine using gaseous fuels. The literature review is divided mainly into two sections viz producer gas-diesel and biogas-diesel used in a dual fuel CI engine.

## 2.1 Engine modification into dual fuel

Due to the low cetane number, gaseous fuels cannot be used alone in CI engines. Naber et al. [17] have found the ignition of methane experimentally and the local temperature in the combustion of 1200 K for ignition in 2ms. Hodgins et al. [18] reported that in a diesel engine at the end of the compression stroke the temperature is about 1000 K. Therefore, the gaseous fuels are unable to ignite in CI engine within the required time. Hence, other methods are needed to be looked into using gaseous fuel in CI engines. According to Liu [19], the following methods can be used to achieve it:

a) Altering the CI engine into a spark-ignition engine (SI) (higher compression ratio SI engine).
b) Supply the gaseous fuel with the air through a suction port of the CI engine and retain the fuel injection system for pilot fuel.
c) Supply the gaseous fuel to the engine directly and retain the fuel injection system for pilot fuel.

The second method for dual fuel engine was appropriate because of little modification required in the conventional CI engine. The working of a dual fuel engine is shown in Figure 2.1. The dual fuel CI engine sucks and compresses the mixture of air and gaseous fuel. The small amount of pilot diesel fuel supplied by a fuel injection system of CI engine for igniting the air-gas mixture. The supply of pilot fuel differs with the engine parameters and engine loading conditions. Mansour et al. [20] investigated that at a part load, the gas supply is reduced with the help of a valve. However, the air supply to the engine reduces which results in a reduction in engine power and efficiency. Therefore, dual fuel engines are not throttled on the air suction sides. There is a need for the optimal ratio of pilot fuel and gaseous fuel supply at different loading conditions of the dual fuel engine.

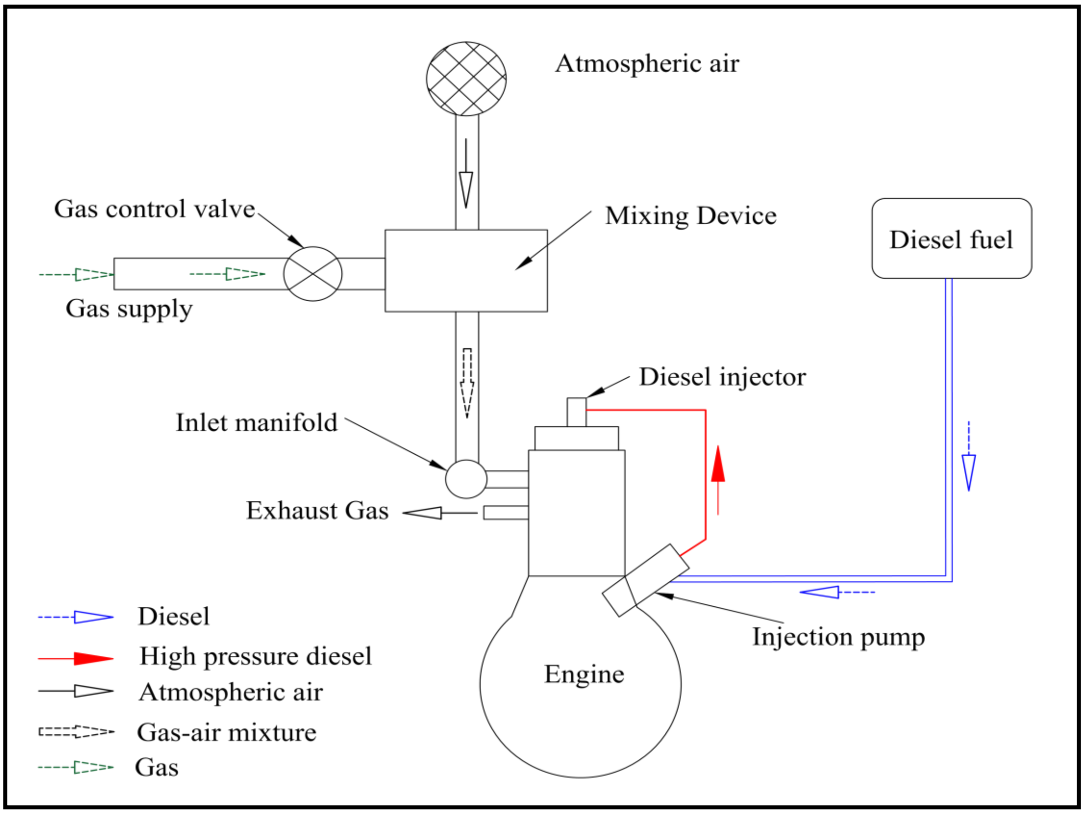



**Figure 2.1 Schematic diagram of a dual fuel system**

## 2.2 Review of dual fuel CI engine using producer gas

The dual fuel technique allows the gaseous fuel to be utilized in a diesel engine. However, limitation to the performance of a dual fuel engine creates several issues that need to be addressed on change of engine parameters like amount of pilot fuel, load, speed, compression ratio and injection pressure, and gaseous fuel compositions. The performance and emissions of the dual fuel engine are affected by the above-said parameters. Different researchers to evaluate the effect of the above parameters on the performance and emission characteristics of a dual fuel engine have done various studies. Many researchers investigated using primary gaseous fuel and different types of pilot fuel. The producer gas, the gaseous fuel generated from the gasifier by using renewable biomass as feedstock. The main constituents of the produced gas are $CH_4$, CO, and $H_2$.

### 2.2.1 Effect of load on the performance parameters of dual fuel engine

Ramadhas et al. [21] reported maximum brake thermal efficiency (BTE) 19.90% and 21.00% at 70.01% load condition in dual fuel mode fueled with producer gas generated from coir-pith and wood chips respectively, as compared to 25.00% in diesel mode. On a further increase of load, the BTE of coir-pith dual-fueled engine declined. Ramadhas et al. [22] reported similar findings using rubber seed oil and producer gas generated from coir-pith in the same engine set-up. The BTE of dual fuel engine has been reported lower than diesel fuel mode. Hence, higher exhaust gas temperature under dual fuel mode than diesel mode.

However, using producer gas with bio-diesel as pilot fuel. Singh et al. [23] found higher BTE using producer gas-diesel mixture. Malik et al. [24] have evaluated the performance of the dual fuel engine using rice straw as feedstock for the gasifier. Maximum diesel fuel replacement was reported by 60% and also found that power generation was cheaper than conventional power generation cost. Martinez et al. [25] reported BTE rises with an increase in $H_2$ percentage of syngas fuel at high loads. However, BTE decreases at a lower load because of poor combustion of gaseous fuels. Shrivastava et al. [26] studied the performance and emissions of a dual fuel CI engine under producer gas-diesel. The BTE of the dual fuel engine was reduced by 5.45% compared to the diesel engine.

Many research [27–31] have reported that for dual fuel mode engine working at low loads, the reduced consumption of gaseous fuel results in lower thermal efficiency and higher CO emission as compared to standard diesel fuel operation. Sahoo et al. [32] reported maximum diesel fuel replacement up to 72.30% by using hydrogen gas as the secondary fuel, but the higher energy content of hydrogen gas results in higher NOx emissions.

Dhole et al. [33] have investigated that the use of $H_2$ gas as primary fuel enhanced the BTE at a higher load. However, it gives the opposite effect at a lower load. The BTE of a dual fuel engine was reduced under producer gas-diesel at all loading cases. The mixture of 60:40 proportions gives better BTE as compared to other ratios. HC and CO concentrations were the higher comparison to diesel mode in all cases of loading

Liu et al. [34] found that the effect of injection pressure on the emissions and performance of the common-rail CI engine in dual fuel mode. HC and CO emission decreased with a rise in injection pressure as shown in Figure 2.2. The maximum cylinder pressure and the peak heat released rate under dual fuel mode increased as the injection pressure increases. BSFC using a CI engine in dual fuel mode decreases with the increasing fuel injection pressure. However, BSFC under dual fuel mode was higher than diesel mode with the raise of injection pressure, as given in Figure 2.3. The NOx emissions decreased with the increase of injection pressure.

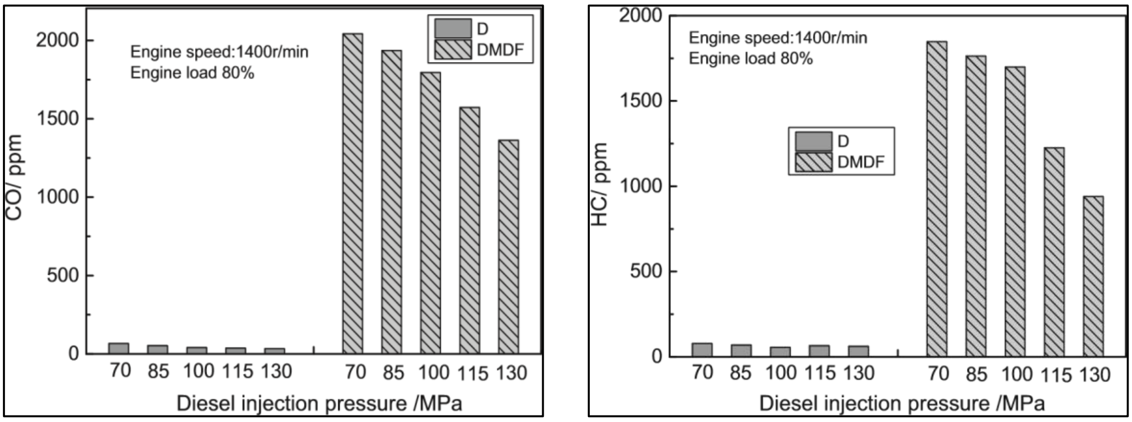



**Figure 2.2 CO and HC concerning diesel injection pressure** [34]

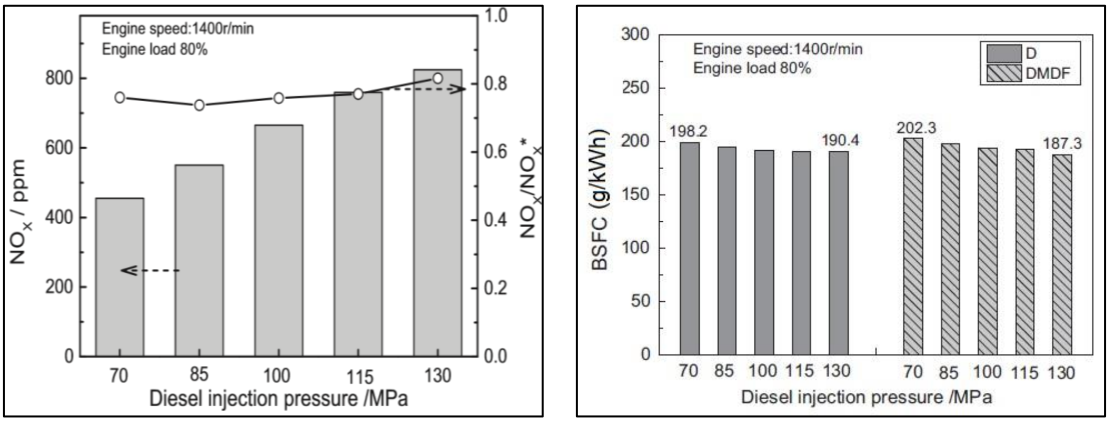



**Figure 2.3 NOx and BSFC concerning diesel injection pressure** [34]

Dhole et al. [33] have reported the effect of $H_2$ presence in producer gas under a dual fuel CI engine. The peak cylinder pressure (PCP) rises to begin with 30% substitution of $H_2$ at 40% load, and on full substitution of hydrogen, PCP falls at all cases of loadings. The ratio of 60:40 indicates lower PCP as compared to other mixture ratios used in experiments.

Jayashankara and Ganesan [35] have investigated experimental and computational studies on the effect of injection timing on combustion and emissions characteristics of a dual fuel engine using natural gas at low load and low speed. The brake thermal efficiency was improved with the increase of pilot fuel injection timing. The optimum pilot fuel injection angle was around a 12-degree crank angle. The HC emissions were not changed much. Lee et al. [36] demonstrated the feasibility and viability of biomass to power generation for rural applications. The maximum efficiency of the gasifier was reported as 84.6%. However, engine efficiency falls by 25.6-29.5%, with 20.3% for cardboard wood used in a gasifier. However, overall system performance, the integrated efficiencies were 20.6, 21.3%, and 23.0% for red oak, horse manure, and pine respectively. Further, using cardboard wood in the gasifier gives the minimum overall efficiency of 15.8%, comparing the performance of dual fuel engine using producer gas-diesel and natural gas-diesel.

Raman and Ram [37] concluded that the calorific value of producer gas improved with the rise in the temperature of supply air in the gasification process. The cold gas efficiency of the set-up was 88%. The dust particles in the generated gas before cleaning was 600 ppm and after the cleaning was 25 ppm. The maximum BTE of the dual fuel engine was 24%. It also reported that maximum BTE was achieved with the use of producer gas beyond the 85% engine loading conditions. Tippayawong et al. [38] used producer gas-diesel in dual fuel mode for water pumping in a remote area. The gas was produced in a gasifier using gasifier feedstock as waste wood and charcoal. An experimental study was done to verify the impact of producer gas and diesel on CI engine performance. The maximum substitution of diesel fuel was 60%. The overall outcomes of the experimental study were exhibited that CI engine with producer gas is a promising technique for rural area irrigation. Perez et al. [39] have reported that biomass gasification is mainly used for combined heat and power generation. The BTE was reduced by 23.4% at higher loads. The maximum BTE by 24% was achieved in the experimental study using producer gas.

### 2.2.2 Effect of load on the emission parameters of dual fuel engine

The generation of nitric oxides ($NO_x$) depends on the presence of $N_2$ in the air-fuel mixture and temperature of the combustion chamber [40]. Papagiannakis and Hountalas [41] reported that the concentration of NOx depends on the gaseous fuel in the gas-air mixture. Lower NOx emission by using natural gas-diesel as fuel for dual fuel engine comparison to diesel mode at similar running parameters. At low loads, the concentration of NOx emission was slightly lower. The low temperature of the combustion during lower load, which results in lower NOx formation. At higher load, in dual fuel mode, lower NOx emission reported comparison to diesel mode. Banapurmath and Tewari [42] and Uma et al. [43] have reported that the NOx formation under producer gas-diesel dual fuel mode was lower under all cases of loading conditions.

The amount of CO formation is a function of unburnt gaseous fuel availability and its temperature; both parameters control the rate of fuel decomposition and oxidation [40]. Papagiannakis and Hountalas [41] have reported CO emission higher in dual fuel mode comparison to diesel mode. Low CO emission was observed with the rise in engine load at a lower speed. This is because of an enhancement in the consumption of gaseous fuel. The CO emission at a higher load cannot be affected by engine speed significantly due to less combustion time availability. Similar results in dual fuel mode for CO emission at varying loads were found by Ramadhas et al. [21,22]. They also reported high $CO_2$ emission in the dual fuel mode in comparison with diesel mode. It was because of the existence of $CO_2$ and CO in the producer gas.

Papagiannakis and Hountalas [41] reported higher HC emission in dual fuel mode at low load compared to diesel mode. This is because of the low temperature and the air-fuel proportion was mainly responsible for higher HC emission under low load conditions. In addition, slower combustion allowing a small amount of fuel escaped from the combustion chamber. However, with growth in engine load, there was a decline in HC emission in dual fuel mode. The HC emissions were found considerably higher for all cases evaluated by authors under dual fuel mode comparison with diesel mode. Under dual fuel engine operation, soot concentrations were reported lower related to diesel mode for all cases of loading [41]. The experimental results by Uma et al. [43] showed that HC emission in producer gas-diesel mode was a little more than diesel mode. Banapurmath and Tewari [42] reported higher CO and HC concentrations under dual fuel mode using producer gas with different pilot fuels, namely, rice bran, honge, and neem oil than diesel mode. Singh et al. [23] investigated that the use of producer gas/bio-diesel, NOx emissions significantly reduced but increased the other emissions parameters.

Sayin and Canakci [44] studied the influence of injection timings on the performance and emissions of a dual fuel engine. The results indicated that $NO_x$ emission slightly increased while the addition of ethanol CO and HC emissions decreased. Hence, $CO_2$

emission increased because of enhanced combustion. Papagiannakis et al. [45] have conducted an experimental investigation using natural gas-diesel in a dual fuel CI engine. They reported the concentration of NOx emissions lower than conventional diesel fuel operation. They reported disadvantages of injection timings of liquid fuel at part loads on BTE, HC, and CO emission. Roy et al. [46] carried out experimental studies on supercharged dual fuel engine using hydrogen and $H_2$ containing fuel. They reported that the engine operates with 100% $H_2$ generates lower CO and HC emissions. The maximum NOx emission was 85-90% lower than other fuels. Sombatwong et al. [47] investigated the performance and emission of dual fuel engine using producer gas with varying the amount of diesel fuel. They reported diesel fuel saving was reduced at a higher engine load. It was because of the rich air-fuel mixture required at a higher load. The BSEC and BTE of dual fuel engine run with producer gas-diesel can be enhanced by raising the diesel fuel quantity. However, CO emissions were reduced by raising pilot fuel quantity.

Shrivastava et al. [26] reported that NOx emissions decreased by 18.6% under dual fuel mode using producer gas at a flow rate of 41 lpm at 100% load. CO and HC emission increased up to 17.39% and 15.38% respectively at 100% load conditions compared to diesel fuel mode. It was because of insufficient oxygen in the combustion chamber [26]. Tarabet et al. [48] experimentally studied using natural gas-biodiesel under dual fuel operation. The BSFC was reduced at high load conditions under dual fuel mode. The HC and CO emissions considerably increased, while NOx emission under dual fuel mode was lower because of the low temperature of combustion. Mittal et al. [49] investigated the characteristics of exhaust emissions of a dual fuel diesel engine using natural gas-diesel. The concentration of NOx emission decreased in dual fuel mode in comparison to diesel fuel in all cases of loads. The intensity of HC and CO concentrations were increased in dual fuel mode. It is because of the low oxygen present in the air-gas mixture and some fuel escape from the combustion chamber.

Yaliwal et al. [50] studied the design parameters of the diesel engine in dual fuel mode. The engine was operated for six hours at different injection pressure in dual fuel mode. BTE was enhanced by 5.65%, with the increase of injection pressure as shown in Figure 2.4. NOx emissions increased by 15.68%. However, the other pollutants, such as smoke, HC, and CO reduced by 19.0, 11.2, and 17.6% respectively. The variation of HC emissions with different injection pressure is shown in Figure 2.5.

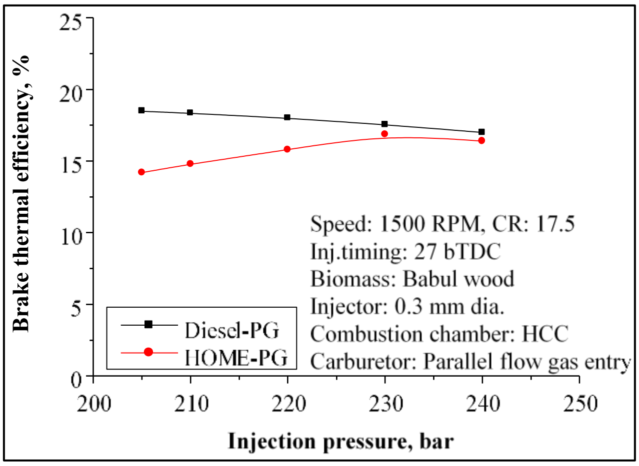



**Figure 2.4 Variation of BTE with injection pressure** [50]

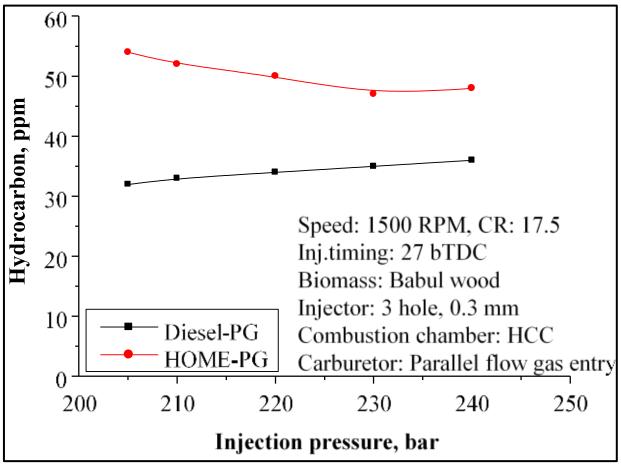



**Figure 2.5 Deviation of HC with injection pressure** [50]

Nayak and Mishra [51] investigated the emission characteristics of a dual fuel engine. The producer gas was generated in the gasifier using waste babul wood as feedstock. The concentration of NOx emission was decreased in comparison to diesel mode. However, CO, $CO_2$ and HC emissions were higher using producer gas-diesel fuel. The results of the study show that NOx reduced using waste wood, as gasifier feedstock.

### 2.2.3 Effect of compression ratio on performance parameters of dual fuel engine

Lal and Mohapatra [52] reported that brake thermal efficiency increases at a higher compression ratio due to higher combustion temperature. Choudhary et al. [53] reported maximum BTE was achieved by 21.1% at a compression ratio of 19.5. The peak cylinder pressure (PCP) has a rising tendency with a rise in compression ratio. The maximum PCP was 69.06 bar at a compression ratio of 19.5. Rahman et al. [54] reported that BTE of dual fuel mode increased with the rise in compression ratios.

### 2.2.4 Effect of compression ratio on emissions parameters of dual fuel engine

Choudhary et al. [53] have reported NOx emission higher at higher compression ratios. However, the concentration of CO emissions was decreased with the rise in compression ratio. This was because of the higher temperature inside the cylinder during a higher compression ratio resulting in better combustion of gaseous fuel. Yaliwal et al. [55] investigated performance and emission characteristics using biodiesel (methyl ester) and producer gas in dual fuel engine at different CRs. BTE improved by 24.40% with an increase of CR from 15 to 17.5. Exhaust gas temperature reduced by 26.02%, with a rise in compression ratio from 15 to 17.5. However, CO and HC concentration decreased by 44.50% and 30.20% with an increase of CR from 15 to 17.5 respectively. Whereas, NOx emissions increased by 24.40%.

The detailed summary of the dual fuel CI engine used in the experimental investigation by the various researchers is shown in Table 2.1.

**Table 2.1 Summary of the experimental investigation using producer gas**

| Researcher(s) | Pilot Fuel | Engine used for the test | Primary Fuel | Key Results |
|---|---|---|---|---|
| **Sridhar et al. (2005)** [56] | Diesel | Ashok Leyland make-ALU680 96 kW @ 1500 rpm naturally aspirated diesel engine | Producer gas (wood and coconut shells) | Oxygen availability limits maximum diesel fuel saving at rated load |
| **Singh et al. (2007)** [23] | Diesel | Naturally aspirated multi-cylinder, diesel Genset (DG) | Producer gas (wood) | CO emission increased by 16.3% and NOx emission decreased by 80%. |
| **Ramadhas et al. (2008)** [22] | Diesel | CI diesel engine, Single cylinder, 4-stroke, DI | Producer gas (Coir-pith) | BTE decreased at part load. |

| Researcher(s) | Pilot Fuel | Engine used for the test | Primary Fuel | Key Results |
|---|---|---|---|---|
| **Malik et al. (2009)** [24] | Diesel | 5 kW, Kirloskar, four-stroke, single cylinder, engine | Producer gas (rice husk) | The maximum diesel fuel saving was 47%. |
| **Banapurmath and Tewari (2009)** [42] | Diesel, Biodiesel | Single-cylinder, 4-stroke, constant speed, 5.2 kW, Make Kirloskar, with an eddy-current dynamometer | Producer gas (wood) | Lower burning velocity of producer gas and lower CV were responsible for lower BTE. |
| **Lata et al. (2012)** [57] | Diesel | 62.5 kW BP, 4-cylinder diesel engine, turbocharged, with intercooler. | LPG and Hydrogen | Knocking was noticed during the DFM with a 50% substitution of hydrogen. However, 30% substitution gives the best results |
| **Sahoo and et al. (2012)** [32] | Diesel | Single-cylinder, 4-stroke, constant speed, 5.2 kW, Make Kirloskar, | Simulated syngas($H_2$ and CO in syngas) | BTE rise with $H_2$ percentage, the higher flame speed reported with 100 % $H_2$. |
| **Singh and Maji (2012)** [58] | Diesel | 2-cylinder, 4-stroke, 7.3 kW engine | CNG | Maximum 90% diesel fuel saving |
| **Tippayawong et al. (2013)** [38] | Diesel | 5.5 kW engine, inline, 4-stroke, NA, DI | Producer gas (Waste wood) | Maximum 60% substitution of diesel fuel reported. |
| **Shrivastava et al. (2013)** [26] | Diesel | 4.4 kW CI engine of power, single cylinder, 4-stroke | Producer gas (wood chips and mustard oil cake) | HC and CO concentrations were observed higher by 17.39% and 15.38% respectively, at full load. |
| **Raman and Ram (2013)** [37] | Diesel | Mitsubishi DI 800 1995, naturally aspirated, Inline, 4-stroke, DI, diesel engine. | Producer gas (wood chips and charcoal) | Maximum diesel fuel saving up to 60%. |
| **Karabektas et al. (2014)** [29] | Diesel | Single-cylinder, 4-stroke, constant speed, DI | CNG | Lower NO emissions and HC and CO emissions were higher in dual fuel mode. |
| **Mittal et al. (2014)** [49] | Diesel | Six-cylinder, 4-stroke, 652 kW engine | Natural gas | Higher HC and CO emissions were reported. |
| **Yaliwal et al. (2016)** [50] | Biodiesel | A 4-stroke, water-cooled, single cylinder, CI engine | Producer gas (babul wood) | Biodiesel/producer gas shows the best combination for |

| Researcher(s) | Pilot Fuel | Engine used for the test | Primary Fuel | Key Results |
|---|---|---|---|---|
| | | | | smooth engine working and the combustion process depends upon the design of the combustion chamber and nozzle parameters. |
| **Dhole et al. (2016)** [33] | Diesel | 4-stroke, CI, constant speed, vertical, water-cooled, DI, turbocharger | Producer gas and hydrogen gas | 60:40 of producer gas and $H_2$ show optimum results. |
| **Nayak and Mishra (2017)** [51] | Diesel, Biodiesel | 14 HP, two-cylinder four-stroke, 1500 rpm and CR 16:1, Make-Prakash Diesel Pvt. Ltd. | Producer gas (babul wood) | CO, $CO_2$, and HC reported higher under dual fuel mode. |
| **Sutheerasak (2019)** [59] | Diesel, ethanol | 43kW, three cylinders, four-stroke, DI, CR 17.2:1, John Deere 3029DF150 | Producer gas (charcoal) | CO and HC increase with the increases in syngas flowrate. |
| **Tripathi et al. (2020)** [60] | Diesel | Variable compression CI engine, 4 stroke, DI, Water-cooled | Producer gas (cotton stalk) | Reduction in NOx emission, however, higher noise and 60% reduction in GHG emission. |
| **Halewadimath et al. (2020)** [61] | Diesel | TV1 type, single cylinder, 4-stroke, CI engine | Producer gas (wood) | BTE increases with the addition of hydrogen, HC and CO reduce. However, NOx emission increases. |
| **Sutheerasak (2020)** [62] | Diesel | Diesel-engine Model Mitsuki, single cylinder, 3.5 kW @ 1800 rpm, CR 17.5:1 | Producer gas (wood pellet) | Lower NOx emission, maximum 23% diesel fuel-saving, higher soot. |
| **Sharma and Kausal (2021)** [63] | Diesel | A 4-stroke, water-cooled, single cylinder, CI engine, CRDI | Producer gas (pistachio shell) | CO and HC higher than diesel mode, lower NOx emission. |

## 2.3 Review of dual fuel CI engine using biogas-diesel

The gaseous fuel generated by anaerobic digester from various waste is known as biogas. To utilize biogas for power generation needs a technique known as a dual fuel engine.

However, limitation to the performance of a dual fuel engine generates several issues that need to be investigated. On change of engine parameters like amount of pilot fuel, load, speed, compression ratio, injection pressure and gaseous fuel compositions.

The performance and emissions of the dual fuel engine are affected by the said parameters. Many researchers have evaluated the effect of these parameters on the performance and emission characteristics of the dual fuel engine using biogas. The main constituents of the biogas are $CH_4$ and $CO_2$.

### 2.3.1 Effect of load on the performance parameters of dual fuel engine

Bedoya et al. [64] have evaluated the influence of pilot fuel mass and mixing system on dual fuel CI engine using biogas-diesel. The 100% diesel fuel substitution was achieved by using palm oil biodiesel and biogas. Yoon and Lee [65] have reported the ignition delays of dual fuel engine longer than diesel mode. The BSFC was improved at load above 80% load. Cacua et al. [66] have studied to explore the effect of air enriched with oxygen in a dual fuel mode. The results showed improvement in combustion with the addition of oxygen. However, the addition of oxygen in the fuel mixture decreased the adverse effect of $CO_2$ in combustion. Barik and Murugan [67] found that advancing in pilot fuel injection timing of 26°CA before TDC showed optimal results in performance, combustion and emissions. The BSFC was reported 25% higher in dual fuel mode than diesel mode at full load. Maximum BTE was obtained by 28.1% at full load, which was higher compared to other timing set trials. The NOx emissions were 33.0% higher than the standard 23°CA before TDC but 16.0% lower than diesel fuel mode.

Ga et al. [68] reported that dual fuel engine operated with biogas-diesel having 80% methane, produced brake power close to 100% diesel fuel mode. The indicated thermal efficiency (ITE) of the dual fuel engine declined with the increase in engine speed. It was because of the incomplete combustion of gaseous fuel. Bora et al. [69] studied the optimization of injection timing and CRs of biogas-diesel dual fuel mode. The outcomes of the experimental study were reported optimum CR and IT of 18 and 29°CA before TDC respectively. The maximum BTE and diesel fuel savings were 25.44% and 82.67% respectively. The HC and CO emissions were lower at optimum parameters, whereas the

NOx, EGT and $CO_2$ emissions were higher than diesel fuel, as some results are given in Figure 2.6.

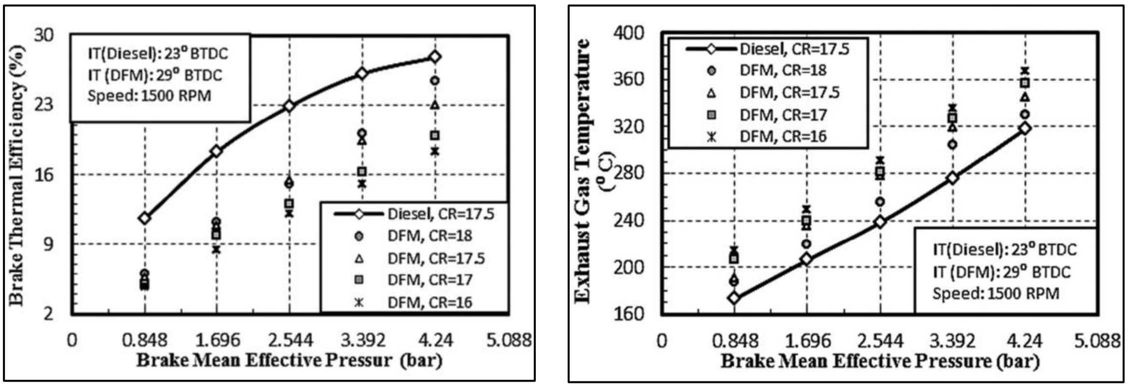



**Figure 2.6 Variation of BTE and EGT with BMEP at different CRs and injection pressure** [69]

Rahman and Ramesh [54] experimentally investigated that with the increase of biogas energy share, BTE improved due to better combustion. A reduction in smoke concentration was noted. The utilization of methane concentration (22%) results in better emissions. However, no significant change in performance parameters under dual fuel mode.

Ambarita [70] has reported that a dual fuel engine generates low brake power in comparison to diesel mode at the same operating conditions. The BTE of dual fuel mode was more in comparison to diesel mode at a low flow rate of biogas. However, it was decreased at a high flow rate of biogas. BTE of dual fuel engine was improved with a higher methane content in biogas under biogas-diesel mode. Diesel fuel replacement was proportional to the methane content in the biogas.

Lee et al. [71] studied the effect of $CO_2$ experimentally under dual fuel mode using biogas-diesel. The BTE of dual fuel engine decreased with the increase of $CO_2$ content in biogas. This was because of the reduction in heat released rate under dual fuel mode. The BTE improved with the rise of inlet gas temperature.

Kalsi and Subramanian [72] reported that the concentration of HC and CO increased with the rise of $CO_2$ percentage in biogas at a part and full load. However, NOx emission was reduced significantly. This was due to the reduction in-cylinder pressure and an increase in the dilution effect [72]. Feroskhan et al. [73] studied the effect of pre-heating of the biogas-air mixture under dual fuel mode operation improved BTE up to 5.06% at full load. However, it reduced volumetric efficiency by 6.05% and increased exhaust gas temperature up to 100°C. The maximum diesel fuel substitution was reported as 90.11% at the low speed of CI dual fuel engine.

### 2.3.2 Effect of load on the emission parameters of dual fuel engine

Yoon and Lee [65] reported that the use of biodiesel as pilot fuel under dual fuel mode EGT reduced marginally compared to single fuel mode. NOx emission reported lower under dual fuel operation. However, HC and CO emissions were significantly higher in dual fuel mode. Nathan et al. [74] reported that efficiencies of dual fuel engine were close to diesel fuel using diesel-biogas. The NOx emissions under dual fuel mode lower than diesel fuel mode. However, the HC emission level increased with the rise in the biogas energy share. Bora and Saha [69] have investigated that advancing the start of injection timing increased NOx and $CO_2$ emissions as shown in Figure 2.7. However, CO and HC pollutants improved as shown in Figure 2.8.

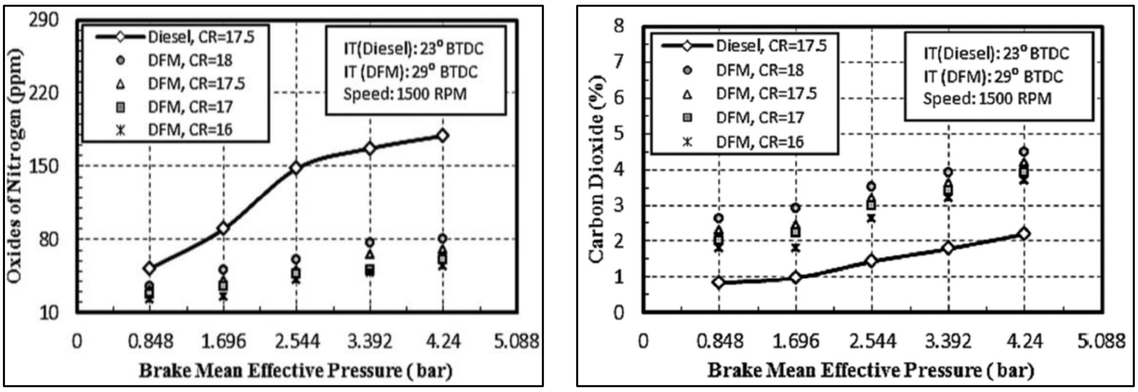



**Figure 2.7 Variation of NOx with BMEP at different CRs** [69]

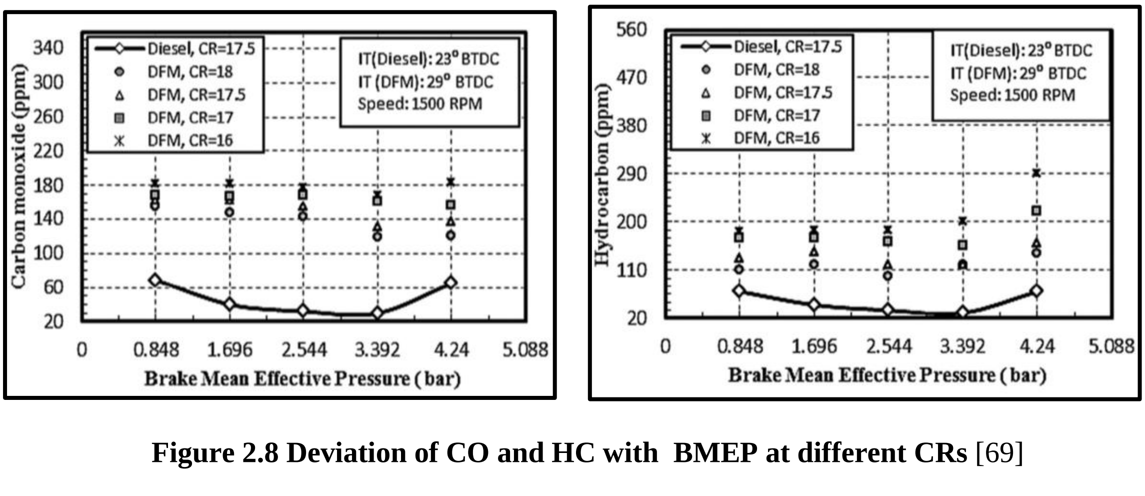



**Figure 2.8 Deviation of CO and HC with BMEP at different CRs** [69]

Barik and Murugan [75] have studied the production of biogas in floating drum digester and reported up to 73% methane content can be achieved. The optimum biogas flow rate was reported as 0.9 kg/hr under dual fuel mode. The BTE was reduced by 6.2% and NOx concentrations by 39.0%. The HC and CO concentrations were increased by 17 and 30% respectively as shown in Figure 2.9.

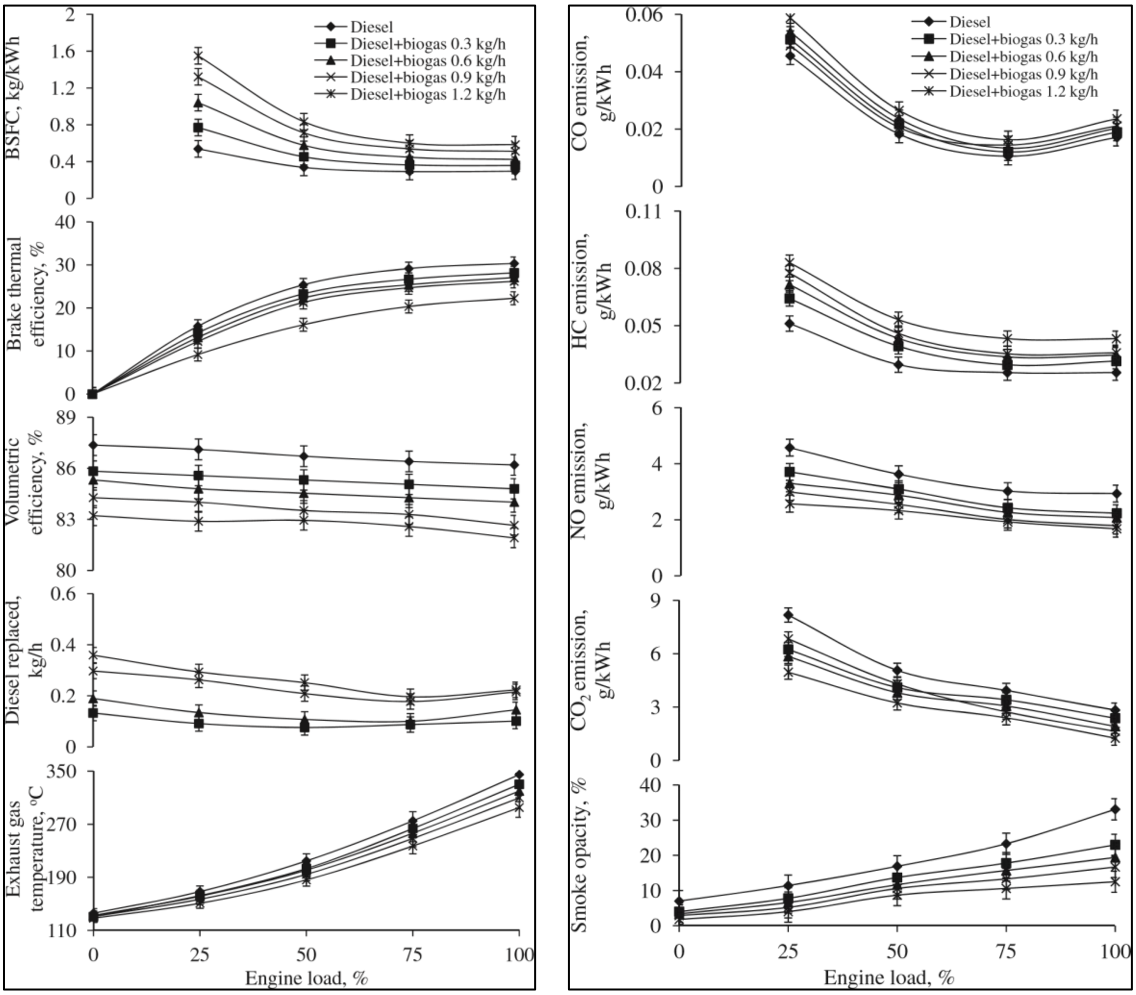



**Figure 2.9 Variation of performance and emissions parameters with load** [75]

### 2.3.3 Effect of compression ratio on performance parameters of dual fuel engine

The effect of compression ratio on the performance and emissions parameters of the dual fuel CI engine using biogas is the review in this section.

Bora et al. [76] reported that the BTE of a dual fuel engine increased with the increase in compression ratio from 16 to 18. The diesel fuel replacement improved by increasing the compression ratio as shown in Figure 2.10. The peak cylinder pressure increased while the

BSEC decreased with the increase in compression ratios. The effect of compression ratio on peak cylinder pressure and brake specific energy consumption are given in Figure 2.11.

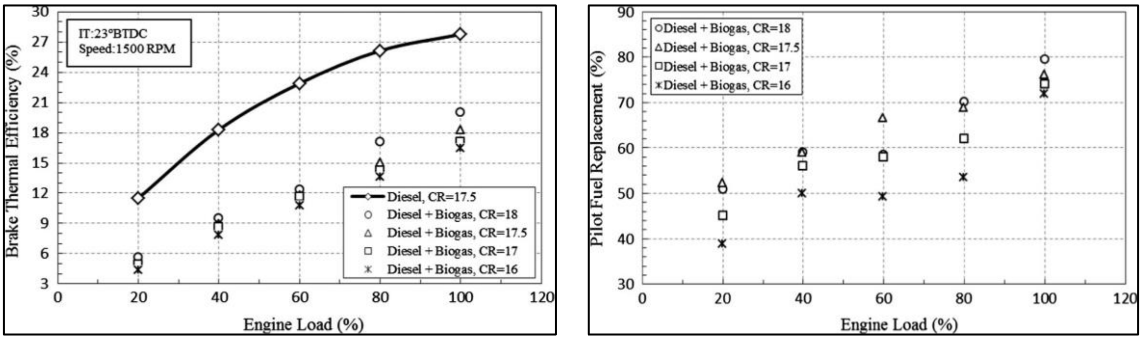



**Figure 2.10 Deviation of BTE and pilot fuel replacement at different CRs** [76]

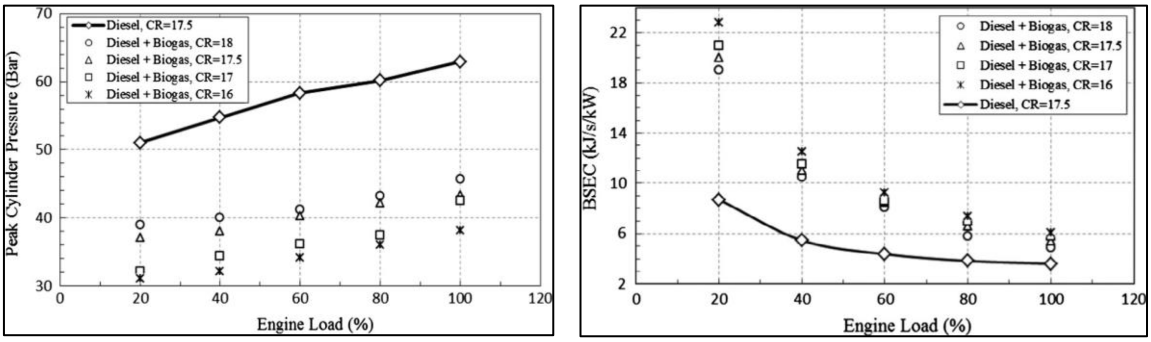



**Figure 2.11 Deviation of peak cylinder pressure and BSEC with load at different CRs** [76]

### 2.3.4 Effect of compression ratio on emissions parameters of dual fuel engine

Bora et al. [76] have investigated the emission characteristics of a dual fuel engine. CO and HC concentrations reduced with the increase of compression ratio as shown in Figure 2.12. NOx and $CO_2$ emissions increased with the rise in compression ratio as given in Figure 2.13 [76].

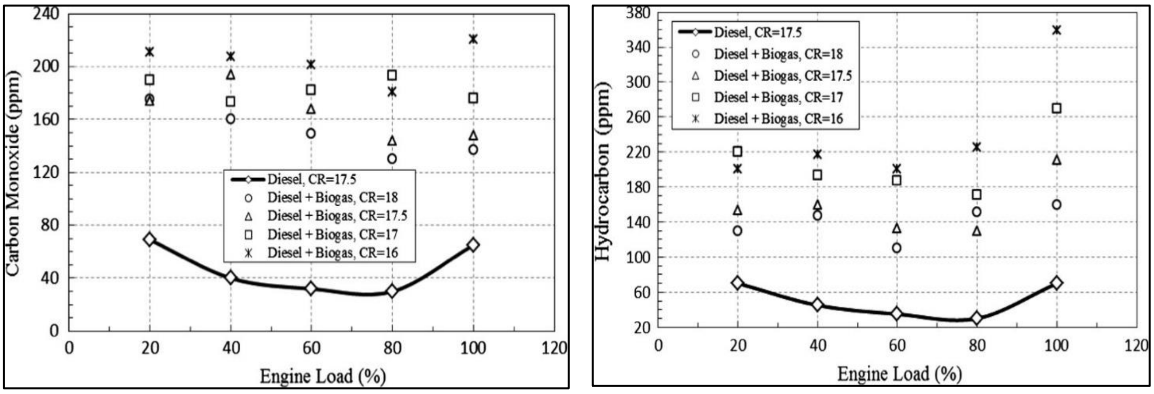



**Figure 2.12 Variation of CO and HC emissions at different CRs** [76]

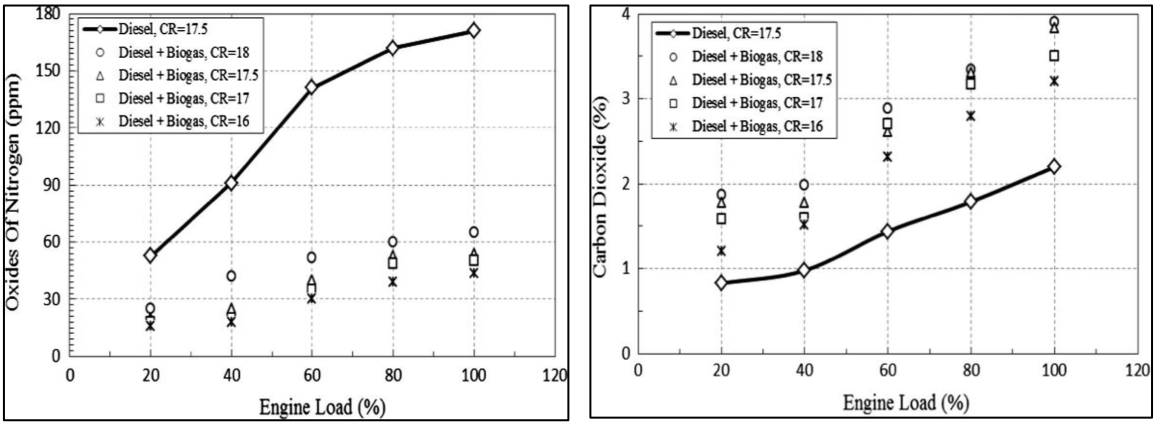



**Figure 2.13 Variation of NOx and $CO_2$ emissions at different CRs** [76]

### 2.3.5 Noise level of dual fuel engine

Elnajjar et al. [77] have studied the noise level under dual fuel mode using LPG, natural gas as the primary fuel. They reported a maximum noise level under LPG-diesel, natural gas-diesel, and diesel fuel mode by 88, 87, and 87 dB(A) respectively at a compression ratio of 22. The dual fuel engine noise level increased with the increase of pilot fuel quantity.

Narayan [78] has studied the analysis of noise levels emitted from a diesel engine. The combustion noise level depends on the engine loading and heat release rate. Selim et al. [79] studied the impact of steam injection on the performance, noise, and emission parameters of a dual fuel engine. The combustion noise increases with the addition of water in the combustion chamber.

Chaichan and Muneam [80] have studied the effect of operational parameters on noise emission of multi-cylinder dual fuel engine. They reported that the combustion noise level decreased with the rise in engine speed. However, combustion noise increased with the rise in pilot fuel quantity.

**Table 2.2 Summary of dual fuel engine used in the experimental study using biogas**

| **Researcher(s)** | **Pilot Fuel** | **Engine used for the test** | **Primary Fuel** | **Key Results** |
|---|---|---|---|---|
| **Yoon and Lee (2011)** [65] | Biodiesel | A direct injection diesel engine, 46 kW, constant speed, 4-stroke, 4-cylinder | Biogas (premixed) | Lower NOx emission, HC and CO emission higher. |

| Researcher(s) | Pilot Fuel | Engine used for the test | Primary Fuel | Key Results |
|---|---|---|---|---|
| **Makareviciene et al. (2013)** [81] | Diesel | 4-stroke 4-cylinder, DI, turbocharged (Bosch) | Biogas (M95%, M85% and M65%) | BTE and NOx emission reduction. |
| **Barik and Murugan (2014)** [75] | Diesel | A direct injection diesel engine, 4.4 kW, constant speed, 4-strokes, single cylinder | Biogas (Pongamia pinnata cake and cow dung) | Biogas flow of 0.9 kg/hr shows the optimal results, reduction in NO, $CO_2$, and smoke concentrations reported as 39%, 42%, and 49% respectively. |
| **Bora et al. (2014)** [76] | Diesel | 3.5 kW CI engine, single cylinder, 4-stroke, WC | Biogas (cow dung) | NOx and $CO_2$ emission increase with compression ratio. |
| **Ga et al. (2015)** [68] | Diesel | AVL single cylinder CI engine | Biogas (cow dung) | Maximum diesel fuel saving up to 79%. |
| **Bora et al. (2016)** [69] | Diesel | 3.5 kW CI engine, single cylinder, 4-stroke, WC | Biogas (cow dung) | BTE (25.44%) and fuel-saving (82.67%) were achieved at an injection time of 29° before TDC and at a CR of 18. |
| **Rahman and Ramesh (2017)** [54] | Diesel | Mahindra Maxximo NA, twin-cylinder engine, one cylinder being deactivated to make a single cylinder. | Biogas (cow dung) | Low levels of NOx and smoke emissions were reported. |
| **Ambarita (2017)** [70] | Diesel | Diesel Engine R175 AN, Horizontal Water-cooled, single-cylinder, 4 strokes | Biogas (cow dung) | Diesel saving varies from 15.3–87.5%. Maximum BTE was 15–18%. |
| **Kalsi and Subramanian (2017)** [72] | Biodiesel | 7.4 kW brake power, single cylinder, diesel engine | Biogas Simulated ($\mathbf{CO_2}$+ natural gas) | Maximum BTE observed by 31.4% with biogas (30% $\mathbf{CO_2}$). BTE decreased with the increase of $\mathbf{CO_2}$ percentage. |
| **Barik and Murugan (2017)** [67] | Biodiesel | 4.4 kW brake power 4-stroke, single cylinder, DI, diesel engine | Biogas (seed cake of Karanja and cow dung) | HC, smoke, and CO emissions were reduced by 18.2%, 2.1%, and 17.1% compared to standard injection timing at full load respectively. |

| Researcher(s) | Pilot Fuel | Engine used for the test | Primary Fuel | Key Results |
|---|---|---|---|---|
| **Choudhary et al. (2018)** [53] | Diesel | Kirloskar 3.5 kW, 4-stroke, single cylinder, DI, CI engine | Acetylene | The optimum injection pressure was 200 bar in terms of performance and emission. |
| **Feroskhan et al. (2018)** [73] | Diesel | 4-stroke, single cylinder, DI, CI engine | Biogas (methane and carbon dioxide) | BTE improved by 5% at full load |
| **Khatri and Khatri (2019)** [82] | Diesel | TV1 type, 4-stroke, single cylinder, DI, CI engine | Biogas (cow dung) and hydrogen | Reduction in HC and CO emission using hydrogen with biogas. |
| **Verma et al. (2019)** [83] | Diesel | TAF1, single cylinder, 4-stroke, DI, water-cooled | Biogas cylinder | CO and HC emission reduced at higher compression ratios. |
| **Singh and Layek (2019)** [84] | Diesel | Single cylinder, CI engine, 4 stroke, DI | Biogas (methane and carbon dioxide) | HC and CO emission higher under dual fuel mode, lower NOx emission reported. |
| **Oishi et al. (2019)** [85] | Biodiesel | 4-stroke, single cylinder, DI, CI engine | Simulated biogas (methane and carbon dioxide) | Achieved Higher BTE, Lower EGT and maximum diesel fuel replacement by 60.20%. |
| **Ahmed et al. (2020)** [86] | Diesel | Six cylinders turbocharge CI engine | Biogas (pig manure and corn straw) | Use of up to 45% of $\mathbf{CO_2}$ in biogas can lower NOx, CO emission and EGT. |
| **Das et al. (2020)** [87] | Diesel | CI engine, Direct injection, water-cooled | Biogas (cow dung and methane and carbon dioxide) | Lower BTE achieved, higher HC and CO emission reported |
| **Bouguessa et al. (2020)**[88] | Diesel | Single cylinder, 4-Stroke, Air-cooled, direct injection, CI engine | Biogas (synthesized by $CH_4$ and $CO_2$) | Reduction in PM emission levels up to 95% under dual fuel mode than diesel mode |

## 2.4 Summary

The review of the earlier work was done mostly on the simulated (mixture of CO and $H_2$) syngas and biogas. Most of the research have been done on dual fuel engine under gaseous fuel such as LPG, natural gas, and hydrogen. In addition, the availability of a large amount of agricultural residue in rural areas necessary to utilized in power generation and the massive amount of kitchen waste (vegetable) available in urban areas forced further study on the dual fuel operation using biogas. Hence, the present work is focused on performing a systematic experimental study on VCR diesel engine under dual fuel mode using gaseous fuel generated from biomass residue. To complete the research objectives, the CI diesel engine with VCR has been chosen for the experimental work. The details of the laboratory set-up required for the dual fuel operations are described in the following Chapter.

Chapter 3

# Experimental set-up and procedures

Dual fuel engines are used for gaseous fuel utilization generated from various types of biomass residue. Biomass residue burnt in open-air generate emissions and contributes to environmental pollution. Researchers have suggested that dual fuel engine using gaseous fuels should be utilized to reduce environmental pollution. The present study decides to perform systematic experimental performance and emission characteristics using producer gas and biogas under a dual fuel VCR diesel engine. The conventional CI engine is employed in the experimentation after converting into a dual fuel engine. A dual fuel technology is in which the diesel injection system is retained for the diesel injection to ignite the air-gas mixture in the engine cylinder. Experiments were conducted on a single cylinder, 4-stroke variable compression ratio diesel engine at a constant speed. The engine set-up specifications, the measuring instruments, production of biogas, producer gas and injection pressure adjustment were discussed.

The diesel engine conversion methodology for dual fuel mode is presented in two parts. In the first part, producer gas was generated with the help of a downdraft gasifier. While in the second part, biogas produced from kitchen waste using an anaerobic digester.

This chapter is concluded with experiment repeatability, testing procedure for various performance parameters, and emission characteristics for both modes of engine operations.

## 3.1 Biomass residue utilization techniques

Biomass residue utilized in energy production depends on many factors. The choice of the conversion process depends on the type of biomass residue and its quantities available for conversion into energy. The biomass residue to suitable energy is utilized using two leading technologies: bio-chemical and thermos-chemical.

Now presently, the generation of power from biomass residue is higher than conventional fuel cost showed uncompetitive technology. However, the increased government pressure to reduce air pollution in all four metro cities and neighboring may change to accept the technology. Out of the biochemical conversions of biomass residue into energy two process is digestion for the generation of biogas and fermentation for production of ethanol. The thermochemical conversion process is four processes, i.e., gasification, pyrolysis, liquefaction and combustion. The two-processes used for gas generation i.e., gasification and anaerobic digestion are currently cost-effective options to generate gaseous fuel for stationary engine or power plants [89,90]. However, all the processes are technically able to generate gaseous fuel. The gasification process commercially viable due to its overall higher gas generation efficiency [91]. Anaerobic digestion has its advantages as a process to generate a gaseous fuel from high moisture content organic waste such as kitchen vegetable waste.

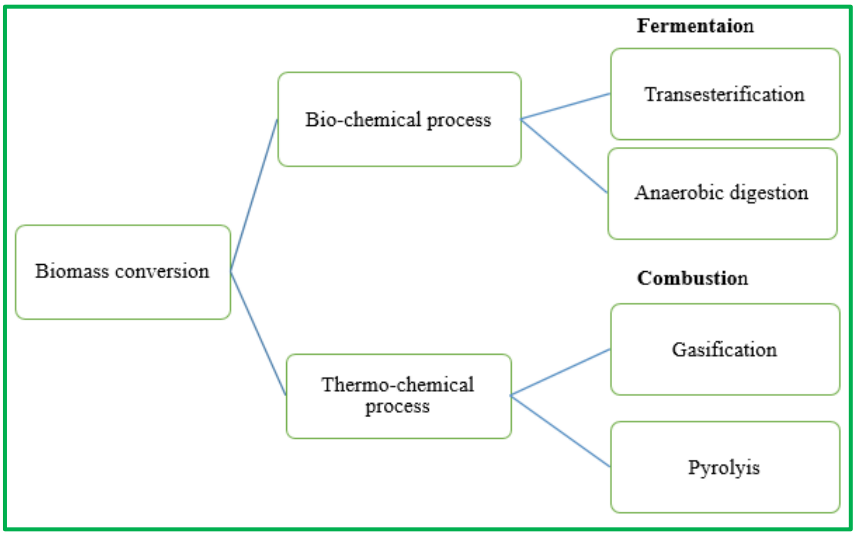



**Figure 3.1 Biomass conversion processes**

The following techniques efficiently use biomass from various agricultural waste and kitchen vegetable waste.

a) Biomass gasification
b) Anaerobic digestion

Keeping in view of the above processes, gasification and anaerobic digestion are more conventional and straightforward techniques for the generation of gaseous fuel at higher efficiency. The details of the methods are discussed in section 3.6.1 and 3.6.2.

## 3.2 Concept of dual fuel engine

Gaseous fuel is more suitable for alternative fuels because of its cleaner combustion. It produces very low NOx and $SO_x$. However, gaseous fuels are typically having a small cetane number and also have high autoignition temperature in comparison to standard diesel fuel. These properties of gaseous fuel make it difficult to use in conventional CI engine alone. Hence, the CI engine under dual fuel type engine plays an active part in the effective utilization of gaseous fuels as alternative engine fuels with a reduction of exhaust emissions [92]. The dual fuel CI engines are established on diesel technology. The gaseous fuel is used as the primary fuel and liquid fossil fuel as pilot fuel.

CI engine with a simple alteration was made to run under dual fuel mode. A gas-air mixture is sucked into the engine cylinder and compressed as a conventional CI engine. The compressed air-gas mixture does not burn because of the high autoignition temperature of gaseous fuel in the combustion chamber. Hence, it is burning through the use of pilot diesel fuel spray. Although a little quantity of diesel fuel use in pilot injection but its burning energy is much higher compared to spark ignition (SI) energy. Because of this, the ignition lag time shorter than the SI engine. Hence, it also solves many post-combustion issues. The main advantages of dual fuel CI engine are as following:

a) Gaseous fuel can be utilized with clean combustion.
b) Operational flexibility.
c) Improved thermal efficiency.
d) Reduction in exhaust such as NOx, $CO_{2,}$ and PM.

## 3.3 Combustion processes in dual fuel engine

The CI engine under diesel mode can be divided into four main stages, as shown in Figure 3.2 (a). The first stage; A–B: ignition delay period; B–C: premixed combustion stage; C–D: normal combustion and D–E: slow combustion. The fuel is injected at point A, and point B is the start of combustion. However, dual fuel mode is divided into five stages. The first stage is ignition delay AB, premixed pilot combustion BC, ignition delay primary fuel CD, primary fuel rapid combustion DE, and diffusion combustion stage EF. Pilot injection Ignition delay AB occurs longer than diesel fuel mode. It is because of the deficiency of oxygen present in the air-gas mixture. The low-pressure rise is seen BC as compared to diesel mode. It is also due to the deficiency of oxygen present in the air-gas mixture. Compared to diesel combustion, the pressure rise BC is low under dual fuel mode. It is because of the lesser amount of diesel fuel injected. The pressure decreased up to point CD until the start of the combustion of the gaseous fuel. The pilot fuel has started the flame propagation; as a result, stage DE is very unstable. The pressure rise at this stage does not affect any engine operational problem. Meanwhile, it helps in increasing cylinder volume. At point E diffusion starts, and point F is the end of rapid pressure rise. The process of pressure rise continues in the expansion stroke. The reason for this is the slower burning rate of gaseous fuel.  During this period, some air-gas mixture may escape from the combustion chamber. The length of the ignition delay decides the success of this stage. The P-θ diagram of a dual fuel engine is shown in Figure 3.2 (c).

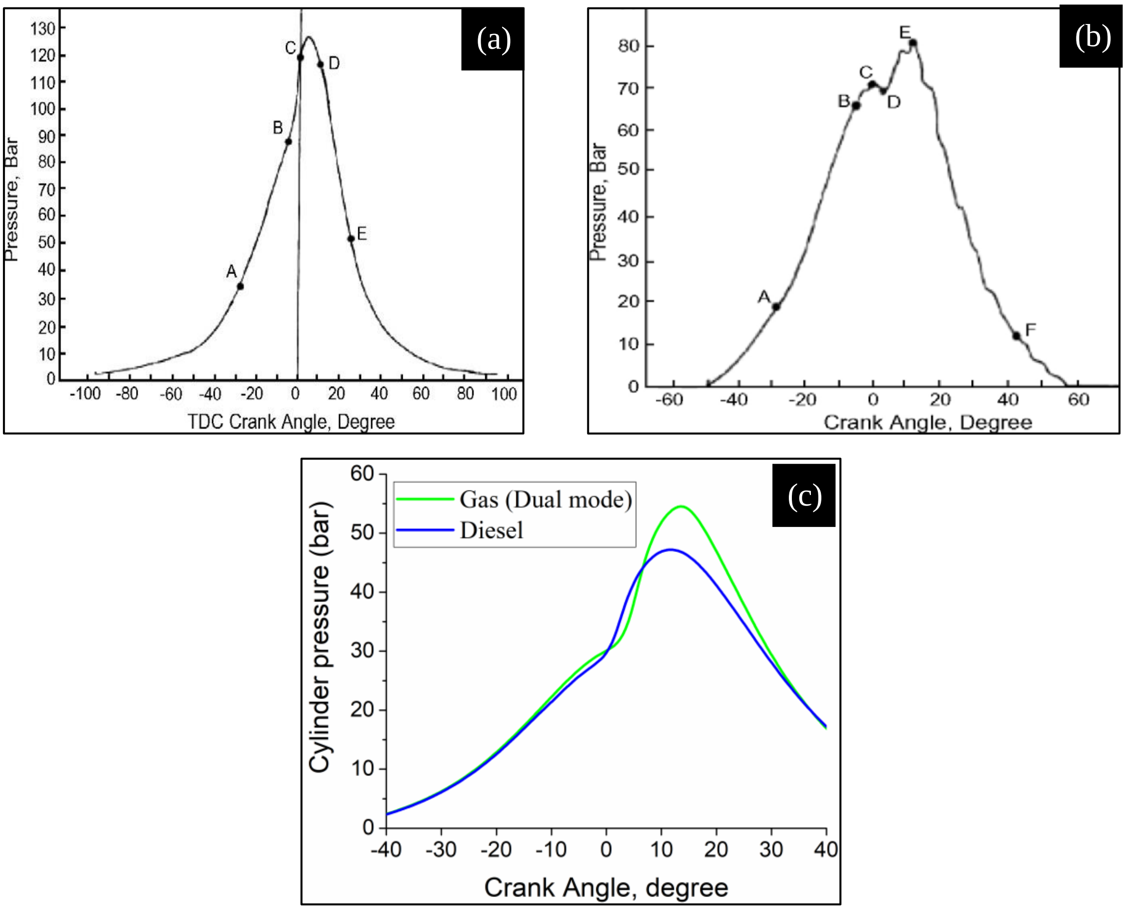



**Figure 3.2 Combustion processes in (a) diesel engine, (b) dual fuel engine** [93]**, and (c) P-θ diagram of a dual fuel engine**

## 3.4 Experimental set-up for producer gas-diesel mode

The experimental set-up consists of a gasifier unit, water pump, gas cooling unit, scrubber, gas filter, bag filter, gas control valves, manometer, eddy current dynamometer with measuring unit, and diesel engine. The flow chart for experimentation is shown in Figure

3.3. A single-cylinder water-cooled constant speed four-stroke variable compression ratio diesel engine (VCR) was used in the present study is shown in Figure 3.4.

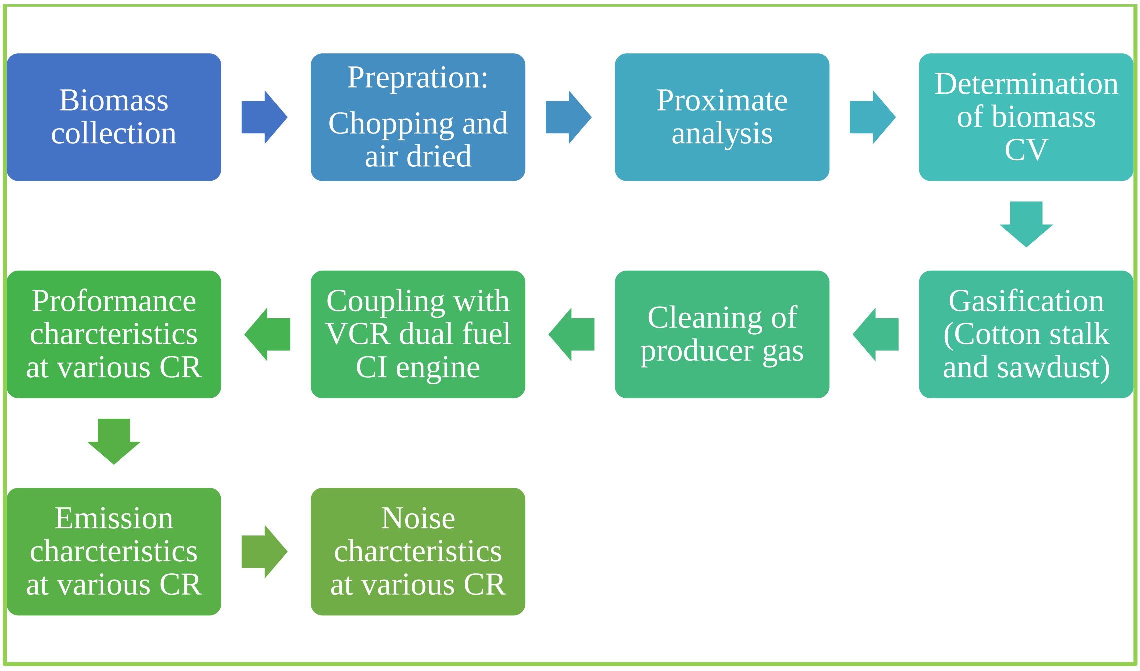



**Figure 3.3 Flow chart for experimentation for producer gas-diesel**

The detailed specifications of the engine are shown in Table 3.1 and Annexure-B. An electrically operated dynamometer (eddy current) was directly coupled to the engine using a propeller shaft. A digital controller unit measured the temperature, load, and pressure. Downdraft gasifier was used for the generation of producer gas from biomass residue (cotton stalks and sawdust). The detailed specifications of the gasifier unit is shown in Table 3.3. The producer gas was collected from the bottom of the gasifier. Now hot producer gas was allowed to pass through a water scrubber for cleaning and cooling. After cooling, the temperature of producer gas was lowered to atmospheric temperature. The scrubber provides cooling to the producer gas and it removes water-soluble gases ($H_2S$ and $SO_2$). The water scrubber also condenses the impurities like tar, soot particles from producer gas. The schematic of the experimental set-up is shown in Figure 3.5. The biomass residue was fed to the gasifier through its top opening.

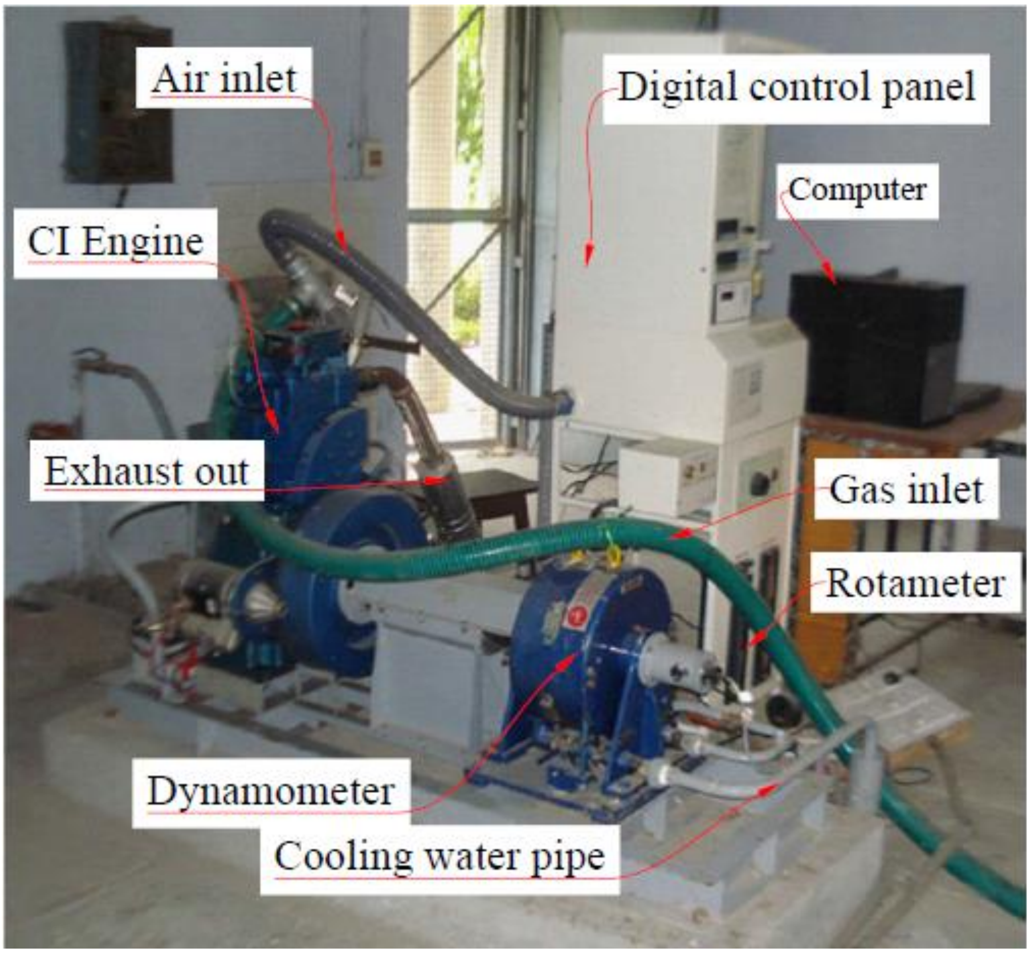
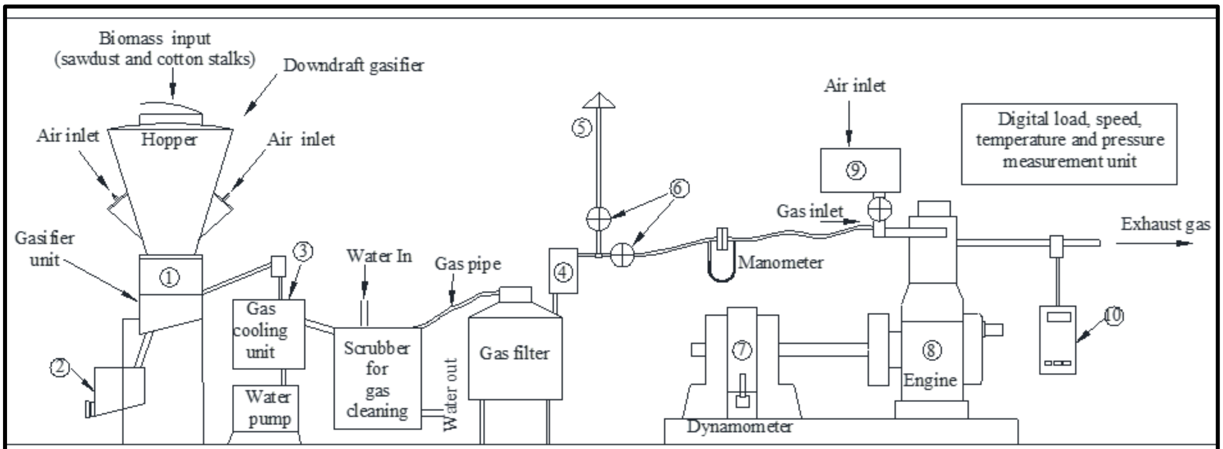



**Figure 3.4 Dual fuel engine set-up**

**Table 3.1 Engine specifications**

| Item | Description |
|---|---|
| Diesel Engine | Variable compression ratio (Kriloskar Make) |
| Rated power (kW) | 3.5 |
| Connecting rod length (mm) | 234 |
| Bore diameter and stroke length (mm) | 87.5 x 110 |
| Dynamometer for loading | Eddy current |
| Crank angle sensor | With 1 degree resolution |
| Data logger | National Instrument16 bit |




1. Combustion chamber
2. Ash box
3. Gas cooling unit
4. Bag filter
5. Flare
6. Valve
7. Eddy current dynamometer
8. VCR engine
9. Air box
10. Exhaust gas analyzer

**Figure 3.5. Schematic of the experimental set-up for producer gas-diesel**

## 3.5 Experimental set-up for biogas-diesel mode

The flow chart of biogas-diesel mode is shown in Figure 3.6. The schematic diagram of the biogas-diesel mode is shown in Figure 3.7.

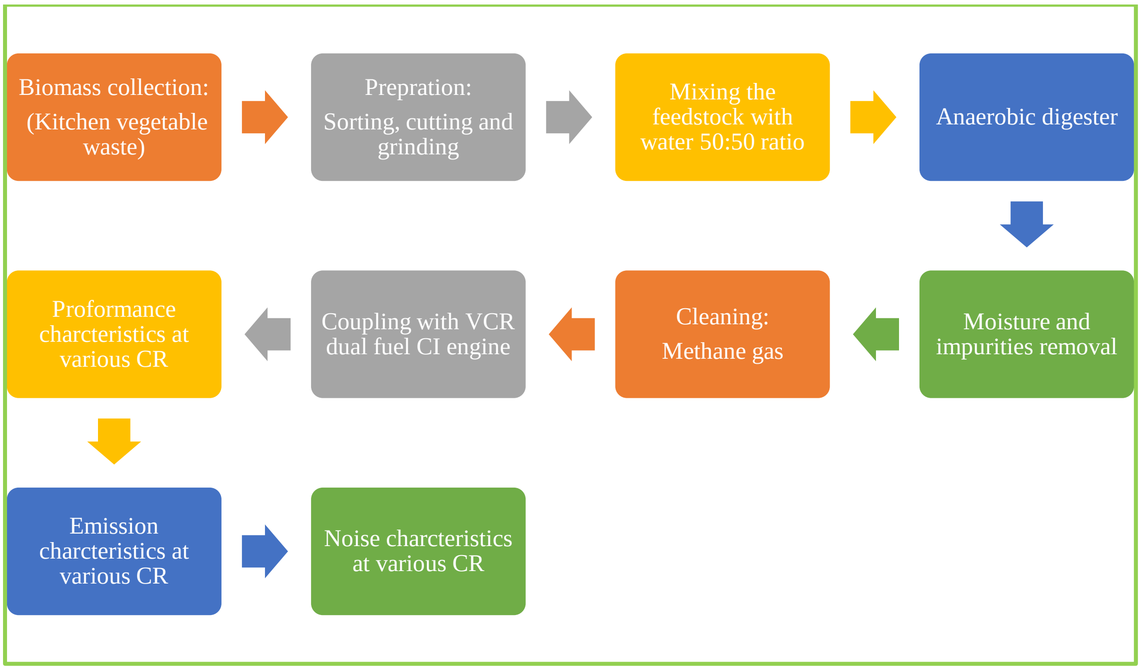



**Figure 3.6 Flow chart for experimentation for biogas-diesel mode**

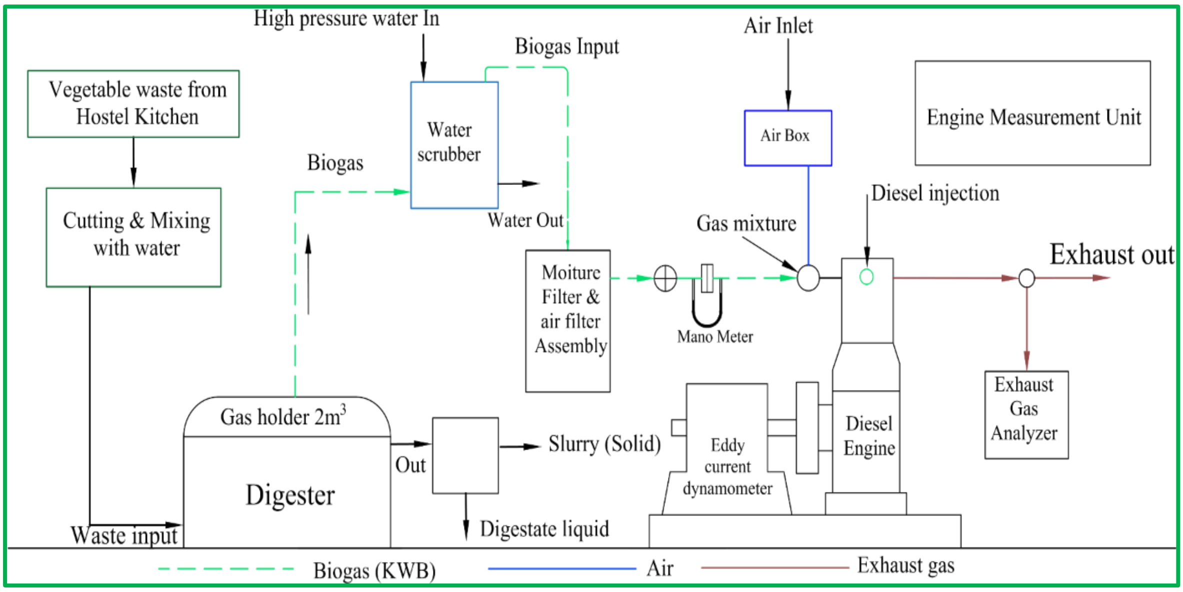



**Figure 3.7 Schematic of the experimental set-up for biogas-diesel mode**

## 3.6 Fuel used in experimentation

Two gaseous fuels, producer gas, and biogas were used in the present investigation. The gaseous fuel is fed to the engine through a suction port for the dual fuel operation. The following techniques were used to generate gaseous fuel.

### 3.6.1 Producer gas from biomass

The cotton stalk is mainly available in states like Maharashtra, Punjab, Karnataka, Haryana, and Rajasthan. Approximately 120 kg of the raw cotton stalk was collected from the state of Punjab (India) district Bathinda. The raw cotton stalk was air-dried for 20 days. The sawdust was also collected from the carpentry shop of Thapar Institute of Engineering and Technology, Patiala (India).

The dried cotton stalk was cut into small size as per the gasifier requirements. The size of the cotton stalk was kept in the range of 15–20 mm. The moisture content of the cotton stalk after chopping was maintained less than 20% as shown in Figure 3.8. The smaller size of the cotton stalk was helpful in the natural burning after being fed into the gasifier.


**Figure 3.8 Biomass used for gasifier (a, b) sawdust, and (c, d) cotton stalk**

### (I) Biomass gasification

This method of transforming biomass feedstock into a combustible gas in the gasifier is thermo-chemical. According to the flow of the gas and feedstock, a fixed bed gasifier can be classified as downdraft or updraft [94]. Biomass feedstock in the downdraft gasifier is fed from the top through a hopper. Biomass is subjected to drying, pyrolysis, oxidation, and reduction during the downward flow in the gasifier as shown in Figure 3.9.

In the downdraft gasifier, the producer gas is collected from the bottom side. The producer gas is comprised of gases (hydrogen gas, carbon monoxide gas, $CH_4$ and non-combustible gas like $CO_2$ and $N_2$). The quality of generated gas is influenced by the calorific value and contents of tar. The high-quality producer gas has high heating value and low tar contents. The quality of producer gas also depends on the biomass characteristics, gasifier design and process parameters. It can be used for heating purposes and fuel for IC engine in dual fuel mode after further cleaning.

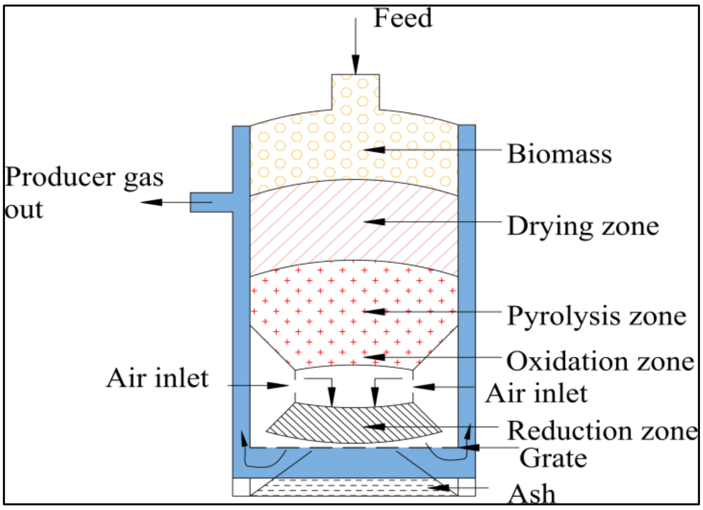



**Figure 3.9 Gasification in a downdraft gasifier**

The sequence process in the downdraft gasifier is drying, pyrolysis, oxidation and reduction as given in Figure 3.9. The drying process removes moisture from the biomass by taking heat from the oxidation zone. Not only drying, but the heat released from the oxidation process is utilized for the pyrolysis and reduction process. Combustible gases are formed in the pyrolysis process. The producer gas is generated through a reduction process, and gas leaves the gasifier through a gas outlet. The various processes in a downdraft gasifier are as following:

**(a) Drying process**

The temperature in the drying process is in the range of 120–200°C [95]. The moisture present in the biomass is converted to water vapor during the drying process. The quantity of moisture remove is in proportion to the water vapor formed. Generally, moisture present in the range of 5–35% is favorable for good quality gas production [95].

**(b) Pyrolysis process**

Combustible Biomass is decomposed into condensable gases, char, and tar in the lack of oxygen. The condensable gases are decomposed into non-condensable gases [96]. Non-condensable gases ($CO_2$ and CO) are cracked from the condensable gases in this process. The biomass pyrolysis process is represented by the reaction as shown in Equation 3.1 [96].

$$C_nH_mO_p(\text{Biomass}) \xrightarrow{\text{Heat}} \sum_{liq} C_xH_yO_z + \sum_{gas} C_aH_bO_c + H_2O + \text{Char} \quad \text{.........} \ (3.1)$$

### (c) Oxidation process

The heat produced in the oxidation process is utilized for drying, pyrolysis, and reduction process. The temperature of this zone is in the range of 800–1400°C [96]. Partial oxidation of char produces CO and heat as shown in Equation 3.2. While total oxidation produces $CO_2$ and more heat (Equation 3.3). The heat released during complete oxidation is three times greater than partial oxidation as shown in the following equations.

$$C + \frac{1}{2}O_2 \rightarrow CO \quad -111 \text{ kJ/mol} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3.2)$$

$$C + O_2 \rightarrow \mathbf{CO_2} \quad -394 \text{ kJ/mol} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3.3)$$

### (d) Reduction process

During the reduction process, all the main gasifier rection are occurred [96]. In the reduction process, combustible gases in the producer gas are formed through the following reactions.

Boudward reaction

$$C + CO_2 \rightarrow 2CO \quad +172 \text{ kJ/mol} \quad \ldots\ldots\ldots\ldots \quad (3.4)$$

Water-gas reaction

$$C + H_2O \rightarrow CO + H_2 \quad +131 \text{ kJ/mol} \quad \ldots\ldots\ldots\ldots \quad (3.5)$$

Water-gas shift reaction

$$CO + H_2O \rightarrow CO_2 + H_2 \quad -41.2 \text{ kJ/mol} \quad \ldots\ldots\ldots\ldots \quad (3.6)$$

Methane reaction

$$C + 2H_2 \rightarrow CH_4 \quad -74.8 \text{ kJ/mol} \quad \ldots\ldots\ldots\ldots \quad (3.7)$$

In the reduction process, endothermic (Bouduard and water-gas) (Equation 3.4, 3.5) and exothermic reaction (water shift and methane) take place (Equation 3.6, 3.7). The heat energy used by the endothermic reaction is 303 kJ/mol. While the exothermic reaction releases 116 kJ/mol heat.

### (II) Downdraft gasifier

In the present work, a downdraft gasifier was used for the generation of producer gas from biomass residue (sawdust and cotton stalks). The biomass residue was cut in the required size i.e. below 20 mm. After cutting and mixing with sawdust, the biomass material was supplied to the gasifier through a hopper.

The producer gas exit from the bottom of the gasifier. Hot producer gas was permitted to pass through a water scrubber for cleaning and cooling.

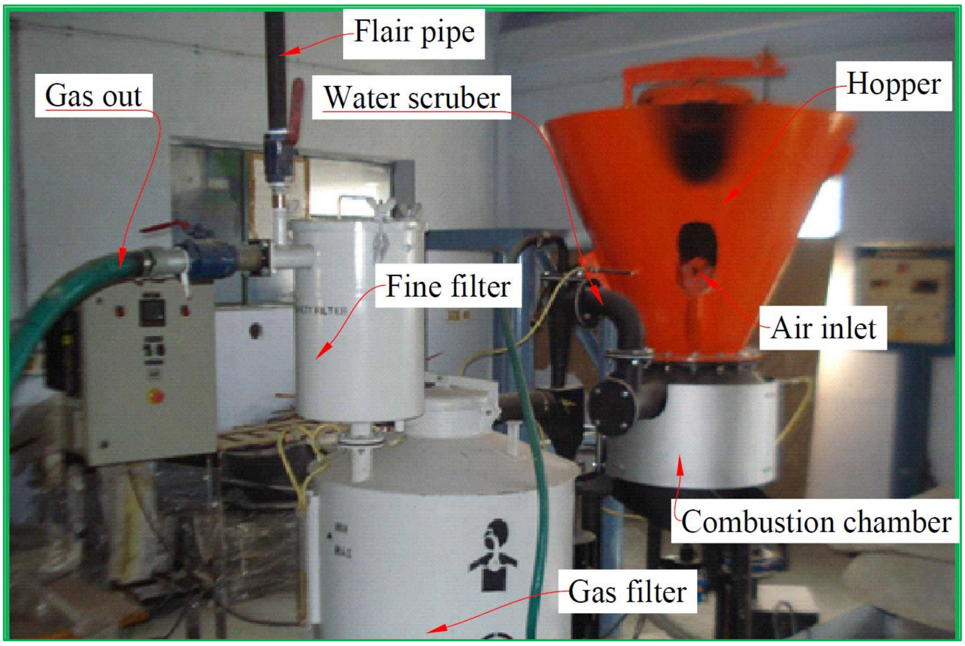



**Figure 3.10 Downdraft gasifier set-up.**

The scrubber provides cooling and removes water-soluble gases ($H_2S$ and $SO_2$) from the producer gas. The water scrubber condenses the impurities like tar, soot particles from the producer gas.

Further, the producer gas was cleaned using a fine filter and bag filter. The ultra-clean producer gas was sent to the IC engine at a temperature of about 30–50°C. The generated and cleaned generated gas is safe to be used in the IC engine under dual fuel mode.

**Table 3.2 Characterization of biomass fuel used (air-dried)**

| Biomass | Ash (%) | C (%) | H (%) | N (%) | O (%) | S (%) | Calorific value (MJ/kg) |
|---|---|---|---|---|---|---|---|
| Sawdust | 1.20 | 22.28 | 5.20 | 0.47 | 40.85 | 0 | 18.50 |
| Cotton stalks | 6.68 | 43.60 | 5.85 | 0 | 43.86 | 0.01 | 17.40 |

**Table 3.3 Technical specifications of the gasifier.**

| Item | Description |
|---|---|
| Model | WBG-10 in Ultra-clean gas Mode |
| Type of Gasifier | Downdraft |
| Gasification temperature (℃) | 1050-1100 |
| Fuel storage capacity (kg) | 100 |
| Gas flow rate ($Nm^3/h$) | 25 |
| Start-up | Through blower |
| Biomass Consumption (kg/h) | 8–9 |
| Gasification efficiency (%) | 66.14 |

**Table 3.4 Composition of producer gas.**

| Item | Composition (%) | Composition (average) (%) |
|---|---|---|
| CO | 16.1–20.1 | 18.1 |
| $CO_2$ | 12–16 | 14 |
| $N_2$ | 62.7–50.5 | 56.6 |
| $H_2$ | 7.1–9.3 | 8.2 |
| $CH_4$ | 2.1–4.1 | 3.1 |
| Calorific Value | | 4.5 $MJ/Nm^3$ |

### 3.6.2 Production of biogas

Biogas is generated during the process of anaerobic digestion. In this process, a biological breakdown of organic material in the oxygen-free environment. Biodegradable waste is transformed into a mixture of $CH_4$ and $CO_2$ with a minor quantity of hydrogen sulfide ($H_2S$), the trace of nitrogen, ammonia, hydrogen, and oxygen. The water vapor and dust particles are also present in it. The energy consumed by a biogas plant is approximately 25–30% of the energy demand [16].

In a developing country like India, biogas is the primary renewable-energy source of energy. The biogas can be utilized directly as fuel in heating applications. The other organic material in the digester is rich in nutrients to be used as a fertilizer.

Anaerobic digestion is a complex process that depends on many operating parameters like pH value, the temperature in the digester, C/N ratio etc. The process of generation of biogas is a linked process step, in which the primary material is continuously broken down into smaller units. The specific groups of microorganisms are involved in each distinct step. These organisms consecutively decompose the products of the previous steps. The detailed flow chart of anaerobic digestion is shown in Figure 3.11.

The main stages of the anaerobic digester to accomplish the production of methane are discussed below.

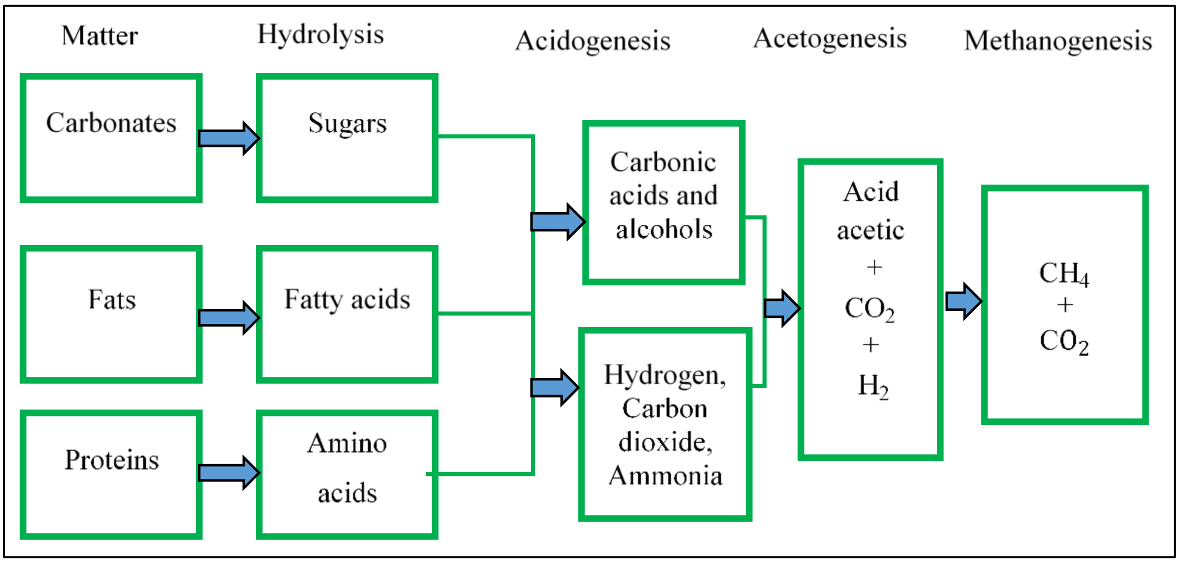



**Figure 3.11 Biochemical stages in methane production**

### (a) Hydrolysis process

Biomass generally contains abundant biological polymers, carbohydrates, fats, and proteins. These are cracked into smaller molecules like fatty acids, amino acids and simple sugars. It is the essential initial stage in anaerobic fermentation. The fermentative bacteria hydrolyze the organic compound into soluble molecules. The significant molecules, which are still comparatively large in the acidogenesis process, further cracked down so that they may be utilized to generate methane.

### (b) Acidogenesis process

It is the second step of anaerobic digestion in which acidogenic microorganisms further craked the organic products and biomass. Generated bacteria produce an acidic environment in the digestion reservoir results in generating hydrogen gas, carbon dioxide, ammonia, $H_2S$, littler volatile fatty acids, and organic acids. The primary acids generated are propionic acid, acetic acid, and butyric acid etc.

### (c) Acetogenesis process

The third process in the anaerobic digestion process is acetogenesis. In this process, the formation of acetate, a derived of acetic acid from carbon and energy sources, is performed. These bacteria catabolize numerous of the products produced in acidogenesis into acetic acid, hydrogen, and carbon dioxide. This process breaks down the biomass for methanogens in which methane is created by utilizing these bacteria.

### (d) Methanogenesis process

It is the fourth stage of anaerobic digestion in which methanogens generate methane from the end products and the intermediate products from hydrolysis and acidogenesis. The acetic acid and carbon dioxide are the two main products of the previous stages of anaerobic digestion to generate methane in methanogenesis.

$$CO_2+4H_2 \rightarrow CH_4+2H_2O \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3.8)}$$

$$CH_3COOH \rightarrow CH_4+CO_2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3.9)}$$

The primary process to generate methane in methanogenesis is acetic acid involvement. The carbon dioxide can be transformed into methane through a water reaction. The last stage leads to the generation of two leading products such as $CH_4$ and $CO_2$ of anaerobic digestion.

### (III) Anaerobic digester

Biogas was produced in the anaerobic digester of capacity 2m$^3$/day as shown in Figure 3.13. The specification of the digester is shown in Table 3.5.

**Table 3.5 Technical specification of the digester**

| Item | Description |
|---|---|
| Type of plant | Floating-drum type |
| Type of waste | Food/vegetable |
| Quantity of waste per day (kg) | 20–40 |
| Biogas yield per day (m$^3$) | 2 |
| Startup time (days) | 15–20 |


**Figure 3.12 Biomass used in digester (a, c) spoil vegetables (b, d) peeling of vegetable**

The vegetable waste from the university hostel was collected. The collected vegetable waste contained peelings and trimmings of cauliflower, coriander, cucumber, fenugreek leaf, onion, peas, potato, radish, tomato, and turnip. All constituents were mixed in equal proportion. The raw waste was cut and ground in a cutter for size reduction. A mixture of crushed vegetables and water (approximate 20 kg) was fed to the digester. For starting an anaerobic digester, 50% cow dung was added to start the generation of bacteria. The production of biogas takes about 30 days at a temperature of 35–45°C. After one month, biogas was exhausted in an open atmosphere and emptied the whole gas holder of the digester because a first-time gas holder has many impurities like air, dust, and moisture. The ground vegetable and water mixture was again fed to the disaster and wait for another 30 days for gas generation. The produced biogas comprises of $CH_4$ (56.80%), $CO_2$ (37.50%) as well as traces of $H_2S$ as given in Table 3.6. Now the biogas in the gasholder is ready to use for experiments.


**Figure 3.13 Anaerobic digester set-up**

Further, the generated biogas was cleaned by passing through a high-pressure water scrubber. In the scrubber, $CO_2$ is dissolved in that water. However, the concentration of $CH_4$ was enhanced [97]. This happened due to $CO_2$ has more excellent solubility in water than $CH_4$. The biogas leaving the scrubber has a higher methane concentration. The water scrubber also removes harmful impurities like sulfur dioxide ($SO_2$) and hydrogen sulfate ($H_2S$). Further, the moisture from the biogas was removed by using a moisture filter. After moisture removal, biogas was cleaned using a bag filter. The biogas was supplied to the engine through a control valve, which helps in controlling the flow of gas.

**Table 3.6 Composition of the biogas**

| Item | Composition (%) | Composition (average %) |
|---|---|---|
| $CH_4$ | 53.4—60.2 | 56.8 |
| $CO_2$ | 35.8—40.2 | 37.5 |
| $N_2$ | 1.8—5.2 | 3.5 |
| $H_2S$ | 1.7—2.7 | 2.2 |
| Calorific Value | -- | 20.39 MJ/Nm$^3$ |

## 3.7 Cooling and cleaning of producer gas

For trouble-free operation of an IC engine clean and cool gas is required. Gas cooling is primarily required because of increasing the density of producer gas before entering into the combustion process. The gas cooling method cleaned the gas impurities such as tar, and dust. The quality of output gas in the gasifier depends upon the design of the gasifier, load and biomass type. The hot producer gas from the gasifier is fed to a water scrubber for cooling and cleaning. In the scrubber, the producer gas is brought in direct contact with water using a pressurized water jet and reduced the gas temperature about 40°C. The water in the scrubber drains from the bottom and cold gas is fed to the fine filter unit.

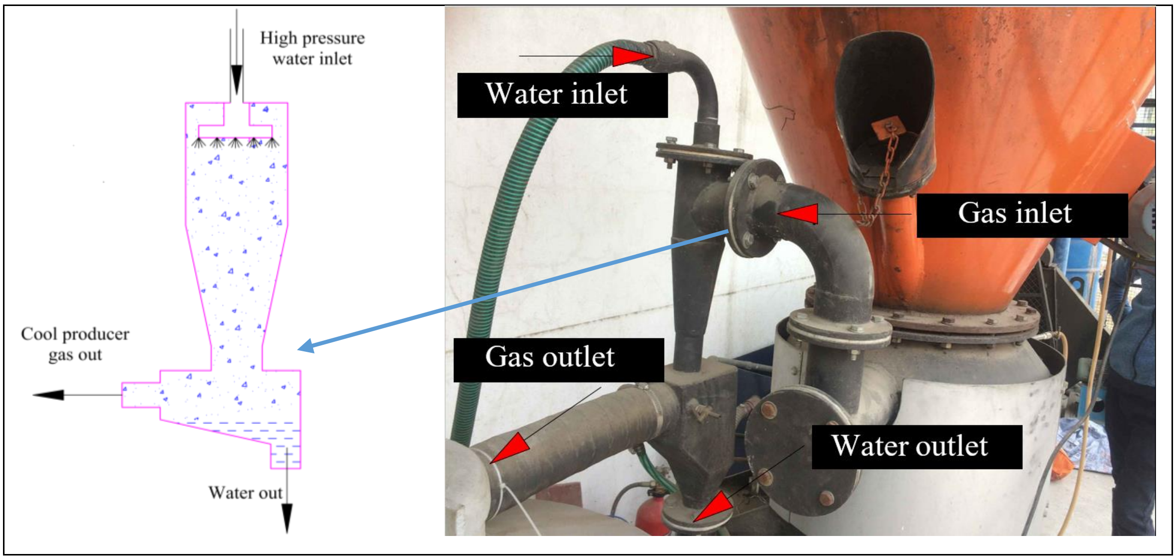



**Figure 3.14 Water scrubber**

The fine filter is filled with sawdust, which acts as filtering media. Fine dust particles from producer gas are trapped in the filtering media. After passing through a fine filter unit, producer gas is further cleaned using a safety filter. The ultra cleaning of producer gas is done in this filter.

## 3.8 Injection pressure variation of dual fuel engine

The injection pressure is varied by changing the fuel injector opening pressure as shown in Figure 3.15. The opening pressure for the injector was adjusted by adjusting the screw available on the injector. The required pressure was recorded with the help of a sensor installed on the fuel line. The clockwise rotation of the screw increased the injection opening pressure and losing the screw reduced the injection pressure. The three sets of injection pressure were used in the present investigations (240, 270, and 300 bar).

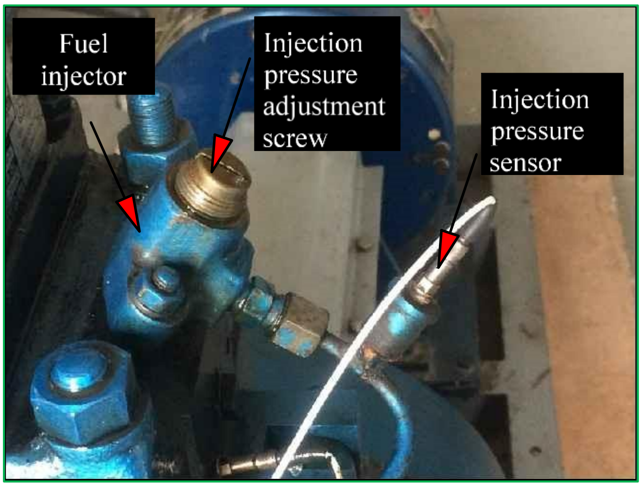



**Figure 3.15 Injection pressure variation**

## 3.9 Instruments used in the experimentation

The various measuring instruments after proper calibration were fitted on the experimental set-up for recording the various engine parameters. The main instruments are discussed below.

### (a) Engine performance measurement

The load on the engine was applied using an eddy current dynamometer through a knob. The performance parameters were measured from a computerized system for different loads and compression ratios setting. The various parameters measured are brake power, BSFC, BSEC, BTE, air, and fuel flow rate.

### (b) Air and gas flow measurement

Air supplied to the engine was measured with an orifice meter and manometer. The mass flow of air was calculated and recorded in the engine software system. During dual fuel mode, the producer gas and biogas supplied to a dual fuel engine were calculated with the help of a separate U-tube manometer installed on the gas pipeline.

### (c) P-θ measurement

For combustion parameters, the cylinder inside pressure was recorded with the help of a piezo sensor fixed on the engine block. The engine crank movement transfer into a digital signal, which was fed to a computer software system at a time interval of two degrees of crank angle. The crank angle was measured by using a 360-degree non-contact encoder.

**(d) Temperature measurement**

Type K thermocouples were installed on the engine test rig at various positions to measure temperatures such as inlet and outlet cooling water from the engine and also from the calorimeter. The temperature of ambient inlet air and exhaust gas were measured by the thermocouples.

**(e) Emission measurement**

The exhaust gas emissions were analyzed using five gas analyzer. From the engine exhaust pipe, a gas sample was fed to the gas analyzer during steady-state engine operation. The direct readings of the gas analyzer show straight forward readings of CO, $CO_2$, $NO_x$, SOx, and HC emissions. The specifications of the gas analyzer is shown in Table 3.7.

**Table 3.7 Specifications of the five gas analyzer**

| Parameter | Range | Resolution | Accuracy |
|---|---|---|---|
| $O_2$ (%) | 0 to 21 | 0.1% | ±2% of reading |
| CO (ppm) | 0 to 5000 | 1 ppm | ±10 ppm < 400 ppm |
| | | | ±5% of reading>400 ppm |
| $CO_2$ (%) Derived | 0 to 21 | 0.1% of reading | ±0.3% of reading |
| HC (ppm) | 0 to 2000 | 0.01% | ±10% of reading |
| NOx (ppm) | 0 to 5000 | 1 | ±5% of reading |
| Thermocouple (°C) (K type) | 0 to 600 | 1 | ±3% of reading |

**(f) Noise level measurement**

For measurement of noise level from a dual fuel engine, SC-310 sound level meter was used. As per standard noise level was measured at ear level and a distance of one meter. Three readings were taken and the average of the three readings was calculated.

## 3.10 Engine conversion methodology

The CI engine was converted into a dual fuel mode operation by inserting a tee and control valve in the suction pipe of the CI engine. The control valve regulated the flow of gaseous fuel in the intake manifold. The following two set-ups were used to perform the experiments on a dual fuel engine using producer gas and biogas.

### 3.10.1 Producer gas-diesel mode

During a dual fuel mode, it is necessary to supply a uniform air-gas mixture into the engine cylinder for combustion. An air-gas mixing device provided a homogenous air-gas mixture to the engine intake manifold. The schematic of dual fuel conversion is shown in Figure 3.5. Producer gas out from the gasifier was coupled to the engine using a gas mixing device. A valve can control the flow of gas.

### 3.10.2 Biogas-diesel mode

The existing diesel engine was altered to operate on dual fuel mode by using a mixture of generated biogas from an anaerobic digester and air in the engine cylinder. Biogas output from the digester is fed to the engine using a gas mixing device.

**Table 3.8 Experiment model (Diesel-A, Producer gas-B, Biogas-C)**

| CR ratio | Injection pressure, bar | Fuel used |
|---|---|---|
| 12 | 270 | A |
| 12 | 270 | A and B |
| 12 | 270 | A and C |
| 14 | 270 | A |
| 14 | 270 | A and B |
| 14 | 270 | A and C |
| 16 | 270 | A |
| 16 | 270 | A and B |
| 16 | 270 | A and C |
| 18 | 270 | A |
| 18 | 270 | A and B |
| 18 | 270 | A and C |
| 18 | 240 | A |
| 18 | 240 | A and B |
| 18 | 240 | A and C |
| 18 | 300 | A |
| 18 | 300 | A and B |
| 18 | 300 | A and C |

A valve can control the flow of gas and gas flow was measured using a u-tube manometer. The schematic of dual fuel biogas-diesel conversion is given in Figure 3.7.

## 3.11 Experimental test procedures

The test procedure for diesel fuel alone, producer gas-diesel and biogas-diesel dual fuel modes are described below. The experiment model is given in Table 3.8.

### 3.11.1 Diesel engine single fuel test

For results comparison and to ensure the reliability of the observations, baseline performance, and emission tests were conducted with the engine running on 100% diesel fuel. The experimental investigations were carried out at different brake power (BP) (0.8, 1.6, 2.4, 3.2, and 4.0 kW) and different compression ratios (12, 14, 16, and 18) at 1500 rpm. First, the engine was warmed up under no-load condition. The flow of water to the engine cooling system was adjusted at 310 LPH and 85 LPH for the calorimeter. Once the engine warms up the temperature of the different regions were stable. The observations from the engine test panel were converted to engineering units. The engine performance data were shown in the form of Brake power, BSEC, BSFC, BTE, air-fuel flow etc. The engine PCP and P-θ data were noted down for each test. Five gas analyzer was used to analyze emission parameters. The experimental investigations process was repeated for (0.8, 1.6, 2.4, 3.2, and 4.0 kW) brake power and different compression ratios (12, 14, 16, and 18). The engine load was adjusted by using a loading knob to control an eddy current dynamometer and the applied load displayed on the engine control panel.

### 3.11.2 Producer gas-diesel fuel tests

The biomass residue (sawdust and cotton stalks) was used to generate the gas using a downdraft gasifier. The configuration of producer gas is given in Table 3.4. Generated gas from the gasifier is allowed to enter into a gas mixing device through a control valve. The mixture of air-producer gas was sucked into the suction port of the engine to utilize in dual fuel combustion. The volume of producer gas was allowed to dual fuel engine until the engine runs smoothly. It shows the maximum amount of producer gas flow for dual fuel mode at a given set of load conditions. Due to excessive energy entering into the engine cylinder, the engine speed increased from 1500 rpm. The amount of pilot diesel fuel was controlled by setting the liquid fuel cut off valve. Finally, the cut-off valve was fixed at a constant engine speed of 1500 rpm. It shows the engine run with the least diesel fuel and maximum energy from the producer gas. The water flow to the engine cooling and calorimeter was set as stated in the diesel test. Once the engine warms up and the temperature of the different regions was stable. Then all input parameters were set as described in a diesel engine test. Combustion and performance results were noted from the test panel. The emission observations are collected from the five gas analyzer.

### 3.11.3 Biogas-diesel dual fuel tests

For biogas diesel dual fuel operation, biogas was generated from kitchen waste collected from the university hostel kitchen. Generated kitchen waste biogas from an anaerobic digester gas holder was passed through a moisture removal unit. After moisture removal, the biogas was passed to a gas mixing device. The mixture of air-biogas was sucked into the engine manifold to operate in a dual fuel mode. The biogas was allowed to the engine cylinder until the engine signs a misfire. It shows the maximum flow of biogas for a dual fuel mode. During this operation, engine speed increased because of excessive energy from biogas taking part in the combustion. To maintain power and speed, pilot fuel quantity was adjusted by regulating the cut-off valve. The water flow to the engine cooling and calorimeter was set, as stated in the diesel test. Once the engine warms up and the temperature of the different regions was stable. Then all input parameters were set as described in a diesel engine test. The combustion and performance results were noted from the software test panel. For emissions parameters measurement five gas analyzer was used to record NOx, CO, $CO_2$ and HC emissions observations.

After the entire above observations were stored, the standard diesel fuel mode of the engine was restored by shutting off the gaseous fuel supply and altering the pilot fuel supply to a normal position. The experimental investigation procedure was repeated as per the experiment design as shown in Table 3.8.

## 3.12 Experiment repeatability

The performance and emission parameters were measured thrice as per experimental design in both modes of operation. An average of each operating parameter was calculated. The average readings were used for analysis purposes.

## 3.13 Analysis procedure

The formula and equations used in the various performance parameters calculation of both modes were illustrated in Annexure-A. The dependent variables calculated from these were compared and analyzed.

## 3.14 Uncertainty Analysis

Uncertainty is associated with an experimental set-up like speed, manometer, temperature, pressure, crank angle encoder, uncertainty in brake power and specific fuel consumption are shown in Table 3.9. The detailed calculations were given in Annexure-C [98,99].

**Table 3.9 Uncertainty errors for performance parameters**

| Performance parameter | Diesel mode error (%) | Dual fuel mode error (%) |
|---|---|---|
| BP (kW) | 0.7 | 0.7 |
| BMEP (bar) | 0.85 | 0.85 |
| BSEC (MJ/kWhr) | 1.5 | 1.5 |
| BTE (%) | 1.6 | 3.0 |
| Volumetric efficiency (%) | 0.7 | 0.7 |
| Air flow rate (kg/s) | 0.5 | 0.5 |
| Air-fuel ratio | 1 | 2.3 |
| Liquid fuel substitution (%) | -- | 1.4 |
| Net heat release rate (J/deg CA) | 2 | 2 |
| Noise level (dB(A)) | 0.5 | 0.5 |

## 3.15 Assumptions

To ensure the repeatability and the correctness of the experiment, keeping all the parameters the same, three trials of all the tests are taken.

- The volumetric efficiency of the engine is assumed constant with varying loading conditions in the engine.
- The flow of gaseous fuel is assumed to remain constant under varying loading conditions.
- There is no change in the mass of the working medium.
- There are no heat losses from the system to the surrounding.
- The working medium has constant specific heats throughout the cycle.

## 3.16 Summary

In this chapter, experiments and procedures followed to carry out diesel and dual fuel mode of operations by varying brake power at different injection pressure and compression ratios are discussed. The necessary equipments are designed, fabricated and installed on the conventional diesel engine to alter it into a dual fuel engine. The repeatability of experimental results, the procedure of both modes and uncertainty with collected experimental data for entire engine operations have been discussed.

# Chapter 4
# Results and discussion

A dual fuel engine is primarily used with a combination of fossil fuel and a gaseous fuel generated from biomass residue. In the present research, the performance and emission characteristics of a dual fuel engine were studied. The producer gas was generated from woody biomass (cotton stalk and sawdust) using a downdraft gasifier and the biogas was generated from kitchen vegetable waste using an anaerobic digester. Required modifications were done to a CI engine to perform in dual fuel mode. This chapter presents the results related to engine performance and emission parameters of a dual fuel engine fueled with producer gas and biogas under different compression ratios (CRs), injection pressure, and loading conditions. The performance of direct injection, 4-stroke, single-cylinder engine, under constant speed run and dual fuel mode was monitored. The observations of the CI engine using producer gas-diesel and biogas-diesel with varying compression ratio and injection pressure are also discussed. The performance and emission outcomes are also equated for the identical engine power output in both modes of engine operation. Further, emission characteristics such as HC, CO, $CO_2$, SOx and NOx are also evaluated. The present work aims to quantify the influence of CR and injection pressure of producer gas-diesel and biogas-diesel in dual fuel mode. This chapter discusses the results in two parts. In the first part, producer gas-diesel fuel mode is discussed and in the second section, biogas-diesel fuel mode is discussed. For comparison purposes, the CI engine was tested on 100% diesel fuel. The engine tests were conducted for the range, 0–4.0 kW brake power (no-load to 100%) for both modes of engine operation.

## 4.1 Combustion process in diesel engine under producer gas-diesel mode

Cylinder pressure vs. crank position engine data at different loads and compression ratios for ten working cycles can be used to study the progress of combustion. As per the first law of thermodynamics, the net heat release rate can be found using equation 4.1 [40].

$$\frac{dQs}{dt} = \frac{\gamma}{\gamma-1} p \frac{dv}{dt} + \frac{1}{\gamma-1} v \frac{dp}{dt} \qquad \text{................(4.1)}$$

Where dQs/dt is the heat release rate. $\gamma$ is the specific heat ratio (Cp/Cv). The range of $\gamma$ is 1.3–1.35 [40].

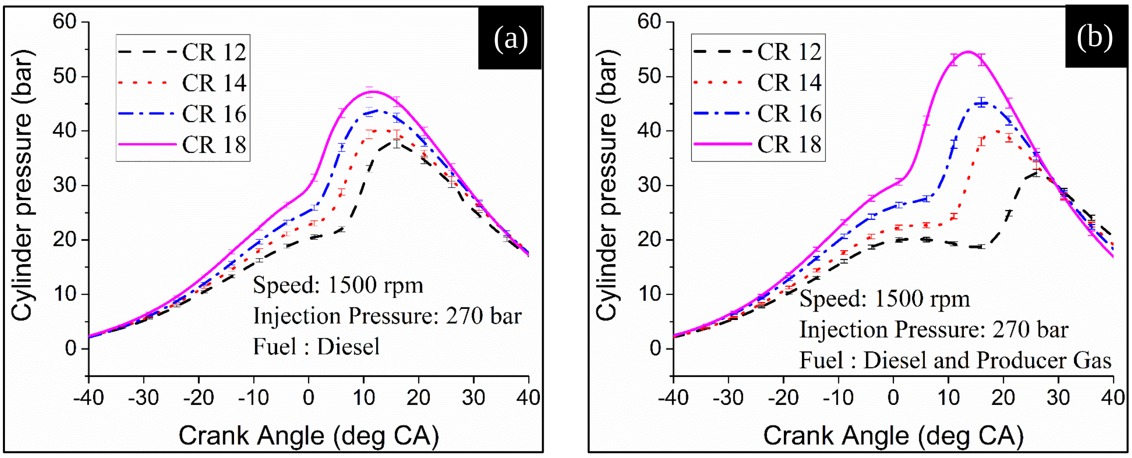



**Figure 4.1. Cylinder pressure at different CRs for the case of (a) diesel, and (b) producer gas-diesel mode**

The producer gas and diesel combustion process (i.e. dual mode) is more complicated than the diesel fuel combustion. During the compression stroke, the producer gas and air-fuel combination undergo pre-ignition. This pre-ignition reaction affects the ignition of diesel fuel. Due to ignition delay, there is an improvement in the fuel conversion efficiency as the combustion duration reduces. Higher ignition delay is observed in dual fuel mode than in diesel mode alone.

Sombatwong et al. [47] reported maximum pressure of 55.01 bar in diesel mode and 56.51 bar in dual fuel mode at a compression ratio of 18. These results are in agreement with those obtained in the present study. From the experimental results ignition delay of the pilot fuel decreases as the compression ratio increases. At higher compression ratios pressure and temperature in the engine cylinder increases. As load increases, the peak cylinder pressure increases for both modes because of the high mass of fuel supplied during this period. From the experimented results, the peak cylinder pressure for diesel mode was found to be 37.74, 40.23, 43.72, and 47.19 bar as compared to 32.26, 39.94, 45.29, and 54.49 bar for dual fuel mode at compression ratio 12, 14, 16, and 18 respectively. Due to the slow combustion of producer gas, peak cylinder pressure was shifted towards expansion stroke (Figure 4.1). It is also observed that the crank angle corresponding to peak cylinder pressure was 15º, 13º, 12º, and 12º after top dead center in diesel mode 26º, 18º, 16º, and 14º after top dead center in dual fuel mode at compression ratios of 12, 14, 16, and 18 respectively.

The net heat release rate was calculated by using cylinder pressure data. At different compression ratios, the net heat release rate vs crank angle is shown in Figure 4.2 (a, b) for diesel and dual fuel mode. It was observed that the net heat release rate decreased with an increase in compression ratio. As the engine compression ratio increases, with an increase in in-cylinder temperature, the heat transfer rate increases during combustion. The net heat release rate was found to be 43.15, 37.16, 35.89, and 32.76 J/degree CA for diesel mode and 60.65, 58.01, 52.96, and 46.93 J/deg CA for dual fuel mode at compression ratio 12, 14. 16 and 18 respectively. The ignition delay was found to be 8º, 5º, 3º, and 1º with regards to after top dead center for diesel mode and 20º, 12º, 9º, and 4º with regards after top dead center for dual fuel mode at compression ratio 12, 14, 16, and 18 respectively.

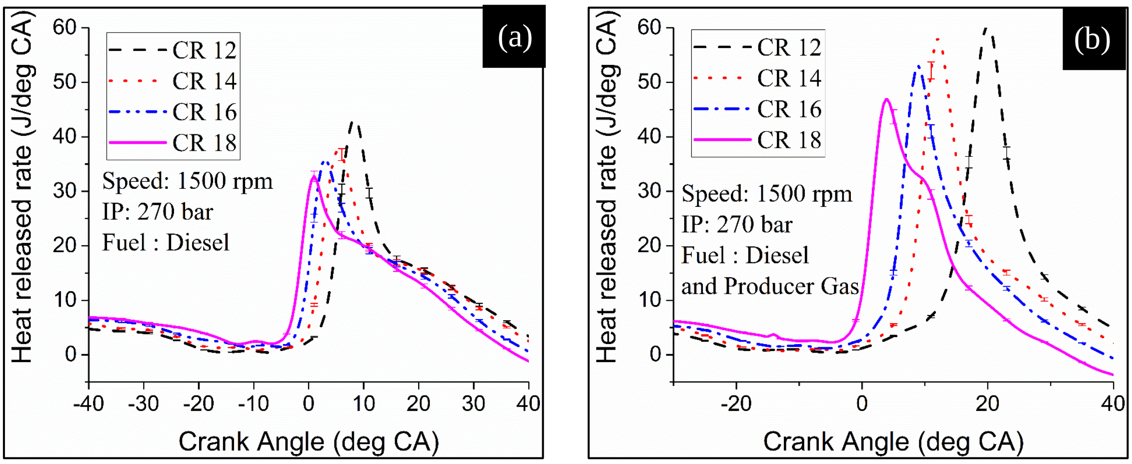



**Figure 4.2 Heat release rate at different CRs for the case of (a) diesel, and (b) producer gas-diesel mode**

Sombatwong et al. [47] have reported the maximum heat release rate of 35.02 J/degree CA for diesel mode and 40.03 J/degree CA for dual fuel mode at a compression ratio of 18. In the present study, the results are closer in diesel mode but higher in dual fuel mode. The reason may be that the net heat release rate depends upon the amount of fuel burnt in the cylinder.

## 4.2 Performance of dual fuel engine with producer gas-diesel mode

The performance of a dual fuel CI engine was evaluated experimentally using producer gas-diesel fuel. The performance results were compared with a single 100% diesel fuel mode operation.

### 4.2.1 Diesel fuel-saving under producer gas-diesel mode

The use of gaseous fuel along with diesel reduced the consumption of diesel fuel. It was observed that the consumption of gaseous fuel increases with an increase in the compression ratio. It may happen because of the higher temperature in the cylinder at a higher compression ratio, which helps in the combustion of gaseous fuel. The maximum diesel fuel saving was observed as 21.10, 33.03, 41.43, and 58.02% at compression ratio (CR) of 12, 14, 16, and 18 respectively.

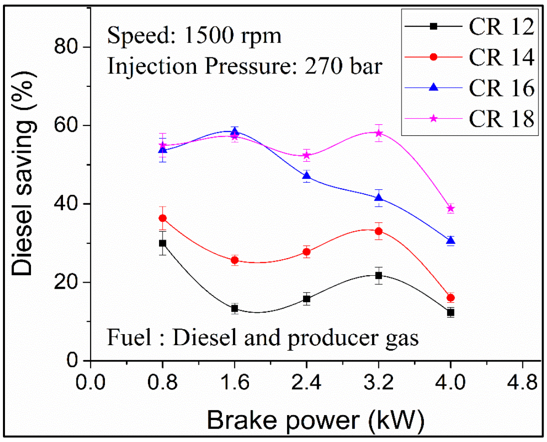



**Figure 4.3 Variation of diesel fuel saving with BP at different CR**

Maximum diesel fuel saving at compression ratios of 12 and 14 at brake power of 3.2 kW and for compression ratios of 16 and 18 at brake power of 1.6 kW are shown in Figure 4.3. At higher load conditions, diesel fuel saving was reduced because a richer mixture was required at higher loads.

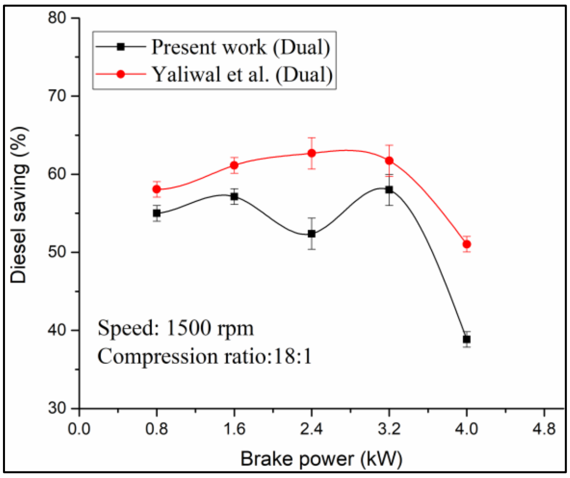



**Figure 4.4 Validation of Diesel fuel substitution at CR 18 with Yaliwal et al.** [50]

The results of the present work are in close agreement with the reported literature [42], [47], [50]. The diesel fuel saving at compression ratio 18 was validated with the work of Yaliwal et al. [50] as given in Figure 4.4. Similar trend of deviation of diesel fuel saving with load was observed. Similarly, for the compression ratio of 18, Sombatwong et al. [47] reported a maximum diesel fuel substitution of 64.21%. Similarly, Banapurmath and Tewari [42] and Yaliwal et al. [50] reported maximum diesel fuel substitution of 70.10% and 58.01% respectively. For the same conditions, a diesel fuel substitution of 64.30% was achieved in the present work.

### 4.2.2 Brake thermal efficiency (BTE) in producer gas-diesel mode

The variation of BTE of a dual fuel engine in conventional diesel mode and the dual fuel mode is shown in Figure 4.5 (a, b). It was observed that BTE increases with growth in load and compression ratio. It is due to the high temperature in the cylinder during higher load and compression ratio conditions. The higher temperature in the cylinder helps in improving combustion efficiency. It was also observed that in all the cases, BTE was lower in dual fuel mode. BTE was highest at 3.2 to 4.0 kW brake power in both modes of operation. Maximum BTE of 23.5, 24.52, 24.53, and 24.54% in diesel mode and 19.63, 21.13, 21.94, and 23.83% in dual fuel mode at a CR of 12, 14, 16, and 18 respectively. BTE improved as the compression ratio was increased from 12 to 18. However, BTE also depends on the share of gaseous energy in dual fuel mode. In the present work, BTE of 16.42, 12.43, 9.04, and 2.52% lower in dual fuel mode in comparison to diesel mode at compression ratios of 12, 14, 16, and 18 respectively was observed.

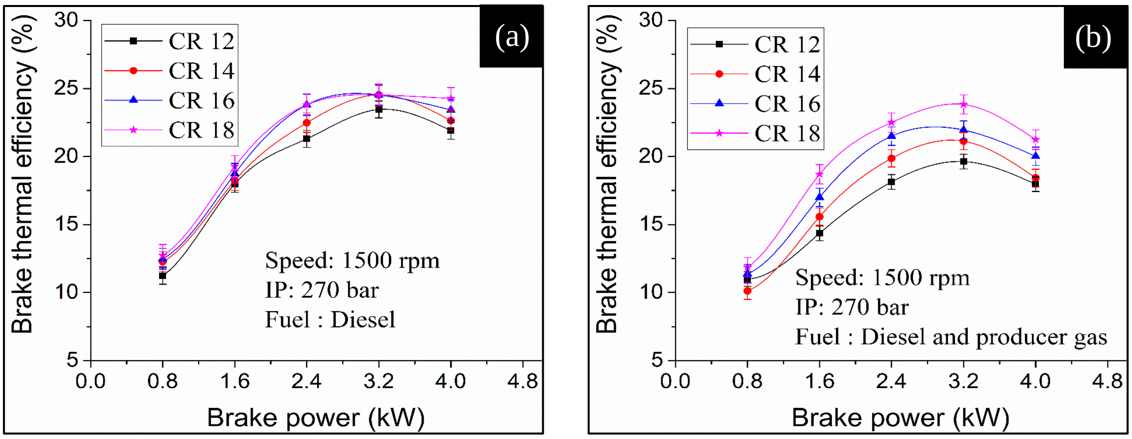



**Figure 4.5 Variation of BTE at different CRs for the case of (a) diesel, and (b) producer gas-diesel mode**

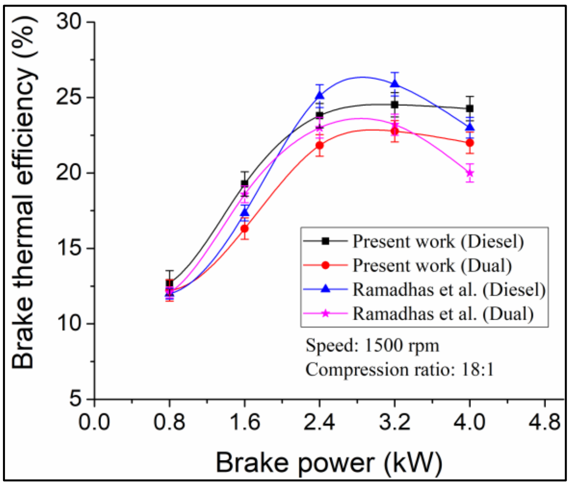



**Figure 4.6 Validation of BTE at compression ratio 18 with Ramadhas et al.**[22]

Although, BTE using producer gas-diesel in dual fuel mode was lesser than fossil diesel fuel mode. It is a renewable fuel and helps in reducing the environmental and natural resources. The BTE at compression ratio 18 was validated with the work of Ramadhas et al. [22] as given in Figure 4.6. Similar trend of variation of BTE with brake power was observed. Bora and Saha [100], Barik and Murugan [67], and Gnanamoorthi and Devaradjane [101] have also reported similar findings.

### 4.2.3 Brake specific fuel consumption (BSFC) in producer gas-diesel mode

BSFC in diesel mode and dual fuel mode depends on calorific value and the fuel consumption of diesel and producer gas (see Annexure-A equation A4). BSFC in dual fuel mode was 34.42 to 68.75% higher than diesel mode at 3.2 kW brake power. This was attributed to the lower efficiency in dual fuel mode. It was also observed that BSFC for the part-load was higher than the higher load. It can be seen from the graph that minimum BSFC was achieved at 2.4 to 3.2 kW brake power, as shown in Figure 4.7 (a, b). Maximum efficiency can be achieved at this load. It was also found that at 70 to 80% load, the BSFC value improved.

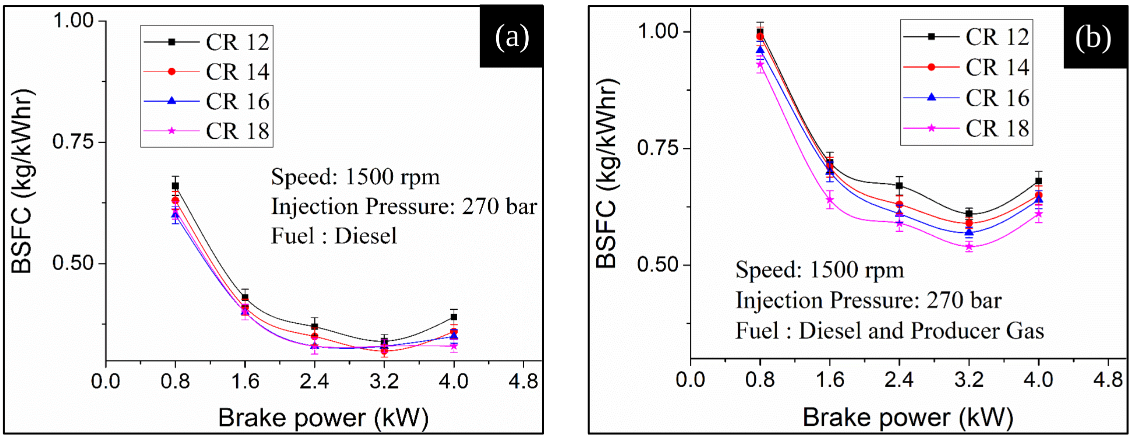



**Figure 4.7. BSFC of the dual fuel engine at different CRs for the case of (a) diesel, and (b) producer gas-diesel mode**

For the compression ratio of 18 and 80% load conditions, Sombatwong et al. [47] reported a maximum BSFC of 0.28 kg/kWhr in diesel mode and 0.41 kg/kWhr under dual fuel mode. The BSFC at CR 18 was validated with the work of Sharma and Kaushal [63] as shown in Figure 4.8. Similar trend in BSFC with load was observed in the literature. Shrivastava et al. [26] found BSFC of 0.3 and 0.41kg/kWhr in diesel mode and dual fuel mode respectively. For the identical parameters, the BSFC of 0.33 and 0.54 kg/kWhr in diesel mode and the dual mode was obtained in the present work respectively. BSFC in dual fuel mode is influenced by the gaseous fuel energy and calorific value of it.

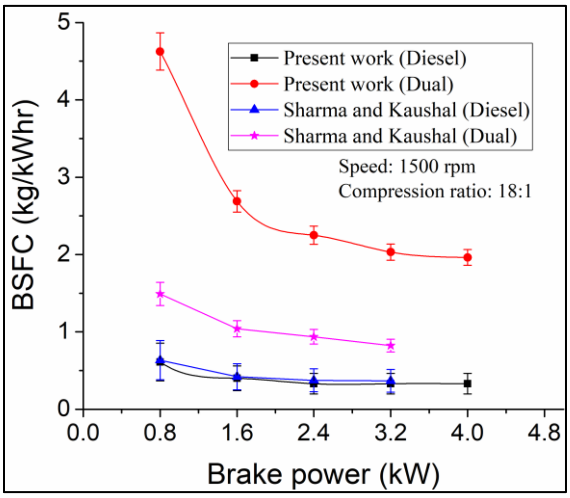



**Figure 4.8 Validation of BSFC at compression ratio 18 with Sharma and Kaushal** [63]

### 4.2.4 Brake specific energy consumption(BSEC) in producer gas-diesel mode

For comparing the performance of two types of fuels having diverse calorific values and density, BSEC was preferred.

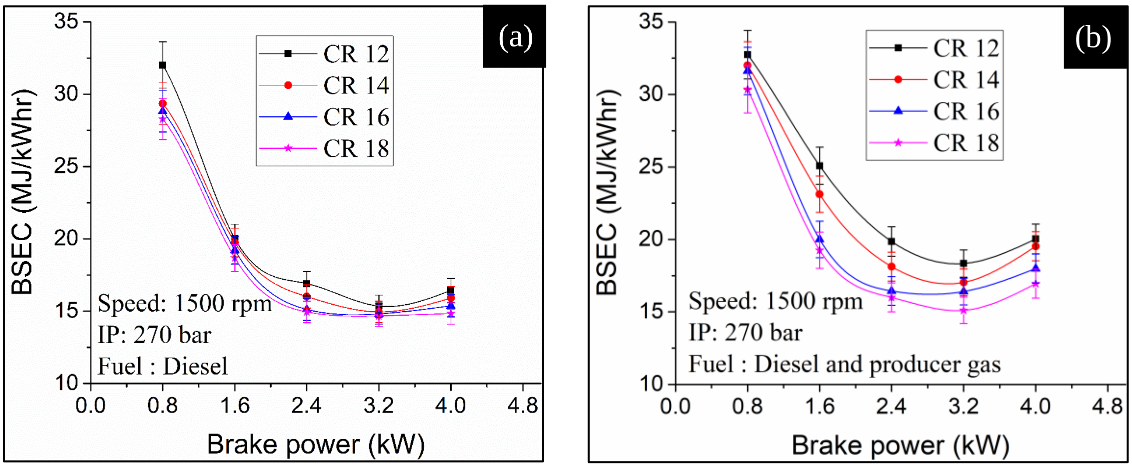



**Figure 4.9 BSEC of the dual fuel engine at different CRs for the case of (a) diesel, and (b) producer gas-diesel mode**

BSEC was calculated based on the calorific value of fuel and consumption of fuel used under BP of both diesel and producer gas. BSEC in dual fuel mode was observed to be higher than diesel mode in all cases of loading, as shown in Figure 4.9 (a, b). The increase in BSEC showed a reduction in BTE under dual fuel mode.

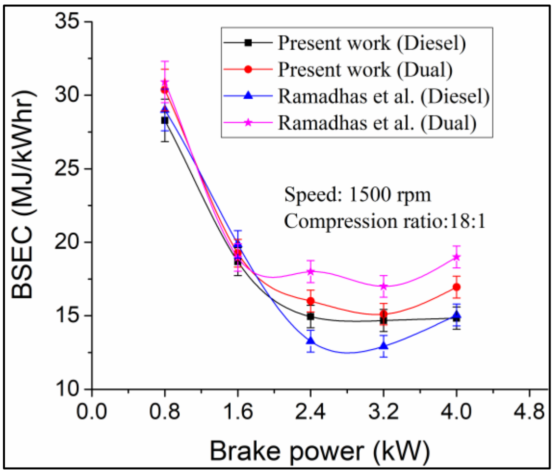



**Figure 4.10 Validation of BSEC at compression ratio 18 with Ramadhas et al.** [22]

The rise in the diesel fuel quantity resulted in a reduction in BSEC. Minimum BSEC of 15.35, 14.95, 14.81, and 14.68 MJ/kWhr in diesel mode and 18.33, 17.02, 16.40, and 15.10 MJ/kWhr under dual fuel mode at compression ratios of 12, 14, 16, and 18 respectively. BSEC improved with the increase in compression ratio. BSEC in dual fuel mode was 18.81, 13.83, 10.70, and 2.8% higher than diesel mode at compression ratios of 12, 14, 16, and 18 respectively. The BSEC at compression ratio 18 was validated with the work of Ramadhas et al. [22] as shown in Figure 4.10. Similar trend in BSEC value with load was observed in the literature.

### 4.2.5 Noise level of dual fuel engine using producer gas-diesel mode

The noise level in the diesel engine is higher due to high pressure and large molecules of diesel [102]. In the present work the noise level of dual fuel engine was measured in both modes of operation with varying brake power at different compression ratios as shown in Figure 4.11(a, b). It was observed that as the compression ratio and brake power increase, the noise level increases. This was because the pressure at higher compression ratio was high. The maximum noise level in dual fuel mode was observed as 88.5 dB(A) at 3.2 kW brake power and a compression ratio of 18. It was observed that the noise level in dual fuel mode was 1.26% higher than the diesel fuel mode at 4.0 kW brake power.

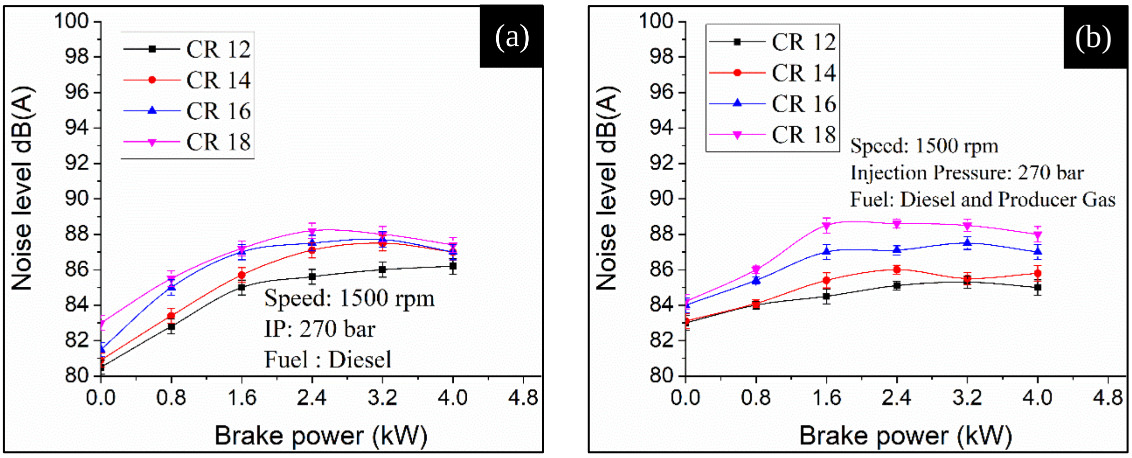



**Figure 4.11 Variation of the noise level of a dual fuel engine at different CRs for the case of (a) diesel, and (b) producer gas-diesel mode**

The noise level at the compression ratio of 18 was validated with the work of Tripathi et al. [60] and Singh et al. [23] as given in Figure 4.12 and Table 4.1. A similar trend in noise level was observed in the figure below. However, the noise levels are more than the permissible limits (75 dB(A)) set by Environment Protection Rules, 1986 [103]. Hence it is recommended that the noise level of a dual fuel engine can be reduced by incorporating a soundproof enclosure around the engine and adding a silencer to the exhaust system [104].

**Table 4.1 Validation of noise level** (dB(A)) **of dual fuel engine at CR 18 for producer gas-diesel**

| **Brake power (kW)** | **Present work** | **Singh et al.** [23] | **Tripathi et al.** [60] | **Noise level in present work in comparison to the literature** [23], [60] |
|---|---|---|---|---|
| No load | 84.20 | - | 86.09 | 2.24% lower |
| 0.8 | 86.11 | - | 88.79 | 3.11% lower |
| 1.6 | 88.12 | - | 90.82 | 3.06% lower |
| 2.4 | 89.20 | 96.90 | 92.77 | 4.00 and 8.63% lower |
| 3.2 | 88.53 | 102.15 | 92.21 | 4.16 and 15.38% lower |
| 4.0 | 88.12 | 100.40 | 90.94 | 3.20 and 13.94% lower |

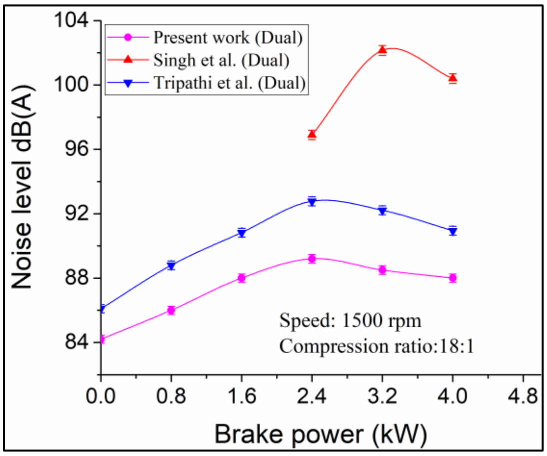



**Figure 4.12 Validation of noise level of dual fuel engine at CR 18 with Singh et al. [23] and Tripathi et al.** [60]

## 4.3 Exhaust emissions of dual fuel with producer gas-diesel mode

An emission analysis was carried out at different compression ratios and loads for the engine in diesel mode and dual fuel mode. The experimental study of pollutants such as CO, $CO_2$, NOx, SOx, and HC are discussed in this section.

The variation of NOx emission of diesel mode and dual fuel mode is shown in Figure 4.13 (a, b). The formation of NOx is favored by high oxygen concentration and high charge temperature [40]. It was observed that as the load and compression ratio increased NOx emission also increased. It happened because of more fuel supplied during higher load conditions and higher compression ratio, the combustion temperature and pressure increased in the engine cylinder. From the emission analysis, the maximum concentration of NOx emission in the exhaust gas in diesel mode was found to be 303 ppm. NOx emission from the engine in dual fuel mode varied between 8 to 130 ppm. It was observed that NOx emission in diesel mode was 35.31 to 56.05% higher than dual fuel mode at 3.2 kW brake power. The lower NOx emission in dual fuel mode was due to the less intense pre-mixed combustion, lower temperature due to the presence of a high amount of producer gas and the lower concentration of oxygen in the combustion chamber.

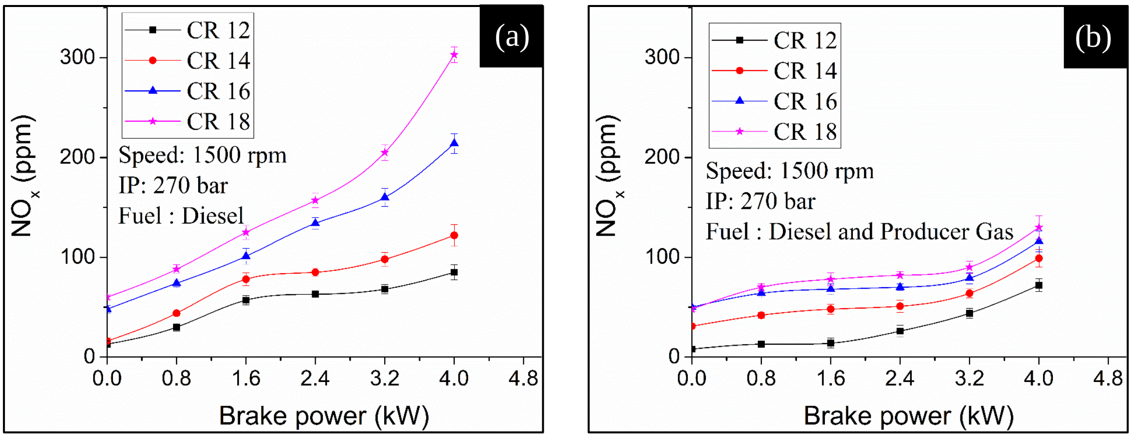



**Figure 4.13 Variation of NOx emissions at CRs for the case of (a) diesel, and (b) producer gas-diesel mode**

For a compression ratio of 18, Shrivastava et al. [26] reported a maximum NOx concentration of 325 ppm in diesel mode and 180 ppm in dual fuel mode at 80% load condition. Dhole et al. [105] reported a maximum NOx concentration of 904 ppm in dual fuel mode at 80% load condition. Yaliwal et al. [50] reported the level of NOx as 110 ppm in dual fuel mode at 80% load condition. The intensity of NOx emission in the present study was 205 ppm in diesel mode and 90 ppm in dual fuel mode.

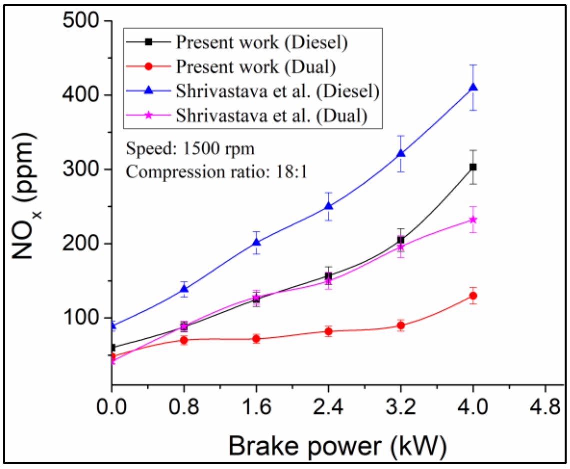



**Figure 4.14 Validation of NOx emission at compression ratio 18 with Shrivastava et al.** [26]

NOx emission at CR 18 was validated with the work of Shrivastava et al. [26] as given in Figure 4.14. Similar trend in NOx emission level with load was observed in the literature. The level of NOx in the present study was lower than the previously reported literature on similar work under the stated condition. The reason for this trend was the lower temperature in the combustion chamber that resulted in lower NOx levels. The concentration of NOx in the present study was within limits, as stated in the Gazette of India: Extraordinary, 2013, Part II-Section 3(I) [106].

The variation of the CO emission levels of diesel mode and dual fuel mode operation is shown in Figure 4.15 (a, b). As we know, the amount of CO formation is a function of unburnt gaseous fuel availability and its temperature, both parameters control the rate of fuel decomposition and oxidation [40]. It was observed that CO emission under dual fuel mode was higher than the diesel mode of operation. The CO emission was found to be 80.23–84.11% higher in dual fuel mode than diesel mode at 3.2 kW brake power. It was observed that with an increase in load, CO emission decreased in dual fuel mode. It happens because as the engine load increases, more fuel is required, and as a result, a richer air-fuel mixture enters into the cylinder. This richer mixture results in incomplete combustion and produces less amount of CO emission. From Figure 4.15, it was clear that CO emission levels decrease as the compression ratio increases. It was also noticed that lower emission was observed at 2.4–3.2 kW brake power in both modes of operation.

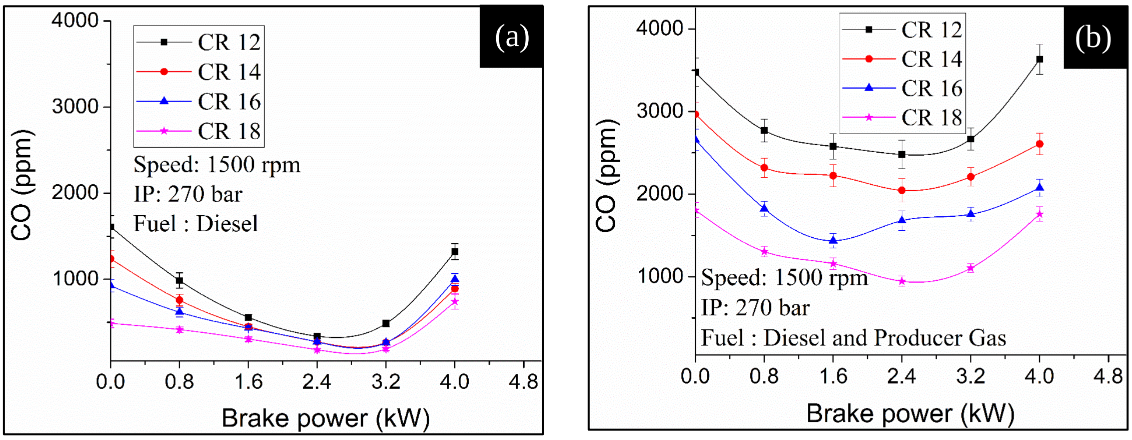



**Figure 4.15 Variation of CO emissions at different CRs for the case of (a) diesel, and (b) producer gas-diesel mode**

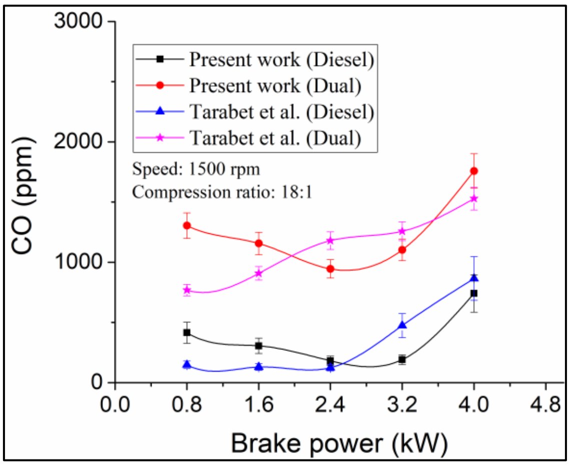



**Figure 4.16 Validation of CO emission at compression ratio 18 with Tarabet et al.** [48]

The reasons for higher CO emission in dual fuel mode included low oxygen presence in the air-producer gas mixture, resulting in incomplete combustion. For the compression ratio of 18 and 80% loading conditions, Sombatwong et al. [47] reported the maximum concentration of CO by 100 ppm in diesel mode and 500 ppm in dual fuel mode. Shrivastava et al. [26] reported the maximum level of CO as 10.0 ppm in diesel mode and 250 ppm in dual fuel mode. Ramadhas et al. [22] reported the maximum level of CO as 700 ppm in diesel mode and 1300 ppm in dual fuel mode. Yaliwal et al. [50] reported the maximum concentration of CO as 250 ppm in dual fuel mode. The CO at CR 18 was validated with the work of Tarabet et al. [48] as given in Figure 4.16. Similar trend in CO emission with brake power was observed in the literature. CO emission level in the present study was 191 ppm in diesel mode and 1103 ppm in dual fuel mode. The level of CO in the current work was lower than Ramadhas et al. [22] and higher than Sombatwong et al. [47], Tarabet et al. [48] and Shrivastava et al. [26].

The variation of hydrocarbon (HC) emission at different brake powers in diesel mode and dual fuel mode at different compression ratios is shown in Figure 4.17 (a, b). The variation of HC in the exhaust gases shows the quality of the combustion process of the engine [40]. It was observed that hydrocarbon emission in dual fuel mode was 63.41–70.04% higher than diesel mode. At a higher compression ratio, it decreased in both the modes of engine operation. This was because, at a higher compression ratio, temperature and pressure at the end of the compression stroke are high. The higher temperature of combustion shows better combustion of fuel. Hence, the average reduction of 63.63% in HC emission was achieved by increasing the compression ratio from 12 to 18 at 3.2 kW brake power.

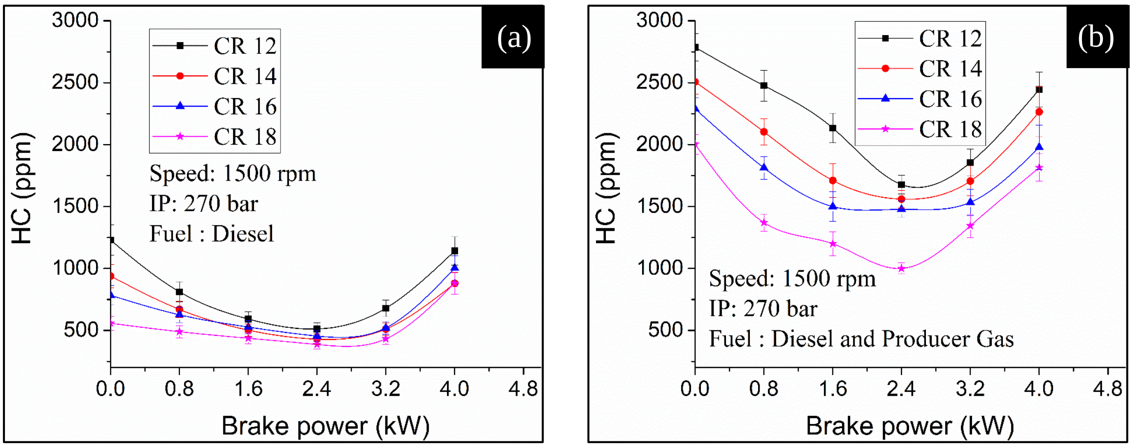



**Figure 4.17 Variation of HC emissions at different CRs for the case of (a) diesel, and (b) producer gas-diesel mode**

For a compression ratio of 18 and 80% loading conditions, Shrivastava et al. [26] reported the maximum concentration of HC by 15 and 20 ppm in diesel mode and dual fuel mode respectively. Dhole et al. [105] reported the level of HC as 2400 ppm in dual fuel mode. Banapurmath and Tewari [42] reported the concentration of HC as 46 ppm in dual fuel mode. The intensity of HC emission in the present study was 480 ppm in diesel mode and 1250 ppm in dual fuel mode.

However, the level of HC in the present work was higher in both modes of operation of the engine as compared to previous literature. The reason may be lower charge temperature, which results in slower combustion and allows a small amount of fuel to escape from the combustion process. HC emission at CR 18 was validated with the work of Sharma and Kaushal [63] as given in Figure 4.18. Similar trend in HC emission with load was observed in the literature.

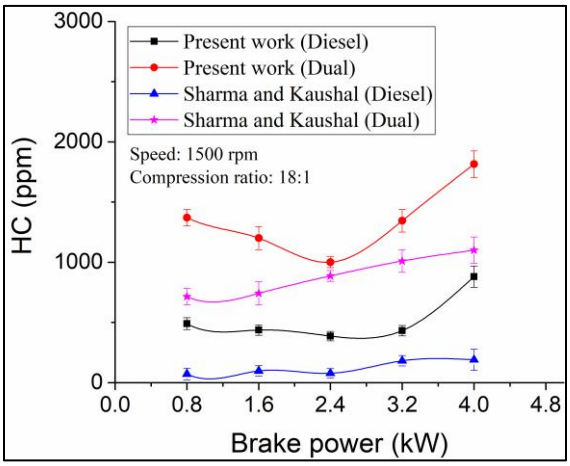



**Figure 4.18 Validation of HC emissions at CR 18 with Sharma and Kaushal** [63]

The variation of sulfur oxide (SOx) emission level at different brake power conditions in diesel mode and dual fuel mode respectively under terms of varying compression ratios as shown in Figure 4.19 (a, b). SOx emission increased with engine load.

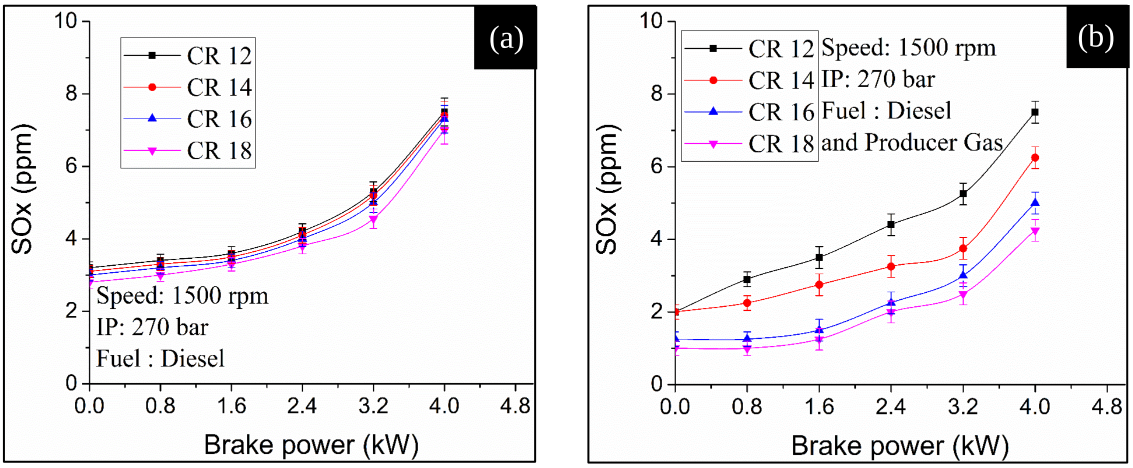



**Figure 4.19 Variation of SOx emissions at different CRs for the case of (a) diesel, and (b) producer gas-diesel mode**

It happened because more quantity of diesel is required at higher engine load. It was observed that SOx emission level in dual fuel mode was lower than diesel fuel mode at different compression ratios. SOx emission in dual fuel mode was 0.09, 27.91, 40.02, and

45.05% lower as compared to the diesel fuel mode at 80% (3.2 kW) BP and compression ratio 12, 14, 16, and 18 respectively. The reason for this trend was the lower sulfur content in biomass residue (0.01%) fuel than diesel fuel (0.05%).

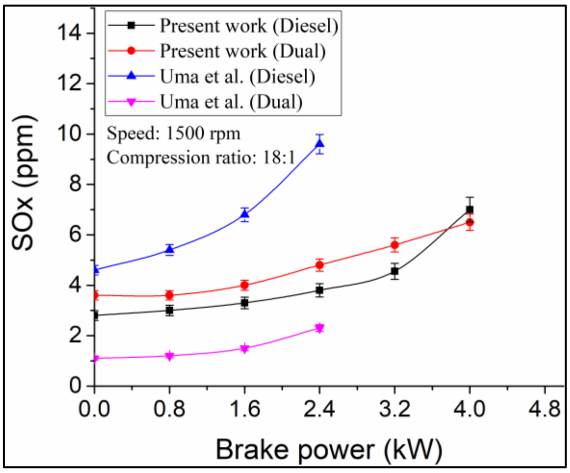



**Figure 4.20 Validation of SOx emissions at CR 18 with Uma et al.** [43]

For the compression ratio of 18 and 80% loading conditions, Uma et al. [43] reported the maximum SOx emission level to be 25 ppm, and Tan et al. [107] reported the maximum concentration of SOx emission level as 4.6 ppm. The SOx emission at CR 18 was validated with the work of Uma et al. [43] as given in Figure 4.20. Similar trend but lower SOx emission value with brake power was reported in the literature.

Exhaust gas temperature (EGT) was higher in dual fuel mode as compared to diesel mode as shown in Figure 4.21(a, b). From the observations, the EGT of diesel mode at full load was found to be 330°C and 380°C in dual fuel mode. The high EGT in the dual fuel mode was because of the higher energy supplied to the engine. It was also noted that the EGT reduced as the compression ratio increased from 12 to 18. The results obtained in the present work are in close agreement with the reported literature. For example, for the compression ratio of 18, Shrivastava et al.[26] reported maximum exhaust gas temperature 250°C in diesel mode and 300°C in dual fuel mode. Sombatwong et al. [47] reported a maximum exhaust gas temperature of 260°C in diesel mode and 320°C in dual fuel mode. Ramadhas et al.[22] reported maximum exhaust gas temperature of 390°C in diesel mode and 480°C in dual fuel mode. The EGT at CR 18 was validated with the work of Shrivastava et al. [26] as given in Figure 4.22. A similar trend in EGT value with load was observed in the literature.

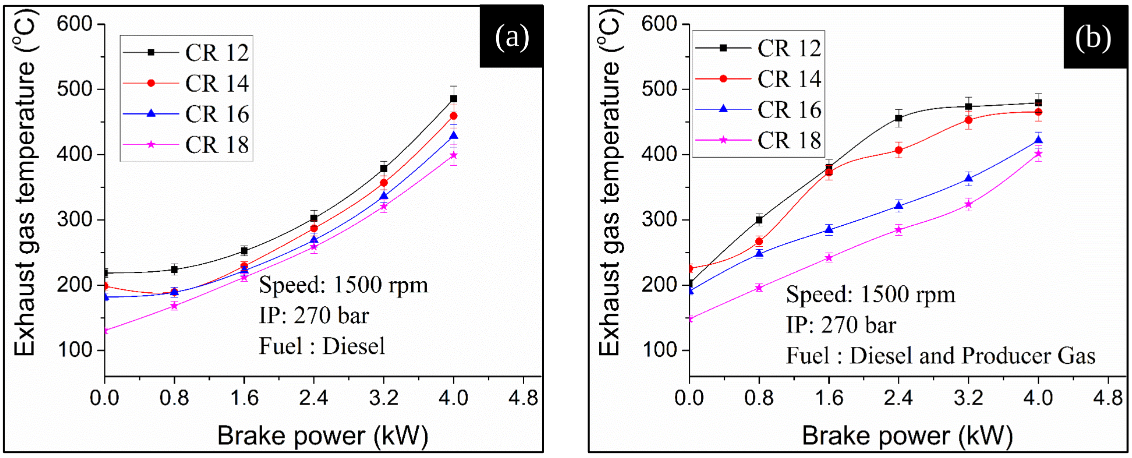



**Figure 4.21 Variation of EGT at different CRs for the case of (a) diesel, and (b) producer gas-diesel mode**

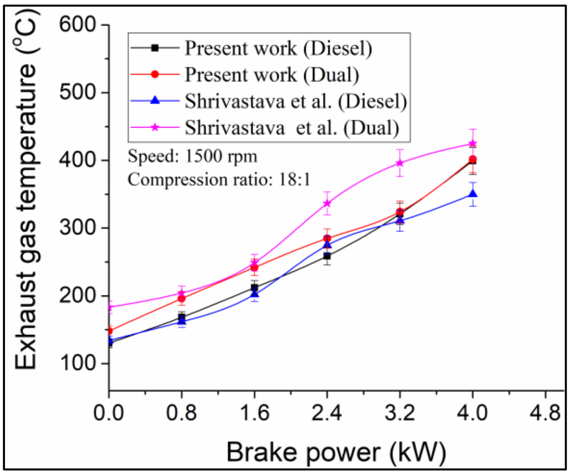



**Figure 4.22 Validation of EGT at CR 18 with Shrivastava et al.** [26]

Carbon dioxide ($CO_2$) emission level in dual fuel mode was 6.05 to 33.72% higher than diesel mode at 3.2 kW brake power. It was because producer gas containing some amount of $CO_2$ supply to the engine cylinder. It was observed that as load and compression ratio increased, $CO_2$ emission also increased in both the modes of operation as shown in Figure 4.23 (a, b). The reason behind this was at a higher compression ratio, temperature and pressure in the engine cylinder increased, resulting in better combustion of fuel. As a result, $CO_2$ emission increases. The results obtained in the present work are in close agreement with the reported literature. The $CO_2$ at CR 18 was validated with the work of

Nayak and Mishra [51] as given in Figure 4.24. Similar trend in $CO_2$ value with load was observed in the literature.

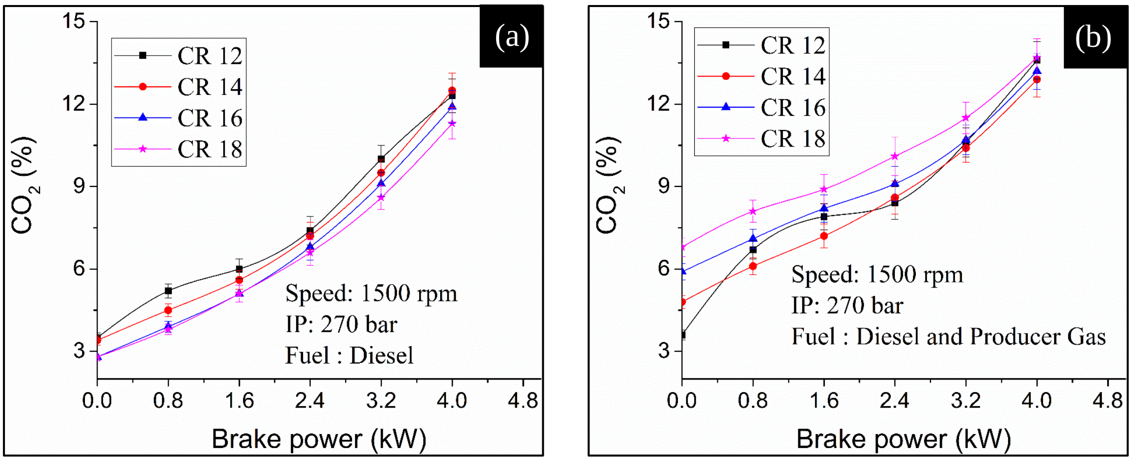



**Figure 4.23 Variation of $CO_2$ emissions at different CRs for the case of (a) diesel, and (b) producer gas-diesel mode**

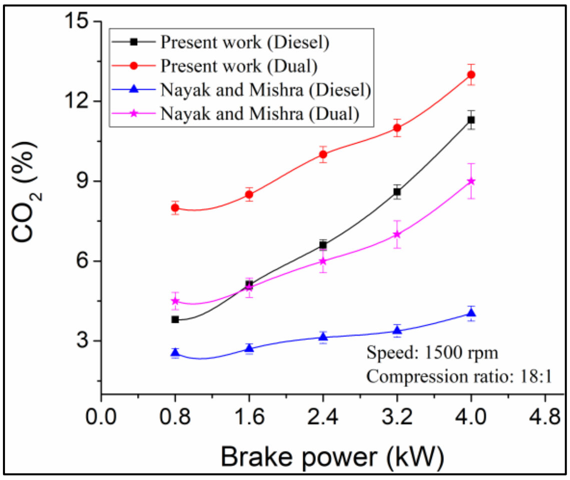



**Figure 4.24 Validation of $CO_2$ at CR 18 with Nayak and Mishra** [51]

For example, for the compression ratio of 18, Singh et al. [23] reported maximum $CO_2$ emission as 3.63% in diesel mode and 7.76% in dual fuel mode, Sahoo et al. [108] reported maximum $CO_2$ emission as 8.02% in diesel mode and 6.22% in dual fuel mode.

## 4.4 Combustion process in a dual fuel engine with biogas-diesel mode

The dual fuel engine performance was evaluated experimentally using biogas-diesel fuel. The performance results were compared to the diesel fuel mode.

The experimental cylinder pressure versus crank position data at different compression ratios for ten-cycles of expansion and compression strokes of the engine working cycle were used for the progress of combustion. The combustion process of the mixture of biogas and diesel is very challenging than that of single diesel fuel. The mixture of biogas and air experiences pre-ignition under compression stroke. Combustion of diesel fuel affects pre-ignition reactions. As a result, improvement in fuel conversion efficiency was observed. It was because of longer ignition delay and shorter combustion time under the dual fuel mode. Similar findings have been in the literature [109]. It occurred due to biogas combustion. From the experimental investigations, it was observed that ignition delay in both modes of operation declined with growth in compression ratio. This was because combustion temperature increases at higher compression ratios. The peak cylinder pressure (PCP) increased with load for both modes of engine operation. It happened because a higher amount of fuel is burnt during high load conditions.

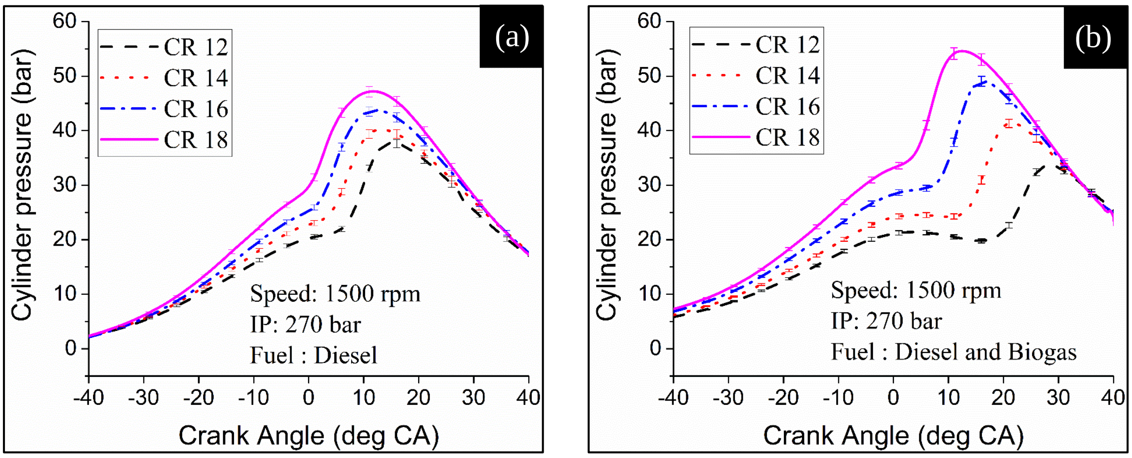



**Figure 4.25 Variation of cylinder pressure at different CRs for the case of (a) diesel, and (b) biogas-diesel mode**

The PCP was found to be 37.74, 40.23, 43.72, and 47.19 bar in the diesel mode and 33.64, 41.32, 49.05, and 54.56 bar for the dual fuel mode at compression ratios of 12, 14, 16, and 18 respectively. Figure 4.25 (a, b), shows that the peak cylinder pressure was shifted towards the expansion stroke. This was due to a delay in the combustion of biogas. It was noticed that the angle of crank corresponding to PCP was 15°, 13°, 12°, and 12° after TDC and 28°, 22°, 16°, and 13° after TDC under diesel mode and dual fuel mode at compression ratios 12, 14, 16 and 18 respectively. Similar results have been reported by the researcher [109].

The net heat released rate was calculated by using cylinder pressure data. The net heat released rate is proportional to the fuel burnt inside the engine cylinder. Net heat released rate at various compression ratios is shown in Figure 4.26 (a, b) for both modes of operation. It was observed that the net heat released rate lowered with an increase in compression ratio.

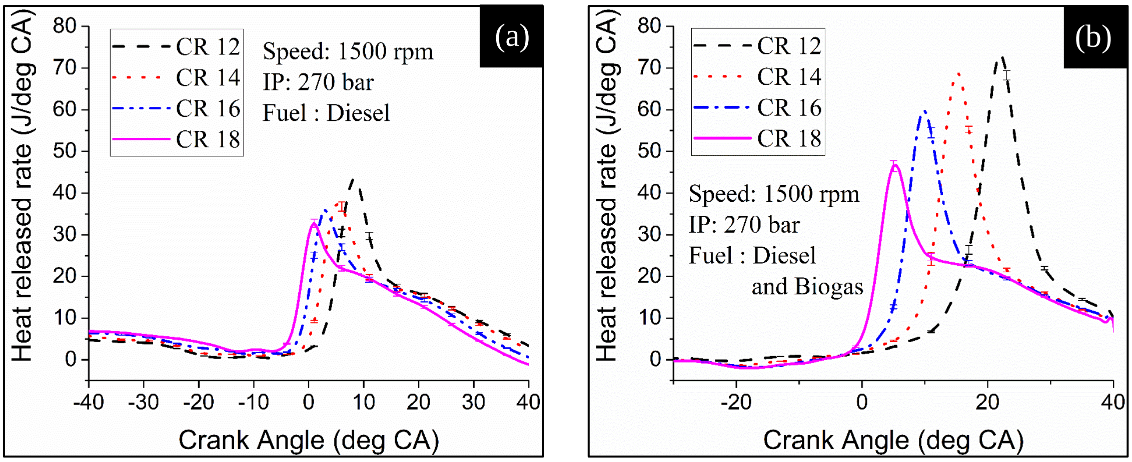



**Figure 4.26 Variation of heat release rate at different CRs for the case of (a) diesel, and (b) biogas-diesel mode**

The heat transfer rate increases during combustion at a higher compression ratio because of the high cylinder temperature at a higher compression ratio. Net heat released rate was obtained as 43.15, 37.16, 35.89, and 32.72 J/deg CA. for diesel mode and 73.18, 68.82, 59.65, and 46.47 J/deg CA for dual fuel mode at compression ratios of 12, 14, 16, and 18 respectively. A similar trend has been reported by the researcher [109].

## 4.5 Performance of a dual fuel engine with biogas-diesel mode

The performance of the dual fuel CI engine was evaluated experimentally under biogas and diesel fuel as pilot injection fuels. The performance results were compared with single diesel fuel mode operation.

### 4.5.1 Diesel fuel-saving under biogas-diesel mode

The gaseous fuel energy reduced the consumption of diesel in the dual fuel mode. It was observed that savings in the diesel fuel increased with growth in compression ratio, as shown in Figure 4.27. The high temperature of the combustion process during a higher compression ratio helps in the combustion of gaseous fuel smoothly. Diesel replacement varied from

8.11–25.21, 18.92–36.44, 20.10–50.07, and 38.81–48.25% at a compression ratio of 12, 14, 16, and 18 respectively. The maximum diesel fuel replacement was obtained as 25.21, 36.42, 50.32, and 48.25% at compression ratios of 12, 14, 16, and 18 respectively at 3.2 kW brake power. Maximum diesel fuel replacement was observed at a compression ratio of 18 at brake power of 3.2 kW as shown in Figure 4.27.

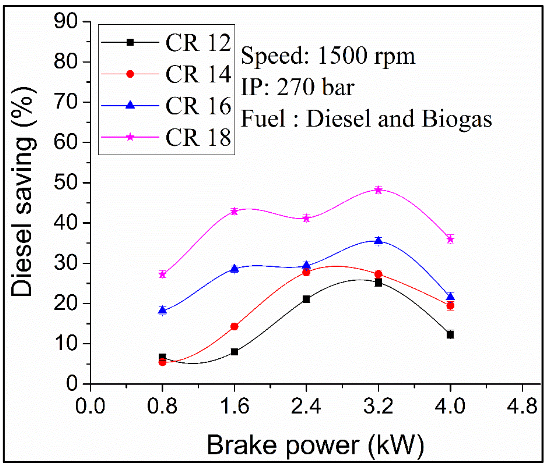



**Figure 4.27 Variation of diesel fuel savings under biogas-diesel mode at different CRs**

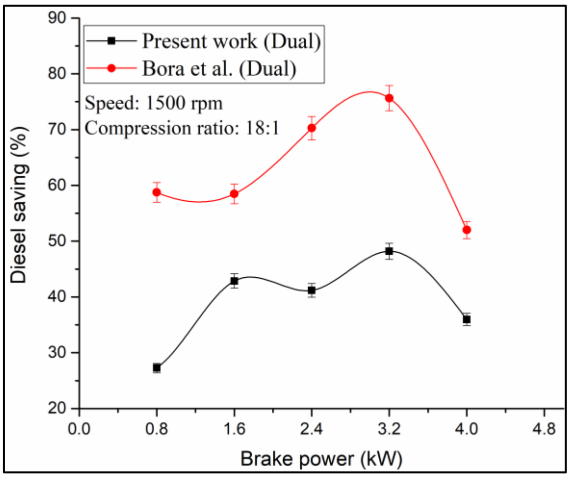



**Figure 4.28 Validation of Diesel fuel saving at CR 18 with Bora et al.**[76]

Diesel fuel replacement was lower at higher load conditions because at higher load a richer mixture is required. The diesel fuel saving at CR 18 was validated with the work of Bora et al.[76] as given in Figure 4.28. Similar trend in diesel fuel saving with BP was observed in the literature. However, it was lower than in previous literature. Bora and Saha [100] and Tippayawong et al. [110] reported maximum diesel fuel savings of 79.46% and

90.01% respectively. For the same loading conditions, a diesel saving of 48.21% was observed in the present study.

### 4.5.2 Brake thermal efficiency under biogas-diesel mode

Variations of Brake thermal efficiency (BTE) for the dual fuel VCR engine in both modes at different compression ratios is shown in Figure 4.29 (a, b). It is observed that the BTE increases with an increase in brake power and compression ratio conditions. This is due to the high temperature inside the cylinder during the higher load and compression ratio. The higher temperature in the cylinder helps in better combustion of fuel.

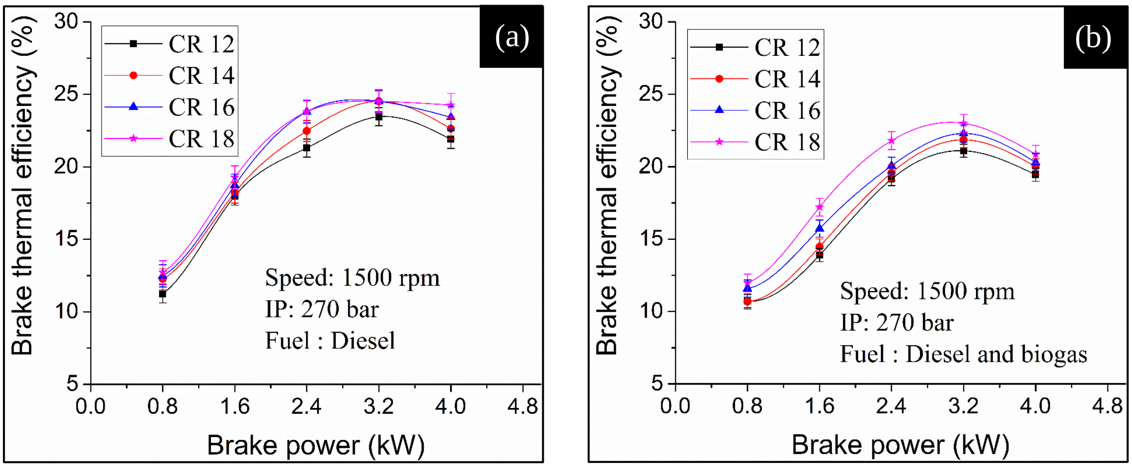



**Figure 4.29 Variation of BTE at different CRs for the case of (a) diesel, and (b) biogas-diesel mode**

It was observed that BTE was lower in dual fuel mode as compared to diesel mode. BTE was highest at 3.2 –4.0 kW brake power in both modes of operation. Maximum BTE of 23.5, 24.52, 24.53, and 24.54% in diesel mode and 21.12, 21.90, 22.32, and 23.08% in dual fuel mode at compression ratios of 12, 14, 16, and 18 respectively. BTE improved as the compression ratio increased from 12 to 18. BTE also depends on the flow rate of biogas in dual fuel mode. In the present work, it was observed that BTE was 10.22, 9.14, 8.23, and 6.15% lower in dual fuel mode as compared to diesel mode at a compression ratio of 12, 14, 16, and 18 respectively. Although, BTE using biogas in dual fuel mode is lower than the BTE of the same engine using diesel fuel. However, biogas is a renewable fuel. Therefore, we have to use this to save the environment and natural resources.

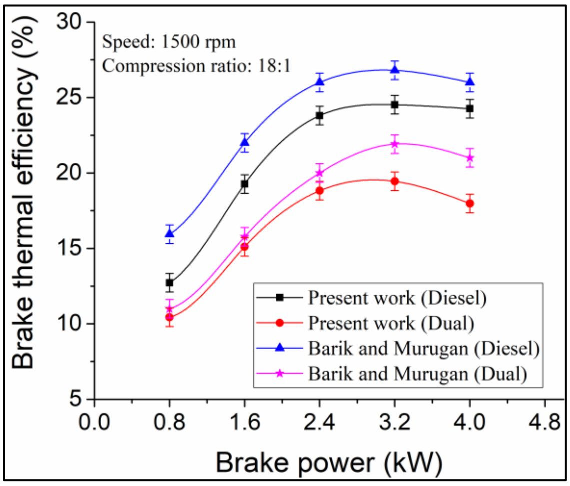



**Figure 4.30 Validation of BTE at CR 18 with Barik and Murugan** [75]

The BTE at compression ratio 18 was validated with the work of Barik and Murugan [75] as given in Figure 4.30. Similar trend in BTE values with brake power was observed in the literature. Bora and Saha [100], Gnanamoorthi and Devaradjane [101], and Barik and Murugan [67] have reported similar findings.

### 4.5.3 Brake specific fuel consumption under biogas-diesel mode

The change in brake specific fuel consumption (BSFC) in both modes at different compression ratios is shown in Figure 4.31 (a, b). It is observed that BSFC under dual fuel mode is higher.

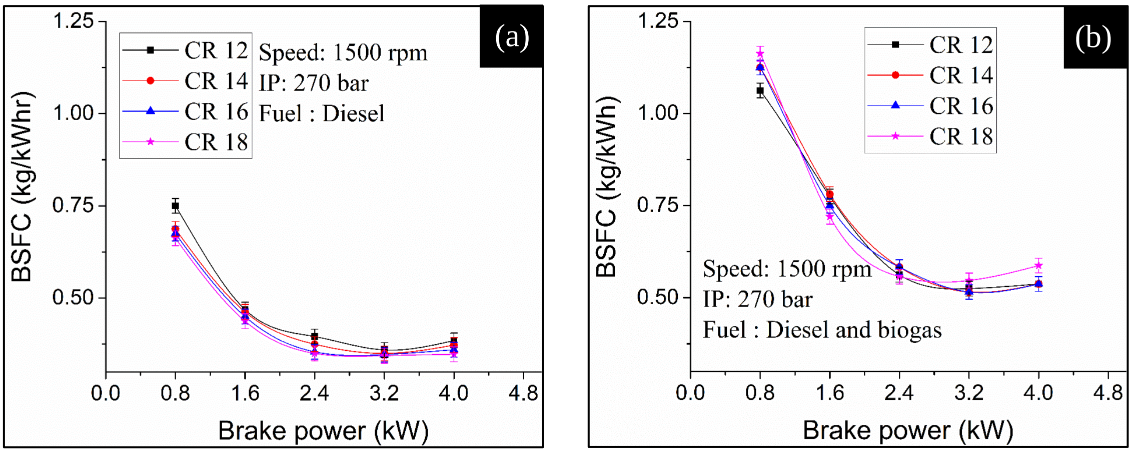



**Figure 4.31 BSFC of the dual fuel engine at different CRs for the case of (a) diesel, and (b) biogas-diesel mode**

It is because of the presence of $CO_2$ in it and the inferior heating value of biogas. It was also noticed that BSFC increased with an increase in biogas intake. It was because of the lower energy content of biogas. The BSFC in dual fuel mode was 50.70, 53.31, 52.04, and 49.22% higher than the diesel mode at compression ratios of 12, 14, 16, and 18 respectively. Minimum BSFC was at 3.2 kW brake power in diesel mode and 2.4 kW in dual fuel mode.

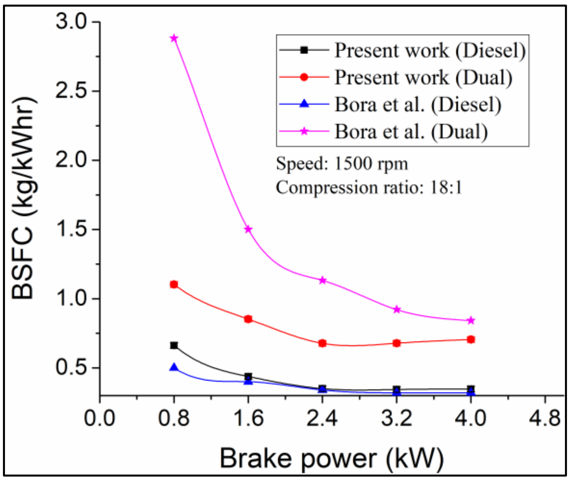



**Figure 4.32 Validation of BSFC at CR 18 with Bora et al.**[76]

It was observed that for 80–100% load, the BSFC improved in both the modes. The BSFC at compression ratio 18 was validated with the work of Bora et al.[76] as shown in **Figure 4.32**. Similar trend in BSFC values with brake power was observed in the literature. Barik and Murugan [75], Ambarita [70], Sombatwong et al. [47], Shrivastava et al. [26], and Sahoo et al. [108] have reported similar findings**.**

### 4.5.4 Brake specific energy consumption under biogas-diesel mode

Brake specific energy consumption (BSEC) in diesel mode and dual fuel mode at different CRs is shown in Figure 4.33. BSEC was calculated from the product of BSFC and calorific value of fuel (i.e., diesel and biogas). At lower loads, the presence of methane resulted in the formation of an extremely lean, incombustible methane-air-mixture, which consequently led to the loss of energy and hence higher BSEC in all cases of engine loading. However, methane addition improved combustion in dual fuel mode at higher loads. It can be noted that above 60% load BSEC slightly improves. From the experimental observations, the BSEC of dual fuel mode under biogas-diesel dual fuel mode was 1.10, 1.08, 0.72, and 0.61% higher than diesel mode at CR of 12, 14, 16, and 18 respectively.

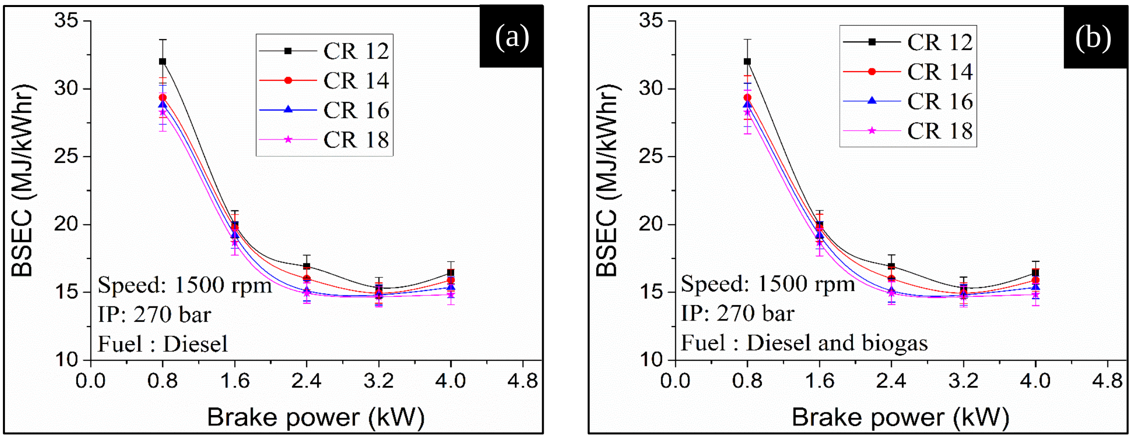



**Figure 4.33 BSEC of the dual fuel engine at different CRs for the case of (a) diesel, and (b) biogas-diesel mode**

The BSEC at compression ratio 18 was validated with the work of Bouguessa et al. [88] as given in Figure 4.34. Similar trend in BSEC with brake power was observed in the literature.

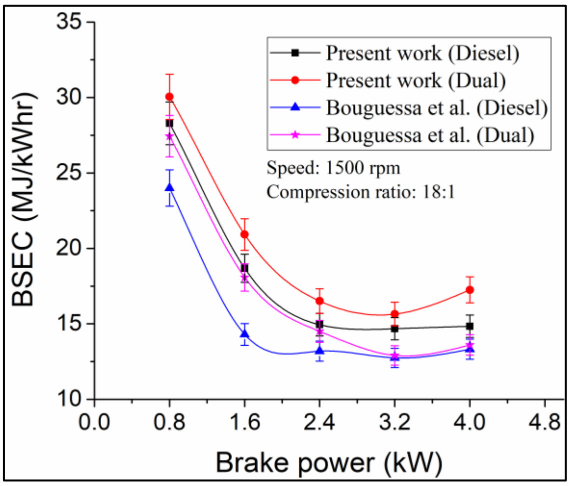



**Figure 4.34 Validation of BSEC at CR 18 with Bouguessa et al.** [88]

### 4.5.5 Noise emission in dual fuel mode engine using biogas-diesel mode

The noise level in the diesel engine is more than the petrol engine due to high pressure and large molecules of diesel and it depends on the physico-chemical properties of the fuel. In the present work, the noise level of dual fuel engine was measured in both modes of operation at different compression ratios with varying brake power as shown in Figure 4.35 (a, b). From the figures, it was observed that the noise level increased with the rise in compression ratios in both the modes of engine operation. This was due to higher pressure at the same injection pressure. The maximum noise level in dual fuel mode was 89.01 dB(A) at 3.2 kW brake power and a compression ratio of 18. It was observed that the noise level in dual fuel mode was 1.13% higher than the diesel fuel mode at 3.2 kW brake power.

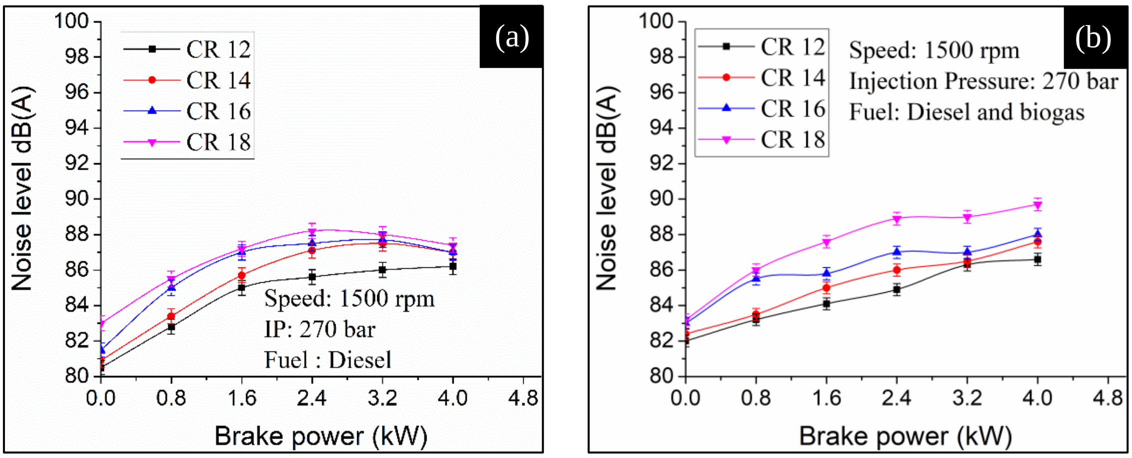



**Figure 4.35 Variation of the noise level of a dual fuel engine at different CRs for the case of (a) diesel, and (b) biogas-diesel mode**

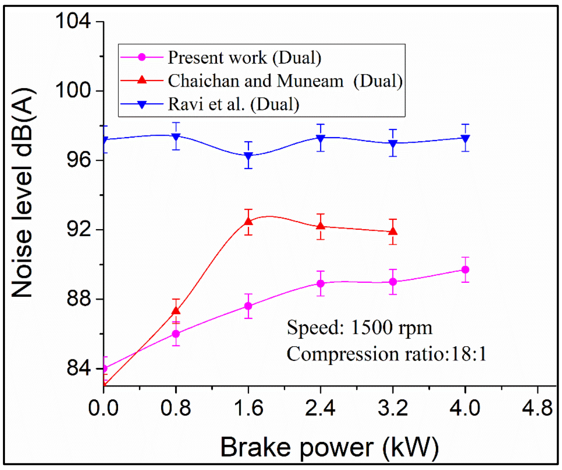



**Figure 4.36 Validation of noise level of dual fuel engine at compression ratio 18 with Chaichan and Muneam** [80] **and Ravi et al.** [111]

The noise level at the compression ratio of 18 was validated with the work of Chaichan and Muneam [80] and Ravi et al. [111] as given in Figure 4.36 and Table 4.2. A similar trend in noise level was observed from the figure. Ravi et al. [111] reported a 9.07% higher noise level. As seen above, the noise levels are more than the permissible limits (75 dB(A)) set by Environment Protection Rules, 1986 [103]. Hence, it is recommended that the noise level of a dual fuel engine can be reduced by incorporating a soundproof enclosure around the engine and adding a silencer to the exhaust system.

**Table 4.2 Validation of noise level** (dB(A)) **of dual fuel engine at CR 18 for biogas-diesel**

| **Brake power (kW)** | **Present work** | **Chaichan and Muneam** [80] | **Ravi et al.** [111] | **Noise level in present work in comparison to the literature** [80]**,** [111] |
|---|---|---|---|---|
| No load | 84.12 | 83.08 | 97.20 | 1.23% and 15.54% lower |
| 0.8 | 86.23 | 87.31 | 97.42 | 1.25% and 12.97% lower |
| 1.6 | 87.60 | 92.44 | 96.34 | 5.52% and 9.97% lower |
| 2.4 | 88.91 | 92.18 | 97.31 | 3.67% and 9.45% lower |
| 3.2 | 89.01 | 91.88 | 97.08 | 3.22% and 9.07% lower |
| 4.0 | 89.72 | - | 97.38 | 8.53% lower |

## 4.6 Emission characteristics of dual fuel engine using biogas-diesel mode

An exhaust emission analysis was conducted on a VCR CI engine for both diesel mode and dual fuel mode at different compression ratios and brake power conditions. The experimental studies of exhaust emission of NOx, CO, HC, SOx, EGT and $CO_2$ are discussed in this section.

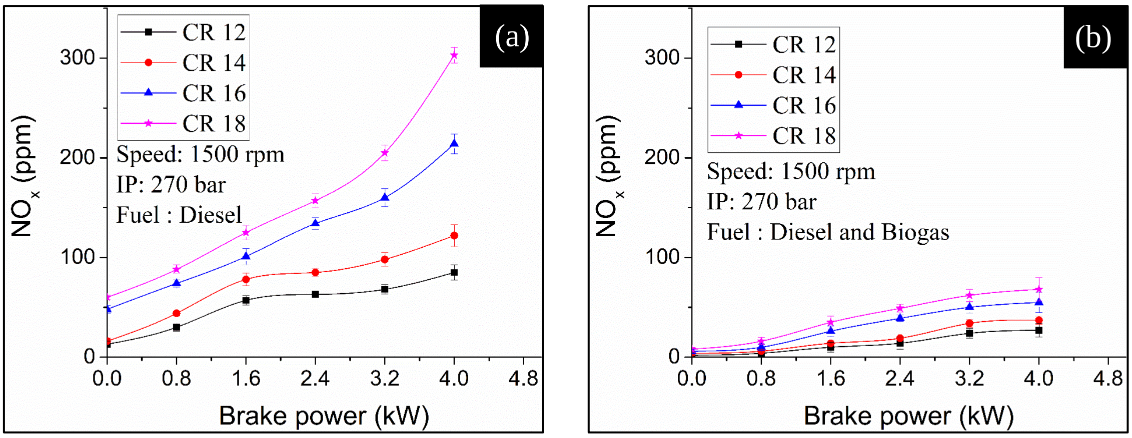



**Figure 4.37 Variation of NOx emissions at different CRs for the case of (a) diesel, and (b) biogas-diesel mode**

The comparison of NOx concentration in both modes is presented in Figure 4.37 (a, b). The generation of NOx is due to higher combustion temperature and oxygen concentration [40]. It was observed that as the compression ratio and load increases, NOx emission increases. This is because the rich fuel mixture supplied during higher load and higher compression ratio increase the combustion temperature. In the present research, NOx concentration was lower in the dual fuel mode compared to the diesel fuel mode. A maximum $NO_x$ concentration of 303 ppm was observed in the diesel mode. $NO_x$ concentration in dual fuel mode was 64.71, 65.33, 68.74, and 69.70% lower than the diesel mode at compression ratios 12, 14, 16, and 18 respectively. It is because of the lower calorific value of biogas and the presence of $CO_2$ which results in lesser combustion temperature.

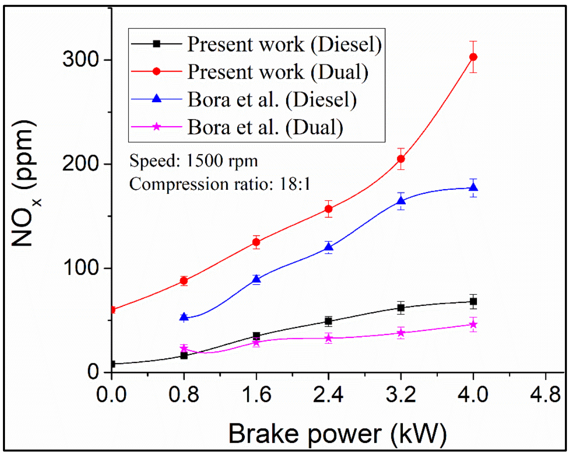



**Figure 4.38 Validation of NOx emission at CR 18 with Bora et al.**[76]

Various researchers have also investigated and reported on the NOx concentration. Bora and Saha [100] reported that at a compression ratio of 18, the maximum concentration of $NO_x$ emission was 170 ppm in diesel mode and 62 ppm in dual fuel mode. Nathan et al. [74] reported that $NO_x$ concentration in diesel mode varied between 250–470 ppm whereas, in dual fuel mode, it was under 60 ppm. In the present study, the $NO_x$ concentration was found to be 214 ppm in diesel mode and 55 ppm in dual fuel mode at 3.2 kW BP. The reason for the lower intensity of $NO_x$ emission in dual fuel mode was the lower combustion temperature. The NOx emission at compression ratio 18 was validated with the work of Bora et al. [76] as given in Figure 4.38. Similar trend in the NOx values with BP was observed in the literature. Importantly, it was also observed that $NO_x$ emission was within limits [106].

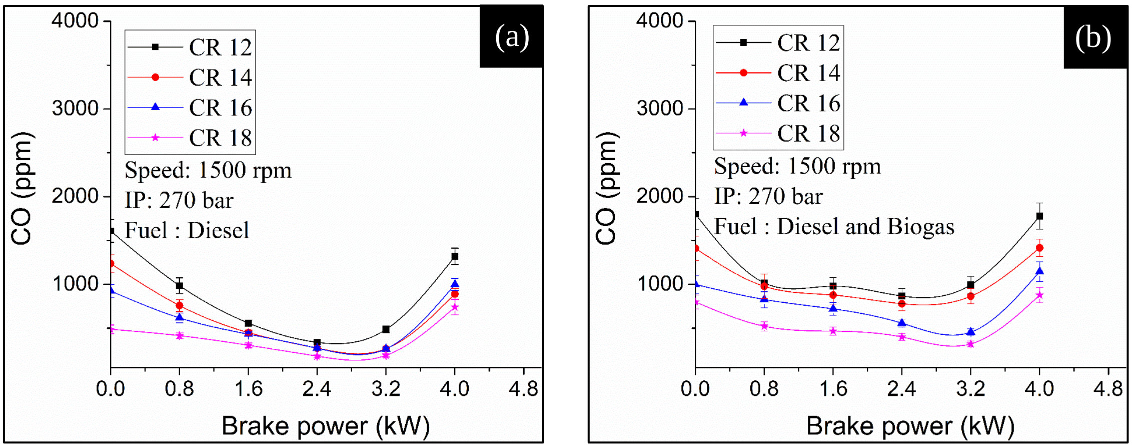



**Figure 4.39 Variation of CO emissions at different CRs for the case of (a) diesel, and (b) biogas-diesel mode**

The change in CO concentration under different brake power conditions in both modes is shown in Figure 4.39 (a, b). CO emission is a result of partial combustion of fuel in an engine cylinder [40]. It was observed that CO emission in dual fuel mode was higher than the diesel fuel mode. It was also observed that CO concentration decreased as the load increased in dual fuel mode. This was due to the rich air-fuel mixture supplied during high load conditions. The high temperature of the combustion process improves combustion efficiency. Further, low CO concentration was observed for the range of 2.4–3.2 kW Brake power in both the modes.

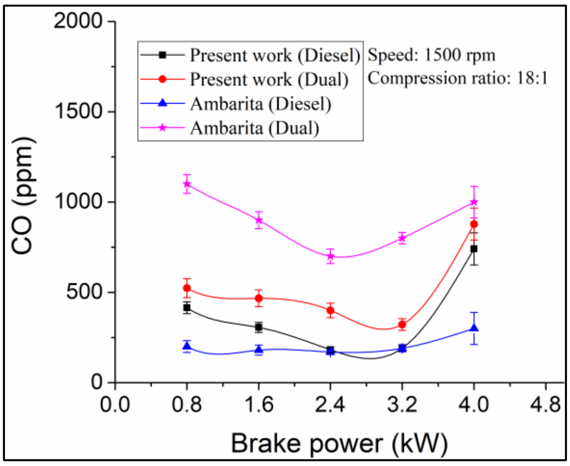



**Figure 4.40 Validation of CO emissions at CR 18 with Ambarita** [70]

At the compression ratio of 18, Ambarita [70] reported the intensity of CO emission as 130 ppm at 80% loading condition. Sombatwong et al. [47] reported maximum CO emission in diesel mode as 100 ppm, and in dual fuel mode as 500 ppm. In the present work, the maximum CO emission concentration in diesel mode was 191 ppm and in dual fuel, it was 839 ppm. The CO emission at CR 18 has been validated with the work of Ambarita [70] as given in Figure 4.40. Similar trend in the CO values with load has been observed in the literature. However, CO emission in the present study was higher in both modes of engine operation. The concentration of CO emission in the present study was within the limit compared to the emission standard of diesel engine [106].

The unburnt hydrocarbon (HC) emission level under different brake power conditions is shown in Figure 4.41 (a, b) for both the modes.

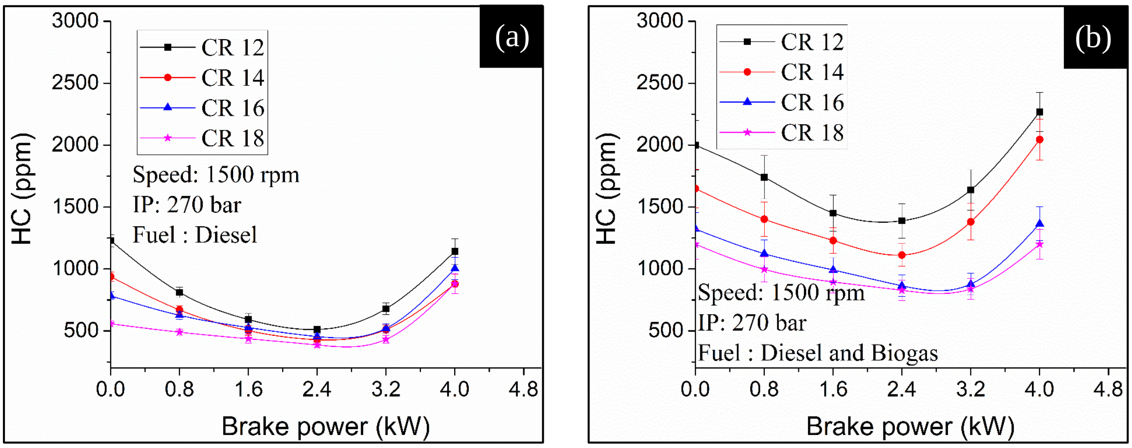



**Figure 4.41 Variation in HC emission level at different CRs for the case of (a) diesel, and (b) biogas-diesel mode**

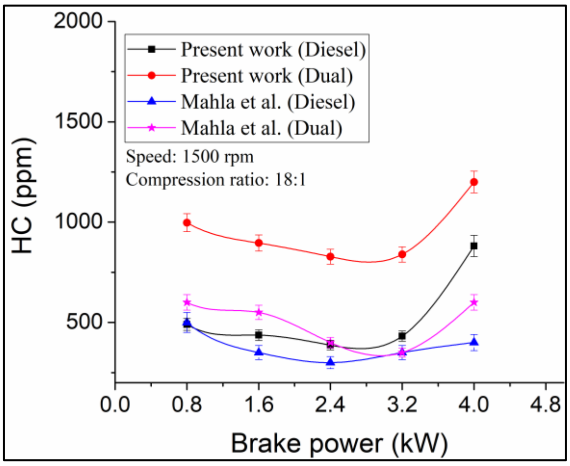



**Figure 4.42 Validation of HC emissions at CR 18 with Mahla et al.** [112]

HC emission levels in exhaust gases reflect on the engine combustion process [40]. HC emission in dual fuel mode was 58.52, 63.03, 40.62, and 48.51% higher than the diesel mode at compression ratios 12, 14, 16, and 18 respectively. It was observed that HC emission decreased in both modes of operation with the increase in compression ratio because the temperature of combustion increased with compression ratio. As a result, better combustion efficiency was achieved. The average reduction in HC emissions was 48.81% in diesel mode and 36.43% in dual fuel mode attained by raising the CR from 12 to 18 at 80% load. For a CR of 18, Dhole et al. [105] reported the HC concentration of 2400 ppm in dual fuel mode. Ambarita [70] reported HC emission of 200 ppm. Bora and Saha [100] reported the maximum HC emission of 125 ppm. The HC emission concentration in the present study was higher in both the modes of operation of the engine as compared to previous literature. The HC emission at CR 18 was validated with the work of Mahla et al. [112] as shown in Figure 4.42. Similar trend in the HC values with load was observed in the literature. However, the level of HC emission observed was within the limits of an emission standard for a diesel engine [106].

Variation in sulfur dioxide (SOx) emission level at different compression ratios in both modes is shown in Figure 4.43 (a, b).

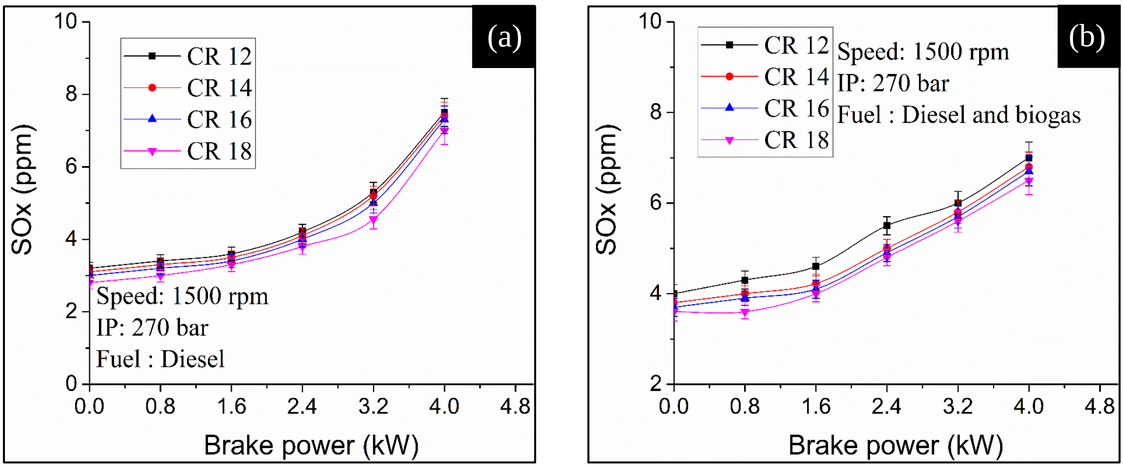



**Figure 4.43 Variation in SOx emission levels at different CRs for the case of (a) diesel; (b) biogas-diesel mode**

The SOx emissions are proportional to the sulfur content present in the fuel. It was noticed that the SOx emission increases with the growth of engine load conditions. The reason for this is due to higher sulfur content at higher loads. The SOx emission in dual fuel mode was 11.72, 10.35, 12.31, and 18.61% higher than the diesel fuel mode at 2.4 kW BP and CRs of 12, 14, 16, and 18 respectively. The trend was attributed to the higher sulfur content in vegetables and food wastes as compared to diesel fuel, which contains only 0.05%

sulfur by volume. The SOx emission concentration in the present study was 4.5 ppm in diesel mode and 5.6 ppm in dual fuel mode.

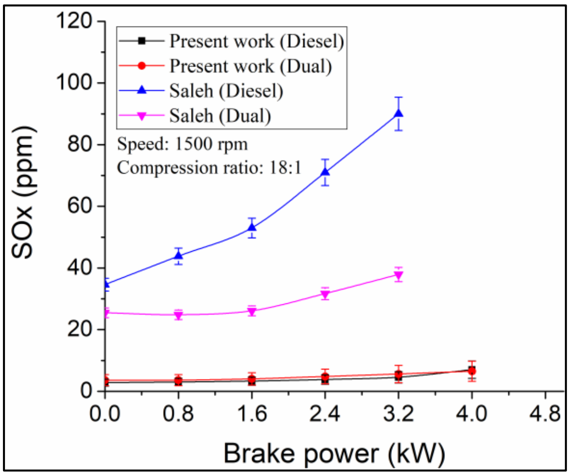



**Figure 4.44 Validation of SOx emissions at CR 18 Saleh [27]**

The SOx emission at CR 18 was validated with the work of Saleh [27] as given in Figure 4.44. Similar trend in the SOx values with load was observed in the literature. However, SOx emission was observed higher in the literature compared to the present work. Tan et al. [107] reported the maximum SOx emission was 4.6 ppm at a compression ratio of 18, and 80% load condition.

The deviation of EGT at different compression ratios in both modes is presented in Figure 4.45 (a, b). It was noticed that EGT decreased as the compression ratio change from 12 to 18. A higher EGT value was noted in dual fuel mode in all cases of loading. It was due to the relative delay in the start of the combustion of biogas. As a result, the high-temperature gases escape from the engine. The maximum EGT of 500℃ was observed at a compression ratio of 12 and the lowest EGT of 399℃ was observed at a compression ratio of 18 for the dual fuel mode. From the experimental observation, EGT of diesel fuel mode was found to be 330°C and in dual fuel mode was 399°C at full load. From the observations, it was also noticed that increases in compression ratio deviation in EGT in both modes of operation were reduced.

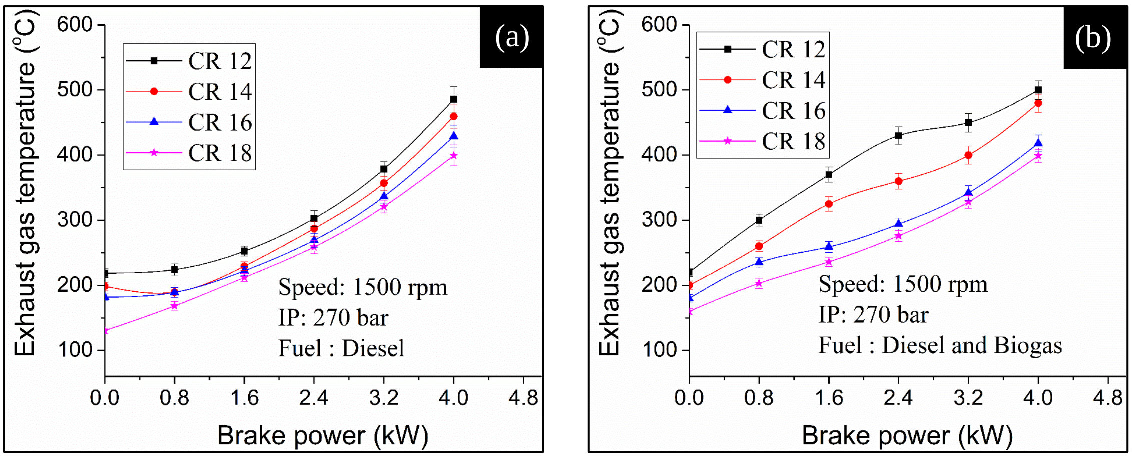



**Figure 4.45 Variation in EGT at different CRs for the case of (a) diesel, and (b) biogas-diesel mode**

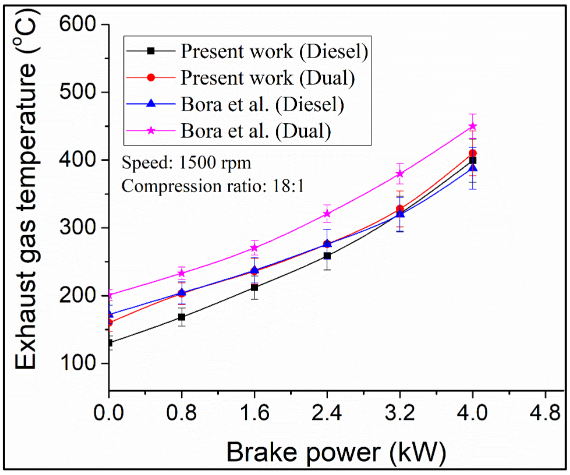



**Figure 4.46 Validation of EGT at CR 18 with Bora et al.** [76]

The Exhaust gas temperature at CR 18 was validated with the work of Bora et al.[76] as given in Figure 4.46. Similar trend in the EGT values with load was observed in the literature. Similar findings were reported for a compression ratio of 18, Shrivastava et al. [26] reported the highest EGT in diesel mode and dual fuel mode as 250°C and 300°C respectively. Sombatwong et al. [47] stated that the highest EGT under diesel fuel mode was 260°C and under dual fuel mode was 320°C. Ramadhas et al.[22] reported maximum EGT in diesel mode was 390°C and 480°C in dual fuel mode.

The variation of $CO_2$ emission level at different compression ratios in both the modes is shown in Figure 4.47 (a, b). The concentration of $CO_2$ emission in exhaust gas signifies the extent of complete combustion [113].

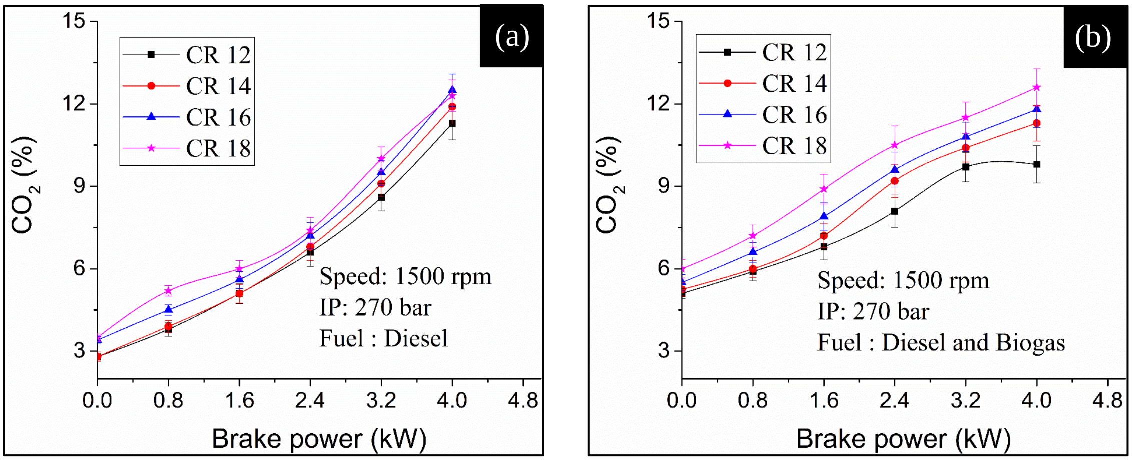



**Figure 4.47 Variation of $CO_2$ emission level at different CRs for the case of (a) diesel, and (b) biogas-diesel mode**

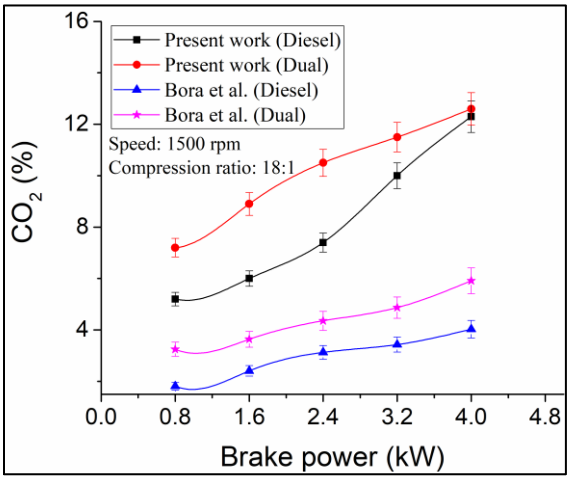



**Figure 4.48 Validation of $CO_2$ emission at CR 18 with Bora et al.** [76]

It was observed that the concentration of $CO_2$ emission increased with an increase in compression ratio and load in both the modes. This happened because the temperature of combustion increased with an increase in compression ratio resulting in better combustion efficiency. There was a reduction in $CO_2$ emission by 11.30, 12.52, 12.04, and 13.03% for dual fuel mode at compression ratios of 12, 14, 16, and 18 respectively. The $CO_2$ emission at CR 18 was validated with the work of Bora et al.[76] as given in Figure 4.48. A similar trend in the $CO_2$ emission with load was observed in the literature. Singh et al. [23] and Sahoo et al. [108] have reported similar findings.

### 4.7 Effect of injection pressure on the performance of dual fuel engine

From the experimental results, it was observed that emission characteristics and performance of a dual fuel engine improved with the increase in compression ratio. So, to compare the performance and emission characteristics of a dual fuel engine at different injection pressures, the compression ratio was set at the highest value of 18. Performance of a dual fuel engine was observed for producer gas-diesel mode at different injection pressures at a compression ratio of 18.

#### 4.7.1 Effect of injection pressure on diesel fuel saving under producer gas-diesel mode

The extent of diesel fuel substitution using producer gas at different brake power values is given in Figure 4.49.

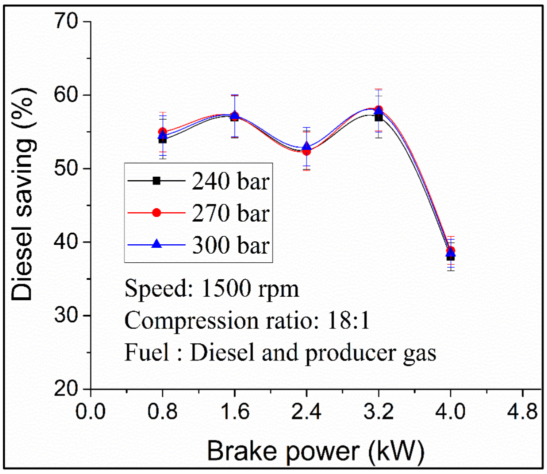



**Figure 4.49 Variation of diesel fuel saving at different injection pressure**

It was noticed that on the initial increase in injection pressure value from 240 to 270 bar, the saving in diesel fuel increased marginally by 1.70%. The combustion of gaseous fuel needs a minimum amount of pilot fuel in the combustion process. However, on further increase in injection pressure from 270 to 300 bar, a reduction in fuel saving was observed. Hence, the effect of injection pressure on diesel fuel saving levels not significant.

### 4.7.2 Effect of injection pressure on brake thermal efficiency (BTE) under producer gas-diesel mode

BTE of the engine shows the performance of an engine. The dual fuel engine was tested at different injection pressures (240, 270, and 300 bar), both in diesel mode as well as using producer gas-diesel in the dual fuel mode. On the increase in injection pressure from 240 to 270 bar, the BTE increased by 2.21% under the diesel fuel mode and 2.82% under the dual fuel mode, as given in Figure 4.50. The BTE value showed till 270 bar injection pressure and decreased thereafter. The increase in BTE value was because of fine droplets created as a result of the precise atomization of the pilot fuel and more spray penetration [114].

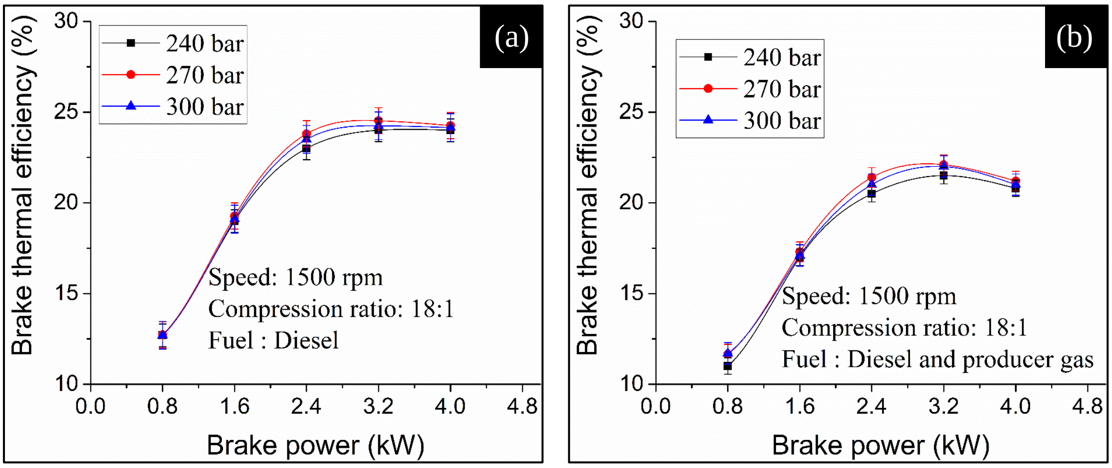



**Figure 4.50 Variation of BTE at different injection pressure for the case of (a) diesel, and (b) producer gas-diesel mode**

It enhances uniformity in the mixing of gas with air that undergoes complete combustion. The reduction in BTE at a higher injection pressure may be attributed to the reduction in the size of fuel droplets that affects the air-gas mixture distribution [115]. Srivastava et al. [114] reported the optimum injection pressure at 200 bar. However, in the present work, the optimum injection pressure was observed at 270 bar.

### 4.7.3 Effect of injection pressure on brake specific energy consumption (BSEC)

The variation of BSEC values in the diesel mode and the dual fuel mode at different injection pressures is shown in Figure 4.51. It was noticed that the BSEC of diesel mode decreased with increases in injection pressure under all loading conditions of a dual fuel engine. The decrease in injection pressure increases fuel particle diameter, which increases fuel penetration and ignition delay period during the combustion process, thereby increasing the BSEC of the dual fuel engine. Further increase in injection pressure caused shorter fuel penetration and shorter ignition delay, which increases BSEC.

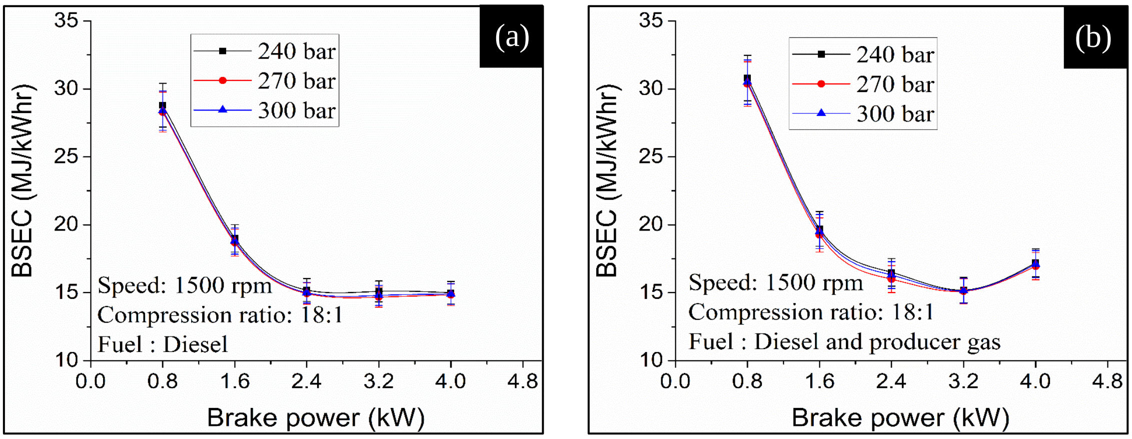



**Figure 4.51 Variation of BSEC at different injection pressure for the case of (a) diesel, and (b) producer gas-diesel mode**

BSEC in dual fuel mode reduced by 0.64% on increase in injection pressure from 240 to 270 bar. It was because of improved vaporization of the fuel, enhanced atomization, and improved air-fuel mixing [115]. On increasing injection pressure from 270 to 300 bar, there was an increase in BSEC. It was because of poor combustion at higher injection pressure.

### 4.7.4 Effect of injection pressure on the noise level of dual fuel engine

The effect of injection pressure on the noise level of a dual fuel engine is shown in Figure 4.52(a, b).

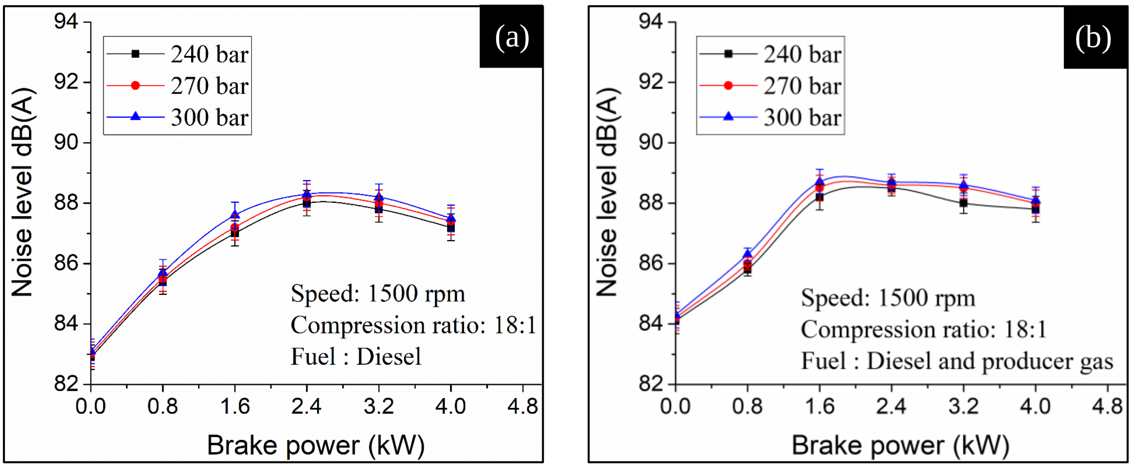



**Figure 4.52 Variation of sound level at different injection pressure for the case of (a) diesel, and (b) producer gas-diesel mode**

The noise level under diesel fuel mode increased with the rise in injection pressure beyond 270 bar. The noise level in diesel mode reduced by 0.56% on increasing injection pressure from 240 to 270 bar at 80% load. However, with a further increase in injection pressure to 300 bar, a growth of 0.11% in the noise level was observed for dual fuel mode.

## 4.8 Effect of injection pressure on emissions characteristics

NOx emissions are generated in the combustion chamber due to the existence of the excess amount of oxygen and high combustion temperature [40]. The variation of NOx concentration at different injection pressures is given in Figure 4.53.

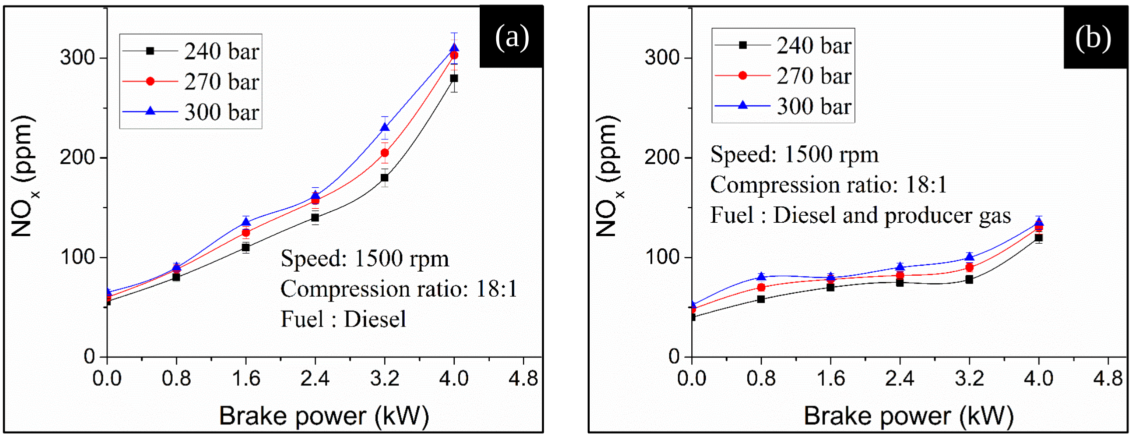



**Figure 4.53 Variation of NOx emission at different injection pressure for the case of (a) diesel, and (b) producer gas-diesel mode**

NOx concentration increased with the rise in injection pressure under both modes of engine operation. It was attributed to quick combustion and higher combustion temperature at higher injection pressures [114]. There was an increase in NOx levels by 13.92 and 13.33% for diesel fuel and producer gas-diesel mode respectively for the increase in injection pressure from 240 to 270 bar.

HC are organic compounds that are formed because of the incomplete combustion of hydrocarbon-based fuels. HC emission in dual fuel mode decreased with an increase in injection pressure as given in Figure 4.54 (a, b). It was because of better atomization, and improvement in air-fuel mixing at higher injection pressures [115]. However, beyond 270 HC emission increased. It was because of a decrease in the fuel droplet size, which reduced the momentum to travel into the air-gas mixture resulting in lower combustion efficiency.

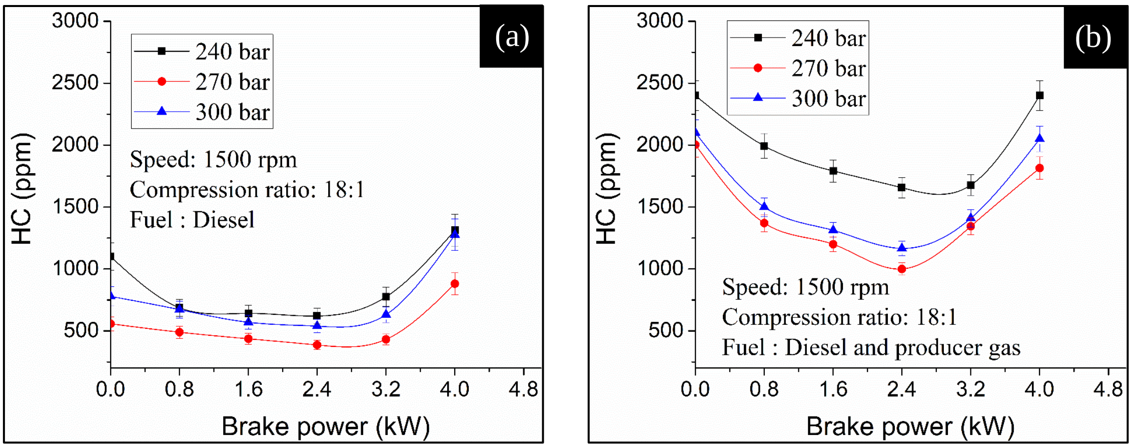



**Figure 4.54 Variation of HC emission levels at different injection pressure for the case of (a) diesel, and (b) producer gas-diesel mode**

CO is a colorless toxic gas. Even a minor quantity of CO concentration when inhaled slows down the mental and physical activities. Figure 4.55 shows the variation in CO emission levels as a function of brake power at different injection pressures.

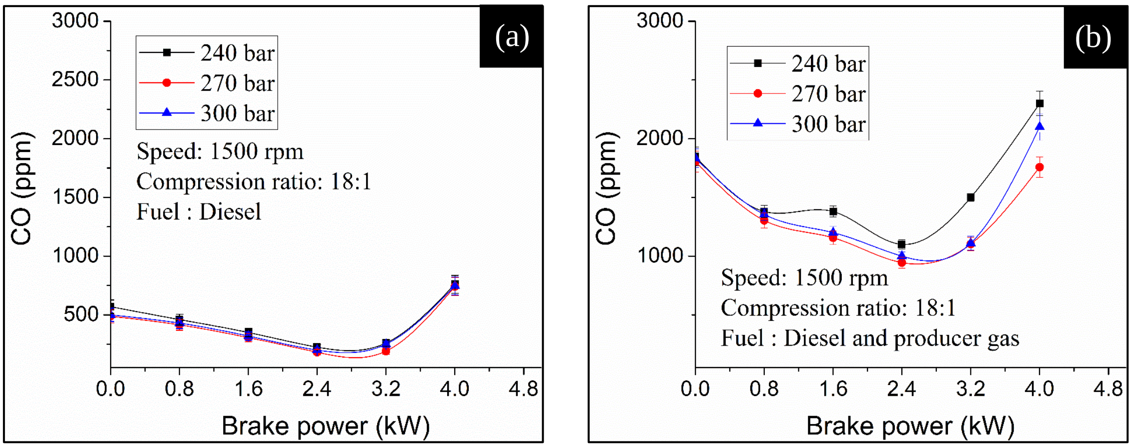



**Figure 4.55 Variation of CO emission at different injection pressure for the case of (a) diesel, and (b) producer gas-diesel mode**

CO emission is generated in the engine exhaust because of incomplete combustion of the air-fuel mixture. It was noticed that higher CO emission occurs at injection pressures 240 and 300 bar. It is because of the lower combustion efficiency at injection pressures other than optimum value. In the diesel mode, a reduction of 15.12% in CO emission was observed by increasing the injection pressure from 240 to 270 bar, and a reduction of 26.52% in CO concentration was noticed under the dual fuel mode.

The variation in SOx concentration at different injection pressures is shown in Figure 4.56. SOx concentration generated is proportional to the amount of sulfur content in the fuel used [116]. The SOx emission increased with the rise in engine brake power in both modes of operation. It happened because the amount of diesel fuel increased with an increase in load. Hence, the amount of sulfur content in fuel increased.

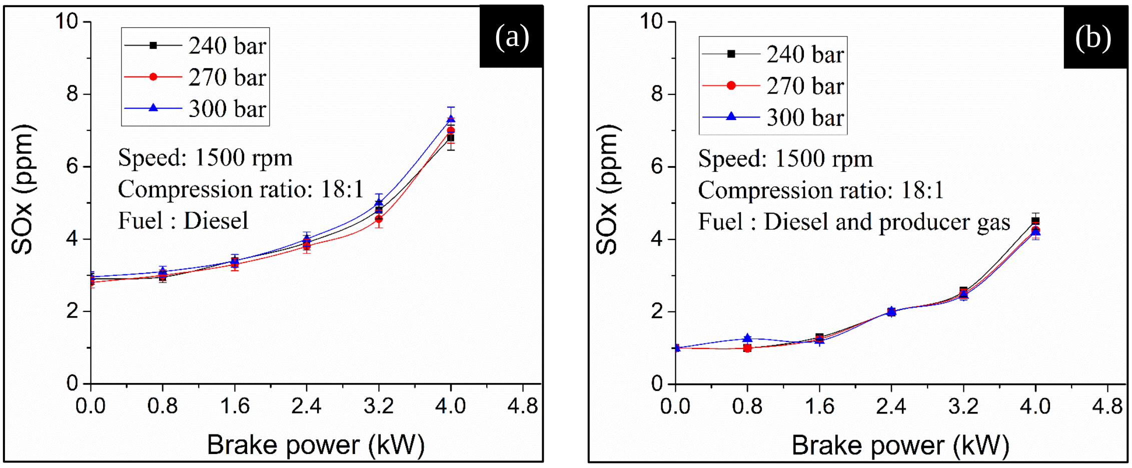



**Figure 4.56 Variation of SOx emission at different injection pressure for the case of (a) diesel, and (b) producer gas-diesel mode**

SOx emission levels were lower in dual fuel mode at all cases of injection pressures. With the increase in injection pressure from 240 bar to 300 bar, SOx emission level increased by 1.22% and 2.04% in diesel mode and dual fuel mode respectively. SOx emission level further increased at a higher injection pressure of 300 bar. It was due to more pilot fuel consumption at higher injection pressure.

## 4.9 Effect of injection pressure on the performance of dual fuel engine using biogas-diesel mode

The dual fuel engine performance was investigated experimentally under biogas-diesel mode by varying injection pressure (240, 270, and 300 bar) at a CR of 18.

### 4.9.1 Effect of injection pressure on diesel fuel saving using biogas-diesel mode

The variation in diesel fuel substitution using biogas-diesel mode is shown in Figure 4.57. It was noticed that with an increase in injection pressure, the saving in diesel fuel increased marginally. This is because the combustion of gaseous fuel needs a minimum amount of pilot fuel in the combustion process. 0.45% diesel saving was noticed under producer gas-diesel dual fuel mode by increasing injection pressure from 240 to 270 bar. However, 0.34% of diesel saving was observed when injection pressure was further increased from 270 to

300 bar. It is because higher injection pressure slows down the flame propagation as vapor of diesel fuel is excessively diluted into the air-gas mixture

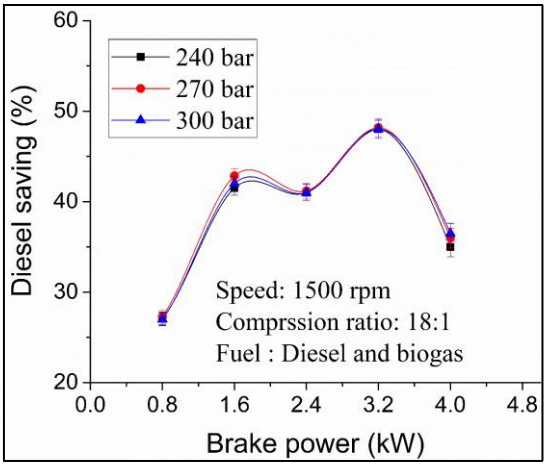



**Figure 4.57 Deviation of diesel fuel saving at different injection pressure.**

### 4.9.2 Effect of injection pressure on brake thermal efficiency (BTE) using biogas-diesel mode

Figure 4.58 shows the influence of injection pressure on the BTE of diesel mode and dual fuel mode. BTE showed an increase with the increase in injection pressure. It is because, at higher injection pressure, the liquid fuel droplets atomize very fast in the air-fuel mixture and improve the combustion efficiency of the engine.

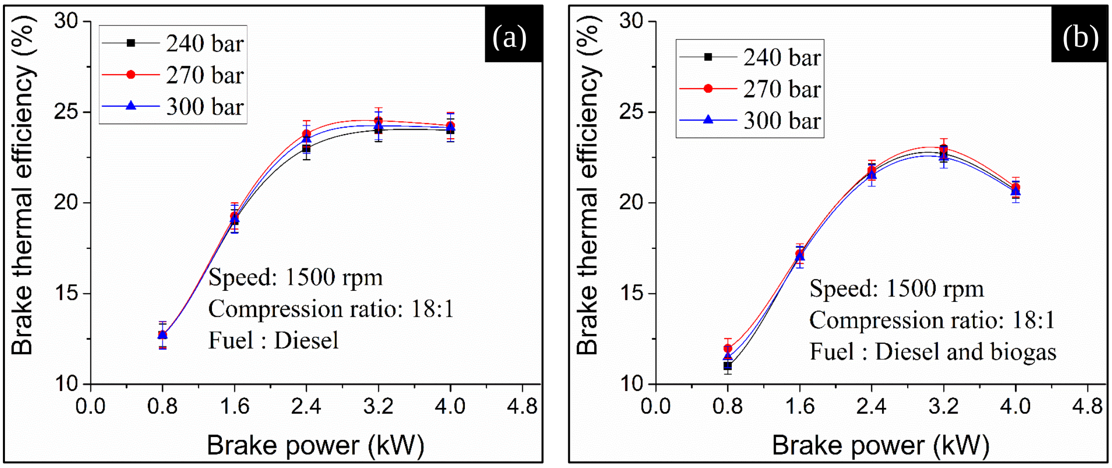



**Figure 4.58 Variation of BTE at different injection pressure for the case of (a) diesel, and (b) biogas-diesel mode**

The BTE was improved by 2.22% when injection pressure was increased from 240 to 270 bar in diesel mode, and by 1.34% under biogas-diesel dual fuel mode for 80% load condition.

However, with further increase in injection pressure, the BTE decreased. The reduction in BTE at this higher injection pressure was because of a decrease in the size of liquid droplets. The decrease in size decreased the momentum effect which influenced the homogeneity of the air-fuel mixture [114,115]. The highest BTE of 24.52 and 22.16% were found at 270 bar in diesel mode and biogas-diesel mode respectively. Srivastava et al. [114] have reported similar findings.

### 4.9.3 Effect of injection pressure on brake specific energy consumption (BSEC) using biogas-diesel mode

The variation in BSEC under diesel mode and dual fuel mode is shown in Figure 4.59. The increase in injection pressure increased the BSEC in both modes of engine operation. The reduction in injection pressure enlarges liquid fuel particle diameter and increases fuel penetration. As a result, the BSEC increases at higher injection pressure.

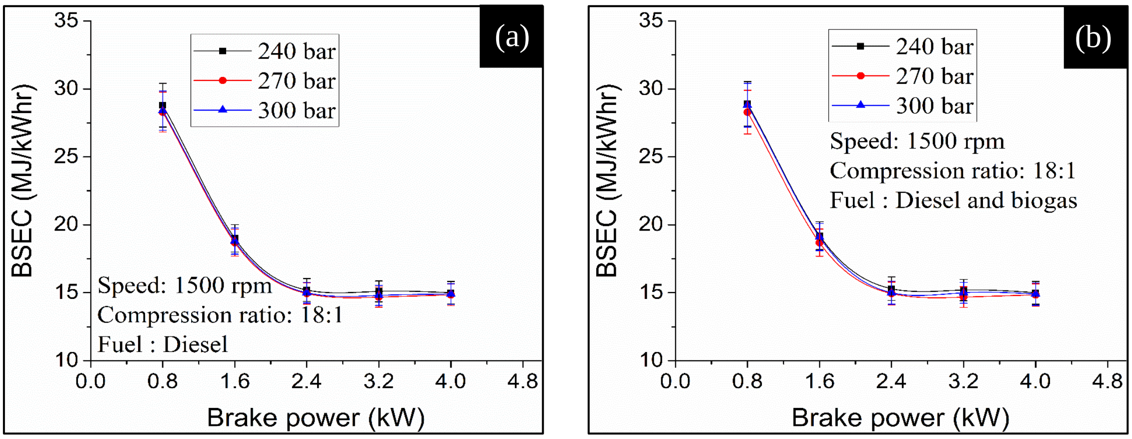



**Figure 4.59 Variation of BSEC at different injection pressure for the case of (a) diesel, and (b) biogas-diesel mode**

It was observed that on increasing injection pressure, the BSEC reduced till 270 bar and beyond 270 bar, the BSEC again increased. It may be due to shorter fuel penetration and shorter ignition delay at higher injection pressure [115]. The BSEC reduced by 2.82% under diesel mode and 9.75% under dual fuel mode when injection pressure was initially increased from 240 to 270 bar.

### 4.9.4 Effect of injection pressure on noise emission in dual fuel engine using biogas-diesel mode

The variation in the noise level of a dual fuel engine as a function of injection pressure is given in Figure 4.60 (a, b). It was observed that as fuel injection pressure increased, the noise level under diesel mode increased. It was because of the higher temperature of combustion and pressure generated at higher injection pressures. It was noticed that the noise level under dual fuel mode exhibited a lesser effect above the 60% load. The sound level was observed in the range of 87.51–87.62 dB(A) under dual fuel mode and 86.14–86.46 dB(A) under diesel mode at 3.2 kW brake power. The optimum injection pressure in the present study was 270 bar.

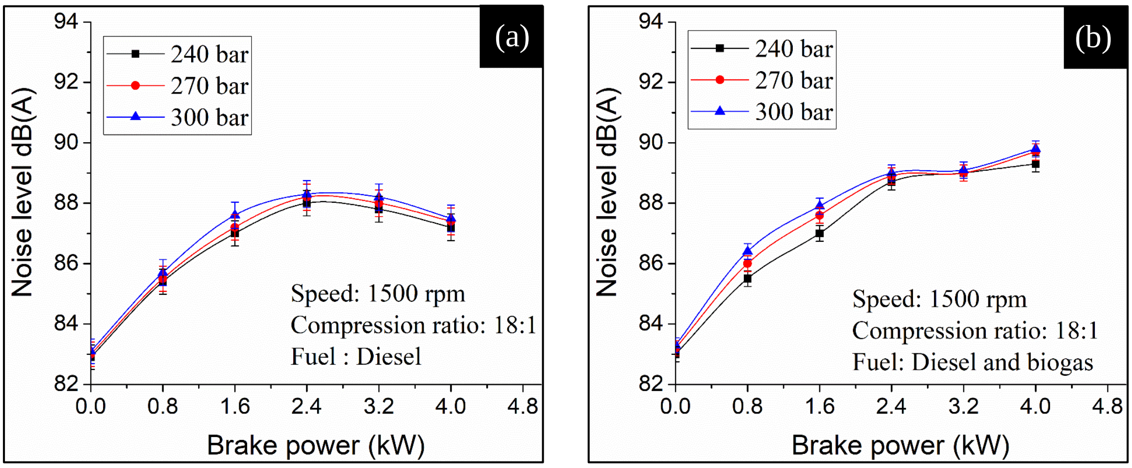



**Figure 4.60 Variation of noise level at different injection pressure for the case of (a) diesel, and (b) biogas-diesel mode**

## 4.10 Effect of injection pressure on emission characteristics using biogas-diesel mode

The variation in NOx emission levels as a function of brake power at different injection pressures is shown in Figure 4.61. From experimental observations, it was seen that NOx emission increased in both the modes of engine operation with the increase in injection pressure. It was because of rapid combustion and higher combustion temperatures [114]. There was an increase in NOx level by 28.25 and 12.55% under diesel fuel and biogas-diesel mode respectively by increasing injection pressure from 240–300 bar.

The variation of HC emission levels under diesel and dual fuel mode is shown in Figure 4.62 (a, b). The HC emission under dual fuel mode was higher than diesel fuel mode.

It was noticed that HC emission was reduced by 44.37% in diesel mode and 13.52% in biogas-diesel dual fuel mode with the rise in injection pressure from 240 to 270 bar. It was because of the quick-burning of air-fuel mixture and high atomization of pilot fuel at a higher injection pressure of 270 bar. However, the increase in injection pressure to 270 bar showed an increase in HC emission. It was because of higher liquid pressure, the liquid fuel droplets were less in size and traveled a shorter distance. As a result, the air-fuel mixture in the engine cylinder was not uniform [115]. Ryu [117] reported similar findings.

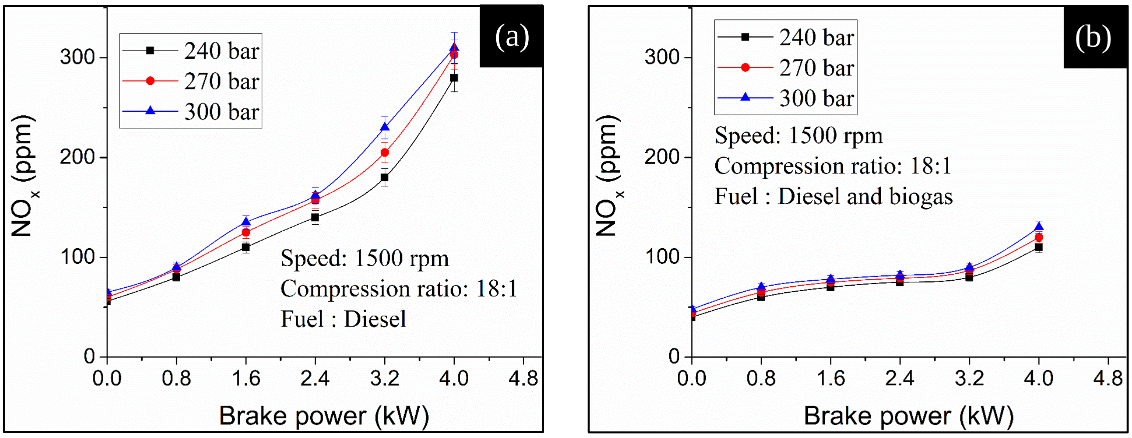



**Figure 4.61. Variation of NOx emission levels at different injection pressure for the case of (a) diesel, and (b) biogas-diesel mode**

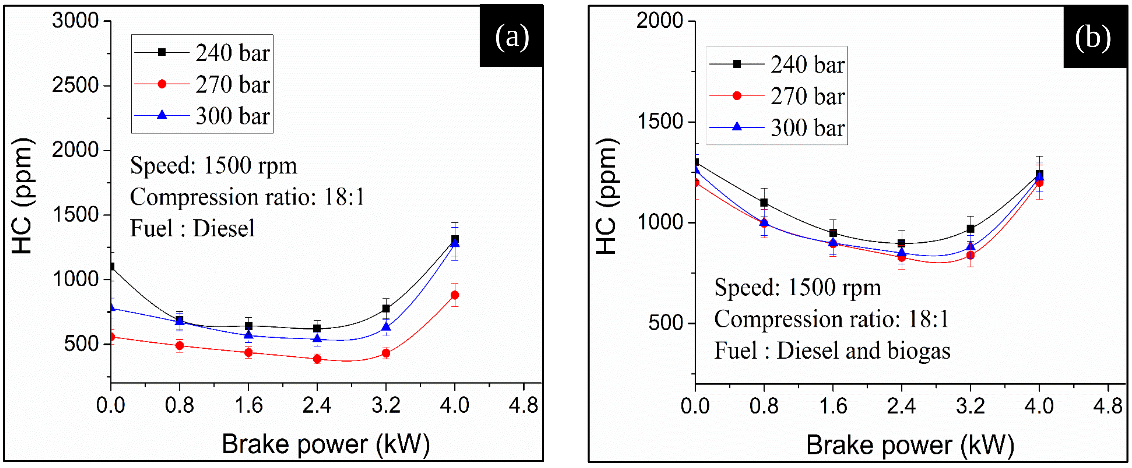



**Figure 4.62 Variation of HC emission at different injection pressure for the case of (a) diesel, and (b) biogas-diesel mode**

The variation in CO emission levels in dual fuel engine at different injection pressures is given in Figure 4.41(a, b). It was noticed that a lower injection pressure under both modes of engine operation generated more CO emission. It was because of the lower atomization of pilot fuel at lower injection pressure. The concentration of CO emission was reduced by 15.15% in diesel mode and 8.05% under dual fuel mode by raising the injection pressure from 240 to 270 bar. Maximum CO emission level occurred at an injection pressure of 300 bar. The lowest CO emission was noticed at 3.2 kW BP (80% load). Quadri et al. [115] have reported similar findings.

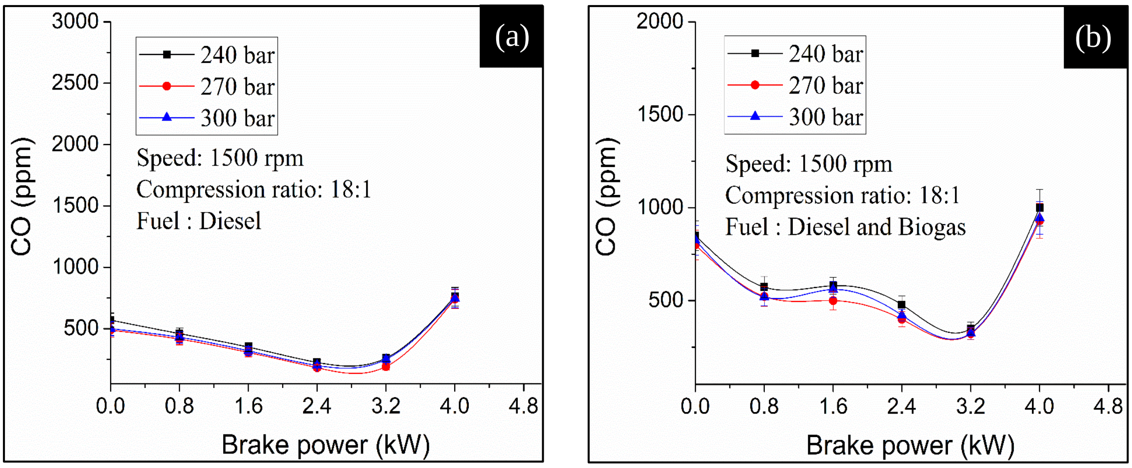



**Figure 4.63 Variation of CO emission at different injection pressure for the case of (a) diesel, and (b) biogas-diesel mode**

The variation in sulfur oxide (SOx) emission levels as a function of brake power for diesel and dual fuel mode under varying injection pressures is shown in Figure 4.64.

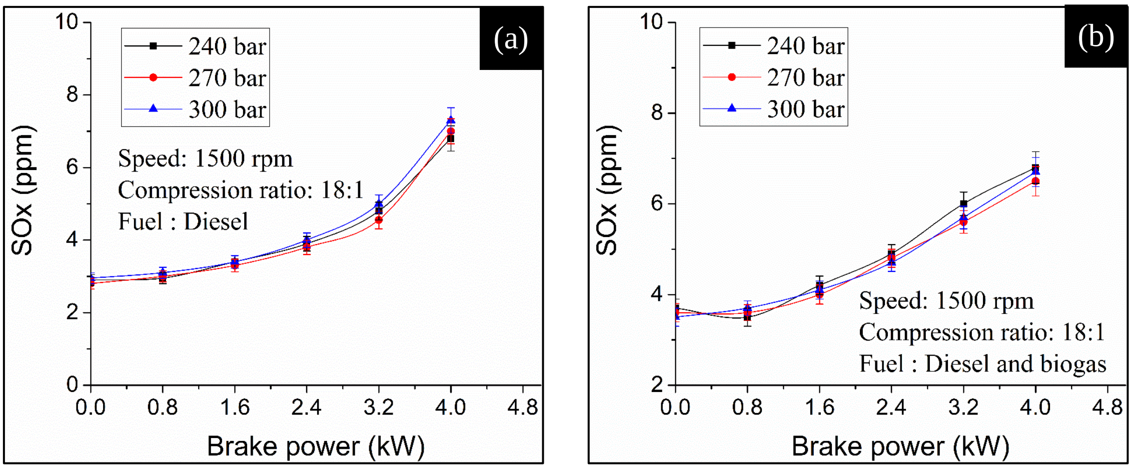



**Figure 4.64 Variation of SOx emission at different injection pressure**

The concentration of SOx emission level increased with the rise of injection pressure above the optimum value. The SOx emission reduced by 4.22% in diesel mode and a 6.75% reduction was noticed under dual fuel mode on an increase of injection pressure from 240 to 270 bar. However, with the increase of injection pressure above 270 bar, SOx emission increased by 8.63 and 1.76% in diesel and biogas-diesel mode respectively. This was due to more pilot fuel consumption at higher injection pressure.

**Table 4.3 Summary of results of dual fuel engine at CR 18 and 3.2 kW brake power.**

| Parameters | Diesel only | Producer gas-diesel | Biogas-diesel |
|---|---|---|---|
| Cylinder pressure (bar) | 47.19 | 54.49 | 54.56 |
| Heat released rate (J/deg CA) | 32.76 | 46.93 | 46.47 |
| Diesel fuel saving (%) | ---- | 58.02 | 48.25 |
| BSFC (kg/kWhr) | 0.33 | 0.54 | 0.34 |
| BSEC (MJ/kWhr) | 14.68 | 15.1 | 14.80 |
| Noise level (dB(A)) | 88.10 | 88.53 | 89.01 |
| EGT (°C) | 320 | 324 | 328 |
| NOx emission (ppm) | 205 | 90 | 62 |
| CO emission (ppm) | 191 | 1103 | 322 |
| HC emission (ppm) | 432 | 1355 | 839 |
| SOx emission (ppm) | 4.60 | 2.5 | 5.6 |
| $CO_2$ emission (%) | 10 | 11.5 | 11.6 |
| Injection pressure (bar) | 270 | 270 | 270 |

## 4.11 Summary

The results of the producer gas-diesel and biogas-diesel in dual fuel mode using the VCR diesel engine were discussed in this chapter. The results of dual mode operation were compared against a single fuel diesel engine at various CRs and injection pressures at varying brake power values. The results of dual fuel mode showed a significant saving in fossil (diesel) fuel. Both the producer gas-diesel mode, as well as biogas-diesel modes, benefited the surrounding environment by reducing agricultural residue, solid waste, and various emission gases viz SOx, and NOx etc. The decrease in BTE at all loading conditions in dual fuel mode was due to ignition delay at lower loads and also because the gaseous fuel had a high self-ignition temperature. A further decline in BTE was because of the lower calorific value of the gaseous fuel. The HC and CO emissions of a dual fuel engine increased as compared to the conventional diesel mode. However, using producer gas-diesel and biogas-diesel dual fuel modes reduced other harmful emissions to the environment and utilized the surplus biomass residue.

# Chapter 5
# Conclusions and recommendations for future work

In the present work, an effort has been made to find a solution to the utilization of biomass waste in the Punjab region which is primarily an agricultural state of the Indian sub-continent. In this context, a test rig of a downdraft gasifier coupled with a dual fuel engine was developed. Producer gas generated by gasifying the agricultural waste was used along with diesel fuel to run the engine in the dual fuel mode. The performance and emission analyses of the engine run, both in the conventional as well as in the dual fuel mode were evaluated and compared.

In addition, an effort has been made to generate biogas from kitchen waste from the hostels of the University at Patiala and utilize the gas in the dual fuel engine. The results are encouraging to adopt this technology for producing electricity from alternate fuels.

Some of the key conclusions are listed below thematically:

## 5.1 Performance parameters

- A substantial increase in brake thermal efficiency of 8.26% in biogas-diesel and 17.99% in producer gas-diesel at CR of 18 was observed due to an increase in mean effective pressure.
- The effect of variation of injection pressure was studied. Maximum BTE was observed at an injection pressure of 270 bar at CR of 18. Higher injection pressure reduced the ignition delay period, which in turn reduced knocking.
- Diesel fuel saving of 54.61% and 48.21% under producer gas-diesel and biogas-diesel mode respectively at CR of 18 were observed. This offered significant savings in the precious fossil fuel.

## 5.2 Noise level

- Efforts were made to measure the noise level to find out the effectiveness of the engine running in the dual fuel mode. The noise level was increased when the engine was operated in the dual fuel mode (Please refer to Figure 4.11 and Figure 4.35).

## 5.3 Emission level

Internal combustion engines generate undesirable emissions of various gases viz. $CO_2$, CO, HC, NOx etc. Efforts were made to study the emissions from the dual fuel engine with biogas-diesel and producer gas-diesel modes. The following are some of the key points.

**Biogas-diesel mode**

- NOx emission in the engine exhaust depends on the availability of oxygen in the fuel mixture and the temperature of the combustion chamber. NOx emission in dual fuel got reduced by 69.70% at CR of 18 at higher loading conditions. The low oxygen fraction and low temperature in the combustion chamber reduced the NOx generation significantly.
- CO and HC emissions were found to be higher in dual fuel mode at higher loading conditions due to incomplete combustion.
- SOx emission was found to be higher in biogas-diesel mode at higher loading conditions as sulphur content in biogas was found to higher in comparison to diesel.
- The effect of variation of injection of diesel in dual fuel mode was studied. It was observed that NOx, CO, and SOx emissions were reduced marginally at higher injection pressure.

**Producer gas-diesel mode**

- NOx emission in producer gas-diesel mode was higher than the biogas-diesel mode at the same CR and injection pressure. A similar trend was observed for HC, and CO emissions levels also.
- At higher injection pressure and temperature, the emissions in both the above cases were found to be lower.

## 5.4 Recommendations for future work

Two alternative gaseous fuels generated from the organic bio-waste in rural and urban areas were utilized in dual fuel mode. There are some shortcomings in the dual fuel operation which can be overcome by further development in this area. In this connection, the scop and suggestions for future studies are discussed below:

- The effect of the composition and flow rate of producer gas and biogas on the performance of dual fuel engine can be studied by incorporating an online gas analyzer in the test rig.
- Biomass pellets and forest residue can be used to generate producer gas which can subsequently be used in dual fuel engine.
- Reforming of biogas can be achieved by adding suitable catalysts and the effect of reformed biogas-diesel fuel on performance and emission parameters can be studied.

## 5.5 Application potential

- Dual fuel engines can be used for constant engine speed operations (stationary applications) like generator set, water pumps for agriculture applications and small backup power stations.

## 5.6 Weakness in the present work

- Biogas digester did not work in winters due to the low ambient temperature.
- Biogas contains $H_2S$ which gives a foul smell and corrodes engine parts.
- Tar formation in the gasifier is also a problem.
- At present, dual fuel engines apply to stationary purposes only.

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# List of publications

**Published in peer-reviewed (SCI) journals**

- S. Lal, S.K. Mohapatra, The effect of compression ratio on the performance and emission characteristics of a dual fuel diesel engine using biomass derived producer gas, Applied Thermal Engineering. 119 (2017) 63–72. doi:10.1016/j.applthermaleng.2017.03.038.
- S. Lal, S.K. Mohapatra, A feasibility study to utilize kitchen waste for power generation in urban areas using CI engine, Energy Sources, Part A: Recovery, Utilization, and Environmental Effects. 42 (2020) 1914–1922. doi:10.1080/15567036.2019.1604895.

# Annexure-A

**Engine performance calculations:**

1. **Brake power (kW):**

$$BP = \frac{2*\pi*N*T}{60*1000}\ kW \quad (A1)$$

$$BP = \frac{2*\pi*N*W*R}{60000}\ kW \quad (A2)$$

2. **Brake mean effective pressure (bar):**

$$BMEP = \frac{BP*60}{\frac{\pi}{4}*D^2*L*\frac{N}{2}*\text{No of Cylce}*100}\ bar \quad (A3)$$

3. **Brake specific fuel consumption (kg/kWhr):**

$$BSFC = \frac{\text{Fuel flow in kg/hr}}{BP} \quad (A4)$$

4. **Brake thermal efficiency (BTE) (%):**

Brake thermal efficiency is the ratio of energy in the brake power, BP to the input fuel energy.

$$BTE = \frac{BP*3600*100}{\text{Fuel flow in}\frac{kg}{hr}*\text{CalValue of fuel}} \quad (A5)$$

5. **Airflow (kg/hr):**

$$\text{Air flow} = Cd * \frac{\pi}{4} * d^2 \sqrt{2 * g * h * \left(\frac{Wden}{Aden}\right)} * 3600 * Aden \quad (A6)$$

**Where**

Cd = Coefficient of discharge of orifice
d = Orifice diameter (m)
g = Acceleration due to gravity ($m/s^2$) = 9.81 $m/s^2$
h = Differential head across orifice (m of water)
Wden = Water density ($kg/m^3$) =@1000 $kg/m^3$
Aden = Air density at working condition ($kg/m^3$)

6. **Air fuel ratio:**

$$\text{Air fuel ratio} = \frac{\text{Air Flow}}{\text{Fuel Flow}} \quad (A7)$$

7. **Engine Configuration data used in experimentation work:**

1. Pulses per revolution: 360
2. No. of cycles: 10
3. Fuel pipe diameter: 12.40 mm
4. Fuel measuring interval: 60 sec
5. Orifice diameter: 20 mm
6. Dynamometer arm length:185 mm
7. Speed scanning interval: 2000 ms

8. **Theoretical constants used**

1. Fuel density: 830 kg/$m^3$
2. Calorific value of diesel fuel: 42000 KJ/kg
3. Orifice coefficient of discharge: 0.60

# Annexure-B

**Engine detailed technical specifications**

| **Technical specifications** | |
|---|---|
| Model | TV1 |
| Make | Kirloskar Oil Engine |
| Type | Four-stroke, Water cooled, Diesel |
| No. of cylinder | One |
| Bore (mm) | 87.5 |
| Stroke (mm) | 110 |
| Combustion principle | Compression ignition |
| Cubic capacity (liter) | 0.661 |
| Compression ratio | 17.5:1 |
| Peak pressure (kg/cm$^2$) | 77.5 |
| Direction of rotation | Clockwise (Looking from flywheel end side) |
| Maximum speed (rpm) | 2000 |
| Min. idle speed (rpm) | 750 |
| Min. operating speed (rpm) | 1200 |

| | |
|---|---|
| Fuel timing for std. engine Valve timing BTDC (°) | 23 |
| Inlet opens BTDC (°) | 4.5 |
| Inlet closes ABDC (°) | 35.5 |
| Exhaust opens BBDC (°)<br>Exhaust closes ATDC (°) | 35.5<br>4.5 |
| Valve clearance Inlet (mm) | 0.18 |
| Valve clearance Exhaust (mm) | 0.20 |
| Lubricating system | Forced feed system |
| Power rating<br>1. Continuous (hp/rpm)<br>2. Intermittent (hp/rpm) | <br>7/1500<br>7.7/1500 |
| Brake mean effective Pressure at 1500 rpm (kg/cm$^2$) | 6.35 |
| Lubricating oil pump | Gear type |
| Lubricating oil pump delivery (liter/min) | 6.50 |
| Sump capacity (liter) | 2.70 |
| Lubrication Oil consumption | 1.5% normally exceed of fuel |
| Connecting rod length (mm) | 234 |
| Overall dimensions (LxBxH) (mm) | 617x504x877 |
| Weight (kg) | 160 |

# Annexure-C

**Uncertainty Analysis**

The error in experimental data of various quantities and propagation of these errors are discussed in this section. The method of estimating uncertainty in the experimental results has been present by Kline and McClintok [98]. The method is based on the specifications of the uncertainties in the primary experimental measurements. These errors are systematic errors, which are dependent on the accuracy of the measuring instruments. Consider an experiment result P is a given function of independent variables $x_1, x_2, x_3, x_4 \ldots \ldots . x_n$. Thus

$$P = P(x_1, x_2, x_3, x_4 \ldots \ldots . x_n)$$

Let $\Delta P$ be the uncertainty in the results of the experiment and $\Delta P_1, \Delta P_2, \Delta P_3, \Delta P_4, \ldots \ldots, \Delta P_n$ be the uncertainties in the independent variables. If the uncertainties in the independent variables are all given with the same odds, then the uncertainties in the results having these odds as follows

$$\Delta P = \sqrt{\left(\frac{\partial P}{\partial x_1}\Delta P_1\right)^2 + \left(\frac{\partial P}{\partial x_2}\Delta P_2\right)^2 + \left(\frac{\partial P}{\partial x_3}\Delta P_3\right)^2 + \left(\frac{\partial P}{\partial x_4}\Delta P_4\right)^2 + \cdots .. \left(\frac{\partial P}{\partial x_n}\Delta P_n\right)^2}$$

The estimated relative errors of each measured independent variable for both mode*s* of engine operation are given in Table. *T*he overall measurement error of the performance parameters *is* given in *the* table.

For example, the uncertainties associated with the measurement of BTE under diesel mode is as follow

$$BP = \frac{2*\pi*N*W*R}{60000}\ , kW \qquad \text{(B1)}$$

The independent var*i*ables for measurement of BP are N, w, and R are system constant

Under diesel mode

$$BTE = \frac{BP*3600*100}{\dot{m}_d*LHV_d} \qquad \text{(B2)}$$

Hence independent variable are N, W, $\dot{m}_d, and\ LHV_d$

The errors in measurement of N, W, $\dot{m}_d, and\ LHV_d$ are *0.1%, 0.5%, 1%, and 1% re*spectively.

The uncertainty in the measurement of BTE of diesel mode

$$\Delta \text{BTE} = \sqrt{(0.01^2 + 0.005^2 + 0.01^2 + 0.01^2)} \qquad \text{(B3)}$$

=0.018

=1.8%

The uncertainty in the measurement of BTE of dual fuel mode

$$\mathrm{BTE} = \frac{\mathrm{BP} * 3600 * 100}{(\dot{\mathrm{m}}_{\mathrm{d}} * \mathrm{LHV}_{\mathrm{d}}) + (\dot{\mathrm{m}}_{\mathrm{g}} * \mathrm{LHV}_{\mathrm{g}})}$$

Where:

$LHV_d$=lower calorific value of diesel
$LHV_g$=Lower calorific value of gas

Hence independent variable are N, W, $\dot{m}_d$, $LHV_d$, $\dot{m}_g$, and $LHV_g$

The errors in measurement of N, W, $\dot{m}_d$, and $LHV_d$ are 0.1%, 0.5%, 1%, 1%, 2%, and 1.5% respectively.

$$\Delta \mathrm{BTE} = \sqrt{(0.01^2 + 0.005^2 + 0.01^2 + 0.01^2 + 0.02^2 + 0.015^2)} \qquad \text{(B4)}$$

=0.030
=3.0%

**Table. 1 Relative error of independent variables**

| Independent variable | Relative error (%) |
|---|---|
| Engine speed (N) | 0.1 |
| Engine load (W) | 0.5 |
| Liquid fuel flow rate ($\dot{m}_d$) | 1 |
| Gas flow rate ($\dot{m}_g$) | 2 |
| LHV of liquid fuel ($LHV_d$) | 1 |
| LHV of gaseous fuel ($LHV_g$) | 1.5 |
| Cylinder pressure (P) | 2 |

**Table 2. Overall measurement error for performance parameters**

| Performance parameter | Diesel mode error (%) | Dual fuel mode error (%) |
|---|---|---|
| BP (kW) | 0.7 | 0.7 |
| BMEP (bar) | 0.85 | 0.85 |
| BSEC (MJ/kWhr) | 1.5 | 1.5 |
| BTE (%) | 1.6 | 3.0 |
| Volumetric efficiency (%) | 0.7 | 0.7 |
| Air flow rate (kg/s) | 0.5 | 0.5 |
| Air-fuel ratio | 1 | 2.3 |
| Liquid fuel substitution (%) | -- | 1.4 |
| Net heat release rate (J/deg | 2 | 2 |
| Noise level (dB(A)) | 0.5 | 0.5 |