Utilization of Biogas with Different Pilot Fuels in Dual Fuel Mode in Diesel Engine

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By

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DECLARATION OF ORIGINALITY

I GAURAV VERMA, Roll No. 2K20/THE/10 student of M. Tech (Thermal Engineering), hereby declare that the project Dissertation titled "**Biogas Utilization with Different Pilot Fuel in Diesel Engine"** is submitted by me to the department of mechanical engineering, Delhi technological university, Delhi in partial fulfillment of the requirement for the award of the degree of masters of technology, is original and not copied from any source without proper citation. This work is not previously formed the basis for the award of any degree, diploma associateship, followship or other similar title or recognition.

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This is to certify that the work presented in this dissertation "***Biogas utilization with different pilot fuel in diesel engine***" *by Gaurav Verma*, Roll No. 2K20/THE/10, is a record of original research carried out by him under my supervision and guidance in partial fulfilment of the requirements for the degree of *Master of Technology* (*Thermal Engineering)*. To the best of my knowledge this work has not been submitted in part or full for any degree or diploma for this university or elsewhere.

Supervisor
PROF. AMIT PAL

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For any errors or inadequacies that may remain in this work, of course, the responsibility is entirely my own.

GAURAV VERMA

ABSTRACT

Industrial dependence on energy and international conflicts has triggered the fuel prices to new highs. As a result, new alternative fuels must be explored that have no or very less harmful emissions without the compromise of the efficiency. One such fuel that can be converted for use in internal combustion engines is biogas. In this experimental study, a Direct Injection Compression Ignition (DICI) engine is converted into a dual-fuel engine that runs on biogas as primary fuel and various fuel blends as the pilot fuel.

It will achieve the goal of lower exhaust emissions although the thermal efficiency remains nearby same. Diesel, biodiesel, alcohol and their blends are used as the pilot fuel, whereas biogas-air mixture is injected through intake manifold. Engine air intake system is upgraded to allow the combination of air and biogas to mix thoroughly before being fed to the cylinder. The engine performance and emission characteristics are measured and compared with conventional diesel engine. The findings of this investigation revealed that a DICI engine can efficiently be converted into a dual-fuel engine that runs on both diesel and biogas.

Brake thermal efficiency decreases from 33.23% (in diesel mode) to 18.86% in dual fuel mode due to the lower calorific value of biogas compared to diesel. Whereas, the exhaust emissions like HC, CO, and CO_2 are also measured and found a increment in their percentage but for NOx percentage was reduced r. Further, the results shows that the ratio 80:20 (i.e., 20% SBDO and 80% biogas) can be used in diesel engine with a very little reduction in engine performance; thus, saving about 80% of the conventional diesel fuel. Therefore, the engine performance characteristics reveal that a considerable amount of diesel can be saved with addition of biogas in dual-fuel mode.

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NOMENCLATURE

BTDC	Before Top Dead Center
BSEC	Brake Specific Energy Consumption
HC	Hydrocarbon
BP	Brake Power (kW)
CI	Compression Ignition
CO_2	Carbon Dioxide
CO	Carbon Monoxide
CR	Compression Ratio
LHV	Lower Heating Value (MJ/kg)
IT	Injection Timing
RBB	Rice Bran Biodiesel
SBD	Soy Bean Biodiesel
SBDE	Soybean Biodiesel-Ethanol
RBBE	Rice Bran Biodiesel-Ethanol
SBDO	Soybean Biodiesel-Octanol
RBBO	Rice Bran Biodiesel-Octanol
NO_x	Oxides of Nitrogen(ppm)
$\dot{m}$ d	mass flow rate of diesel in Kg/sec
$\dot{m}$ pf	mass flow rate of pilot fuel(diesel) in Kg/sec
$\dot{m}$ g	biogas Flow rate in Kg/sec
$\dot{m}$ a	mass flow rate of air in Kg/sec
$\dot{m}$ g	mass flow rate of biogas in Kg/sec
A_i	Cross Sectional Area of Intake
A_2	Area of Nozzle

C_i	Intake Area
CFD	Computational Fluid Dynamics
D_B	Bore Diameter
D_i	Diameter of Inlet Manifold
d_2	diameter of nozzle
F_c	Vol. Flow of Fuel
fc_1	vol. flow of biogas
K	No. of Cylinder
L	Stroke Length
LCV	Lower Calorific Value
N	Speed
P	Power
Rpm	Revolution per minute
S (%)	Substitution of Diesel by Biogas
SFC	Specific Fuel Consumption
Vs	Swept volume
V_{air}	Volumetric Air Intake
η_{vol} (%)	vol. efficiency
θ1	Converging Angle of Venturi
θ2	Diverging Angle of Venturi
Θn	Nozzle Angle with Horizontal
Β	(Diameter of Throat)/ (Diameter of Inlet Tube)
LFR	Liquid Fuel Replacement
EGT	Exhaust Gas Temperature

Chapter 1

INTRODUCTION

1.1 World Energy Scenario

Issues identified with the lack of fossil fuel have been perceived for quite a while. An overwhelming reliance upon this vitality source similar to the primary if not by any means the only vitality bearer in transportation and power era for a long time has created concerns and activated broad research into alternative fuels. The worry of the low non-renewable energy source holds began a long time back when a hypothesis about top production of oil was presented. Despite the fact that the indicators used to predict topmost oil production have been upgraded by include various new variables, most experts and organizations agree that topmost production of oil will occur before 2020 [1].

After some time, fuel costs are fundamentally unstable, as appeared in Figure 1.1. Aside from the fast rate of vitality request, the pattern of the raw petroleum cost is likewise delicate to many components for example, political reasons, money variance, and supply [2].

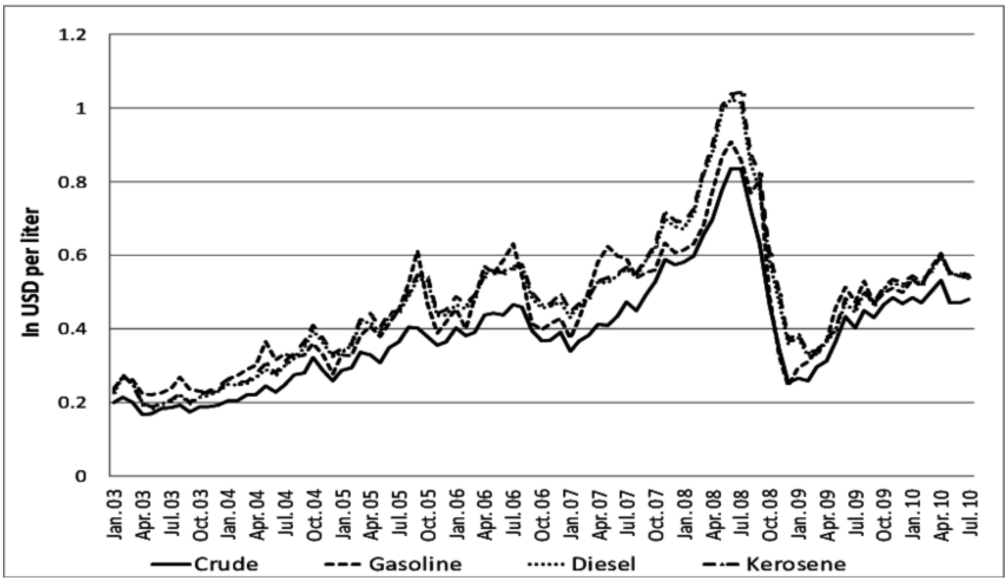

Figure 1.1 International fuel prices from 2003-2010[3]

Notwithstanding fuel security, there is another variable that has urged an endeavor to supplant the entire or part of customary petroleum derivatives utilized as a part of engine vehicles. This is on the grounds that consuming petroleum derivative produces destructive outflows (counting CO2) for people in general and causes unfriendly impacts on the earth. Outflows of CO_2 delivered from the petroleum products burning even contribute fundamentally to global warming. Along these lines, sustainable fuel advancement and fuel security can be enhanced with the substitution of non-renewable energy sources renewable and alternative fuels, both gaseous and liquid, are being used. Further fuel investment funds in transportation ought to be enhanced by improving motor innovation keeping in vision the final aim to likewise have the capacity to use gaseous fuels. Be that as it may, their usage in diesel motors still has disadvantages as far as engine performance and [3] [4]. The low energy content and low cetane number (CN) of most vaporous energies cause this.

To address the difficulties depicted above, research work has been done which means to investigate some alternative fuels (renewable and non-renewable), both gaseous and liquid fuels, to supplant part or all of regular fuels to be utilized in diesel engines.

1.2 Alternative Fuels for IC Engines

There are so many fuels are available for IC engine like methanol, ethanol and biodiesel. Where Petrol and diesel are the so common. Liquified Petroleum Gas (LPG) is also commonly used in IC engines, particularly in the food processing and manufacturing industries, while hydrogen, syngas, and biogas are alternate gaseous fuels for IC engines [5].

1.2.1 Biodiesel

Usually, biogas is utilized for methyl or ethyl esters of the unsaturated fats in characteristic oils and fats that meet the details for their utilization in the CI engines. Straight vegetable oils are not measured as biodiesel despite the fact that endeavors have been made to operate these also in the CI engines. Biodiesel is ordinarily formed by a response of vegetable oils or animal fats with alcohol, for example, methanol or ethanol within the sight of catalyst to yield mono alkyl esters. Glycerin is obtained as a by-product, which is removed and can be used in personal care products or a variety of chemical applications [6].

Biodiesel feedstocks are divided among crops, such as soybean and rapeseed, among others. Their findings also included oil trees such as palm oil. They pointed out that these oils are biodegradable and harmless because they come from crops or animal fat. Biodiesel may not be a particularly reliable fuel in the face of climatic change and soil degradation because its feedstock is highly dependent on climate and local soil conditions. Because some of the plants used as biodiesel feedstock take a long time to mature, biogas is a more reliable bio-fuel than biodiesel [7].

1.2.2 Biogas

Gaseous fuel biogas is produced by produced fermenting organic waste, we can get it from more other sources like organic residue of agriculture, animal manure, derivative of biodiesel or ethanol production, waste water consisting organic compounds like food processing and industrial and municipal wastewater [8]. The process is known as anaerobic fermentation. Basically, four main steps involved in anaerobic fermentation: hydrolysis, acidogenesis, acetogenesis and methanogesis. Because of its environmentally favorable character, it is considered an alternative green resource of energy for CI engines [9].

Sustainable power source and its transformation to warmth are important variables for maintainable advancement in nations with low or we can say no oil fuel saves. Biogas, which is mostly composed of methane (29-71 percent by volume) and carbon dioxide (21-41 percent by volume) and that's a very reliable alternative fuel for IC engines since it is renewable and environmentally beneficial, is one of the most important inexhaustible energies [10].

Using biogas in engines, when contrasted with petroleum derivatives keeps away from any extra ozone depleting substance emanation. Because of the organic nature of the parts of biogas, consuming for power generation unit in a gas engine an indistinguishable measure of CO_2 into the air from was initially caught up during the time spent photosynthesis in the regular CO_2 [10].

1.3 Dual Fuel Concept

The CI engine can be re fixed with a dual fuel system. The dual fuel engine uses a CI engine but has characteristics of both SI and CI engines in its combustion process. Air and high-octane index fuel mixed with carburetor is worked as same as a regular diesel engine in a dual fuel diesel engine. Due to the excessive auto ignition temperature, the compressed air and vaporous fuel mixture does not auto ignite. As a result, it is released by a jet of liquid fuel, which spontaneously ignites the charge as compression nears its finish. The gas–air mixture in the vicinity of the injected spray ignites at many places, resulting in multiple flame fronts [11]. As a result, the combustion process gets off to a quick start. The amount of liquid fuel required for competent ignition is between 10percent and 20percent of what is required for operating only on liquid fuel [12]. Dual Fuel combustion is an effective technique for decreasing toxin emanations, particularly soot and NOx, from direct injection diesel motors. The drawback of this concept is that it uses a flammability difference of the utilized fuels. The downside is that liquid fuel must be accessible for dual fuel engine operation [13]. Thus, the dual fuel concept can be applied for extensive utilization of the feasible accessibility of gaseous and liquid fuels.

1.3.1 Dual Fuel Advantages and Applications

Reduced PM and NOx emissions, as well as cheaper fuel costs due to natural gas's lower cost than diesel, are two significant benefits of the dual fuel gas-diesel engine. Load dependent thermal efficiency can be increased and fuel transport impact will be reduced if natural gas is accessible provincially are two other potential influences. The cost of maintenance will not increase after the conversion because the majority of the engine parts will remain the same. Carbonization is reduced in the dual fuel engine because a substantial amount of diesel is supplanted by natural gas, up to 70percent. Transitioning between diesel and dual fuel modes in a dual fuel natural gas diesel engine can be done while the engine is running without interrupting the required engine load. If natural gas is not obtainable, the control valve is off, and the engine runs on diesel fuel as same as traditional diesel engine.

Natural gas-powered diesel engines are used in everything from on-site power generation to transportation vehicles. Land-based oil, well-drilling, and fracking applications have all exploited dual fuel technology. Engine’s which can drive the hydraulic fracturing pumps use dual fuel, they are generally 100 percent diesel.

1.4 Effects and standards of Exhaust emission

Figure 1.2 offers a breakdown of each country's share of worldwide CO2 emissions from fuel combustion due to energy use in 2015. The USA contributes 13.9% of global CO2 emissions while the rest of the world (29%), China (25.9%), European Union (9.3%), India (6.4%), Russian Federation (4.9%) and Japan (3%). Consequences of greenhouse gas and other pollutants are extremely concerning. These emissions can result in harmful pitch of air pollution, global climate trade, acid rain and respiratory muddle [14].

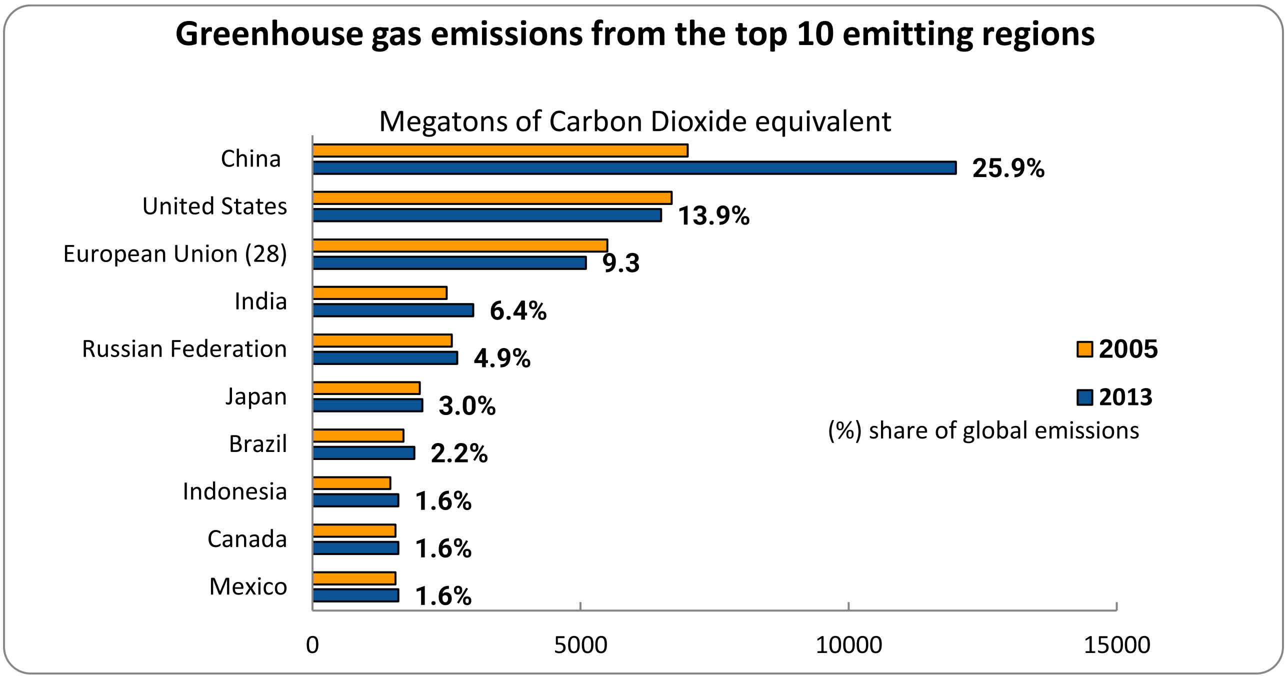

Figure 1.2: Global CO_2 emissions from fossil fuel burning (in megatons of CO_2 equivalent)

Short-term and repetitive exposure to these harmful pollutants has acute health impacts, whereas long-term and recurrent exposure has chronic health effects. As a result, it's critical to keep air pollution under control. Engine design and control of engine settings are one technique to reduce emissions, however this often has unintended consequences. Engine design and control alone cannot reduce PM emissions to an acceptable level in a CI engine. It usually necessitates the use of an after-treatment system to bring emission pitch beneath regulatory restrictions. Light-duty, heavy-duty, and stationary engine’s emissions are regulated and controlled through emission limits.

The cost of fuel is another challenge for CI engine manufacturers and operators. Generally, the cost is high for diesel fuel and it increase because of hike in demand of diesel fuel and especially in Europe, USA and the China other distillate oil is increased [15]. Because alternative fuels, such as natural gas, are much cheaper, using natural gas in diesel operations could save money.

Many have proposed different ways to blend cleaner and cheaper cost gaseous fuel with diesel fuel as a diesel fuel supplement in an effort to turn down undesired emissions and high-price diesel fuel costs. Emission may be divided into two categories: visible emissions and invisible emissions [11]. CO_2, CO, NO, unburned hydrocarbons, and aldehydes are categorized as visible emissions. They are basically construct to meet air quality standards and thereby preserve human health. The principal measure to control automobile air pollution was implemented in the United States in 1964. From that point forward there have been an expansive of quantities of changes in types of measures were actualized with the view to ensure the earth. As of now, BS-6 emission standards are executed in the vast majority of nations over the globe as demonstrated in Table 1.1 [16].

Table 1.1 Indian Standards for passenger cars, g/km

NORMS	**YEAR**	**CO**	**HC+NO$_X$**
1991	-	14.3-27.1	2.0
1996	-	8.68-12.40	3.0-4.36
1998	-	4.34-6.20	1.50-2.18
INDIA 2000	2000	2.72	0.97
BS-2	-	2.2	0.5
BS-3	2005	2.3	0.37
BS-4	2009	1.0	0.18
BS-5	2011	1.0	0.16
BS-6	2014	1.0	0.170

1.5 Problem Statement

By the time we used diesel fuel with CI engine all the particular matter, NO, CO, UHCs and CO_2 are released. Oxygen concentration, combustion temperature, and fuel type all influence the number of pollutants emitted. These contaminants have negative health and environmental consequences. Exposure to these emissions may cause health problems that lower labor productivity, resulting in an increase in poverty. Furthermore, pollution accumulation leads to an increase in greenhouse gas emissions, which causes global warming. These effects, together with the constraints of rising fuel prices due to increased demand, necessitate the production and use of alternative fuels like biogas [10].

Biogas use in CI engines needs a mechanism to admit the gas into the engine combustion chamber as well as a way to regulate the pilot fuel, implying that biogas cannot be used without modification in a diesel engine. These challenges emphasize the importance of developing a dual-fuel engine that employs biogas as an alternative fuel. Compression ignition engine manufacturers and operators are concerned about rising fuel costs and increasingly rigorous emission standards. Lower fuel costs and emissions can be accomplished by changing the CI unit to dual fuel operation and using lower cost natural gas. However, the economic savings from eliminating diesel and lowering emissions must be evaluated. As a result, the primary goal of this study is to overcome the issues associated with dual fuel engines in order to increase efficiency and improve exhaust pollution control. Because dual fuel engines emit more HC and CO than the standards allow, solutions to better control combustion and emissions should be developed. As a result, the following are the research's main problems and theories regarding the questions:

1) Does dual fuel operation save money compared to diesel operation?

Hypothesis: Fuel prices of natural gas have stayed significantly lower than diesel fuel prices over the last five years and have been more constant. When we put natural gas in a diesel engine, it is expected to supersede some of the diesel fuel in the CI engine, lowering the dual fuel engine's fuel costs.

2) Do dual-fuel engines produce excessive amounts of HC and CO?

Hypothesis: The proportion of natural gas to air that enters the cylinder is not controlled. The natural gas air mixture is thought to be below the flammability limit in the cylinder under these load settings, and the diesel fuel jet penetration does not extend throughout the cylinder.

As a result, natural gas and air combinations below the flammability limit near the combustion chamber's edge are either unburned (HC emission) or partially burned (CO emission).

1.6 Research Objectives

The major purpose of this research was to convert a diesel engine to run on biogas with multiple fuels diesel and blends as a dual-fuel engine, with diesel as the pilot fuel and biogas as the primary fuel, with optimum thermal efficiency and minimal gaseous emissions.

The following precise purpose had to be met in order to attain these goals:

1. For the dual fuel engine, design and develop an effective biogas and air mixing system.
2. To compare and contrast the updated dual fuel engine's performance with that of the diesel engine.

1.7 Dissertation layout

The thesis has been organized by focusing the concentration towards the clean and efficient power production from a biogas run dual fuel diesel engine using different types of driver fuel.

Chapter 1 offers the motivation acquired towards the use of renewable fuel. This is followed by dicussing alternative fuels, standards of Exhaust emission, problem statement and aim of the research.

Chapter 2 Past literature related to research in the field of Dual Fuel is reviewed. It briefly also discusses the scope and justification of present study.

Chapter 3 discussed the VCR engine setup used and the instruments used for measurements. This is followed by discussion regarding the modification carried out to make it suitable for dual fuel operations and the procedure carried out to perform the experiments.

Chapter 4 briefs the selection of fuel and also discussed the results obtain from various fuel mode.

Chapter 5 gives the dissertation conclusions.

Chapter 6 suggests the scope of future work.

Chapter 2

LITERATURE SURVEY

Overview

In the starting time of development of IC engine was based on employing gaseous fuel. At that time, the wide spread use of coal gas, produced by processing coal lead to the evolution of ‘Dual Fuel Engines’. It was further transformed in countries including Italy, Russia, Germany, Holland, the United States of America, and the United Kingdom prior to World War II. Because excellent liquid fuels were few during World War II, there was a lot of interest in using gaseous fuel engines, especially in Germany. The recent fossil fuel crisis and the rise of pollution have once again initiated active research in dual fuel engines using different test fuels. Nowadays, dual fuel engines are widely employed especially for stationary power generation applications. Biogas, a promising renewable fuel, has a great potential for dual fuel application. This chapter is dedicated towards a detailed literature review on biogas run dual fuel diesel engines. Besides, a discussion on thermodynamic potential study on diesel and dual fuel engines is incorporated. Finally, the aims outlined in the previous chapter are supported by the literature review's identification of major scopes of activity.

Chapter Outline:

2.1 Dual Fuel Diesel Engine

During the second half of 19th century, many divergent configurations of internal combustion (IC) engines were built and tested. These kinds of engines operated with varying degrees of success and fidelity using different mechanical system and engine-cycles. The first practical engine was invented by J. J. E. Lenoir in 1860 [17]. It was a single cylinder, two-stroke, double acting horizontal engine having a mechanical efficiency up to 5%. The next significant milestone achieved was the Otto and Lagen atmospheric or free piston engine in 1866.

The key characteristic of this engine was that it used half the amount of fuel as the Lenoir engine. Around the same time, the commercial exploitation of oil well has stated in USA. This led to the availability of liquid fuels. In 1876, the Otto silent engine commonly known as Spark Ignition (SI) engine based on four stroke cycle was patented and produced. Akroyd Stuart and Rudolf Diesel collaborated on the development of contemporary compression ignition (CI). The engine of Akroyd Stuart was patented in 1890 and manufactured in 1892. It was effectively a three-to-one compression ratio 4-stroke compression ignition engine.

This engine's compression ratio was too low to allow for spontaneous fuel ignition during compression. This required an external heating system. Later on, Rudolf Diesel found a way to eliminate this limitation. Diesel's notion of compressing air to the point where fuel would spontaneously ignite after injection was published and copyrighted in 1890. The prototype had a 25 percent efficiency, which is roughly double that of any modern power plant [18]. There has been evolution of a different technology in the early part of 20th century for running gaseous fuel with the aid of liquid fuel in CI engines. These types of engines are known as dual fuel diesel engines. Dual fuel technology is used for efficient combustion of gaseous fuel like biogas, hydrogen, natural gas, producer gas, syngas, butane, etc. which have having low ignition characteristics.

The earliest experiment performed on dual fuel diesel engines by Cave dates back to 1929 using hydrogen gas. The National Gas and Oil Engine Co. in Great Britain introduced the first commercial dual fuel engine that ran on town gas in 1939. The National Gas and Oil Engine Co. in Great Britain launched the first commercial dual fuel engine powered by town gas in 1939 [13].

The main driving force that initiated active research in dual fuel engines was due to shortage of liquid fuel during World War-II. During this period, the versality of dual fuel engines enabled them to be used for military as well as for civil applications. The popularity gained by dual fuel engines during World War-II plus the depletion of fossil fuels lead to their further development for different usage like power generation, on–road and off- road applications.

2.2 Working Principle

Garnier *et al.* [19] defined that dual fuel engine is an ideal multi-fuel engine that can operate on a long range of fuels while also having the ability to operate as a traditional diesel engine. During dual fuel operation, a carbureted combination of air and gaseous fuel with a high-octane index is sucked and compressed in the same way air is sucked in a regular CI engine. Because of the poor ignition quality of the gaseous fuel, the mixture of air and fuel gas fails to auto ignite in the compressor.

As a result, a little amount of liquid fuel, known as a driver fuel, is injected and ignites spontaneously at the end of the compression phase. As a result, a dual fuel engine consumes both primary and secondary fuel. The pilot fuel, such as diesel, is the primary fuel, while the secondary fuels are natural gas (NG), liquefied petroleum gas (LPG), hydrogen, or biogas [20].

In 2006, J. Stewart *et al.* [21] researchers transformed a direct-injection CI engine into a dual fuel engine and fed it three different gaseous fuels: methane, propane, and butane, to test performance at varied gaseous concentrations.

They reasoned that the most cost-effective and uncomplicated approach of admitting a gaseous fuel to the dual fuel engine is through a simple central point mixing mechanism. To ensure thorough mixing of air and fuel, a simple venturi type gas mixer was added upstream of the inlet manifold at a distance of 10 times the gas pipe diameter. Engine performance data was gathered at three loads equivalent to 1/4, 1/2 and 3/4 load under steady state running circumstances (relative to 100 percent loading being 18.7 kW). They came to the conclusion that propane showed the most promise as a dual fuel engine fuel, with a reduction in energy usage of up to 20%. This was attributable to the fuel's increased reactivity. The preferred fuel, methane, was discovered to increase brake specific energy consumption in all circumstances studied. According to the study, the engine clearly demonstrated the benefits of lower CO_2 emissions, which must be taken into account if targeted global reductions in the transportation sector are to be met.

CO_2 emissions have been decreased by up to 20%. This demonstrated that dual fuel engines can be used to reduce exhaust pollutants. Modern fuel injection equipment, such as high-pressure common-rail and multiple injection systems, may also aid increase engine performance and minimize CO_2 emissions, according to the study. Dual fuel engines have been found to operate within the International Maritime Operations Tier 3 regulation's NO_X and SO_X constraints, giving them a viable alternative to diesel engines with EGR or exhaust gas after-treatment [22].

According to a comprehensive simulation study by C. Christen *et al.* [22], To achieve minimal NO_X emissions and reduce the risk of knocking combustion, a homogenous combination of fuel gas and air is required. Continuous fuel gas entry produced the best mixing outcomes. Rich fuel gas accumulated in cylinder fissures as a result of high gas admission, which was a major source of unburned hydrocarbons.

2.3 Engine Modifications

By making the following changes to a standard diesel engine, it can easily be converted to a dual fuel diesel engine:

1. A gas mixer needs to be connected at the inlet manifold for mixing of gaseous fuel and air [23]. The gas mixer should provide a homogeneous mixture of air and gaseous fuel taking into account of the appropriate air-fuel mixture.
2. A control mechanism to be installed for maintaining the desired fuel setting on the injection pump [23].

2.4 Biogas Run Dual Fuel Diesel Engine

Biogas run dual fuel diesel engine biogas, a renewable gaseous fuel, can be successfully used in CI engines with minor modifications. This is especially true in emerging countries, where meeting the rising demand for fossil fuels is a serious concern [24].

On compressed natural gas (CNG), a biogas enriched with methane, and biogas, the overall performance of an IC engine having constant speed was evaluated by R. Chandra et al. [25]. The gaseous fuel was fed into the engine using a venturi type air intake and fuel gas supply mechanism. In case of natural gas (compressed), biogas enriched with methane, and raw biogas, the loss observed in brake power owing to conversion of a diesel engine to that of a spark ignition engine was observed to be 31.8 percent, 35.6 percent, and 46.3 percent, respectively. The findings show that power losses were significant, with raw biogas suffering the greatest loss. Rather than using a venturi style air intake, a mixing chamber can provide a more homogeneous mixture of air and fuel, leading to increased combustion and performance of the engine. The major reason for this was a mixing chamber that allows air and fuel to stay within for longer, resulting in a more homogeneous mixture with improved ability of combustion.

In a detailed study by S. Siripornakarachai et al. [26], the diesel engine of a bus was converted to run on biogas to generate power at a farm. The biogas carburetor was added for air-fuel mixing, the spark ignition system was used instead of a fuel injection system, and the compression ratio was found to be reduced from 16:1 to 8:1 using a cylinder head spacer. The results revealed that CO emissions were considerable, despite the engine's adequate power output of 134.20 kW.

According to the literature, the compression ratio for biogas operation should be between 10:1 and 12:1 , [27]. Since the ratio of compression was reduced to 8:1, which could have led to incomplete combustion, therefore leading to a rise in CO emission.

In another study by H.S. Sorathia et al., The quantity and quality of the energy used in an engine having single-cylinder, direct injection in a diesel engine having petrol, diesel oil and biogas used as fuel were observed using the first and second laws of thermodynamics. Biogas had a maximum braking thermal efficiency of 27.50 percent vs 28.25 percent for diesel oil. Calculations revealed that heat transmission from the engine lost 7.31 percent and 6.48 percent of the fuel exergy input for diesel and diesel-biogas dual fuels. The study found that using a diesel-biogas fuel having a dual mode resulted in less energy conversion efficiency, which was somewhat mitigated by using biogas instead of diesel and induction air. The study also came to a conclusion that using biogas (premixed) charge dual fueling for the engine resulted in nearly no performance loss at all speeds during the test phases. The findings of this theoretical research in the field of thermodynamics are impressive and should be confirmed using experimental approaches. N.S. Ray et al. [28] modified an engine (CI) into a dual fuel engine and therefore used biogas for the partial substitution of diesel to study the factor of effect over the performance of the engine. The study also indicated good performance engines modified in terms of the output power as well as reduction of emissions from the exhaust. There was a significant reduction in the emissions of CO and HC, though the power of break engine decreased slightly with a slight increase in the biogas percentage. It was due to the reduced energy content of biogas as compared to that of diesel. The conclusion was that, the 50% substitution of diesel with the biogas was the best (Optimum) ratio for the dual fuel operation. The highest ratio that was tested was Biogas: Diesel (50:50). therefor the requirement to investigate over the performance of the engine at higher substitution ratios such as Biogas: Diesel in the ratio of 90:10. As a result, biogas-powered dual-fuel engines have undergone constant development over time. Many studies have investigated the performance and emission characteristics of various types of biogas and pilot fuels, as shown in Table 2.1.

2.5 Combustion Characteristics

The study carried out by Mansour *et al.* [29] suggests that dual fuel combustion is significantly more complicated than pure diesel operating. Cylinder pressure data analysis is an important technique for troubleshooting engine combustion behaviour since cylinder pressure history has a direct impact on engine performance and emission characteristics. The combustion processes in CI engine comprises of four phases as indicated in Fig. 2.1. They are as follows: AB is the ignition delay, BC is the uncontrolled combustion, CD is the controlled combustion and DE is the late combustion. Point 'A' is the start of injection whereas point 'B' is the point of initiation of combustion. However, in case of dual fuel engines, it comprises of five stages. The pilot fuel ignition delay is AB, the driver premixed combustion phase is BC, the primary fuel ignition delay is CD, the primary fuel quick combustion is DE, and the diffusion combustion phase is EF. The critical analysis of *P-θ* diagram diagrams reveals the following points:

1. When the cylinder pressure is on peak of a diesel mode found to be greater than the dual fuel mode.
2. The pressure rise rate of diesel mode is getting to be decreasing than that of dual fuel mode.
3. The shift of highest cylinder pressure away from top dead centre (TDC) is higher in case of dual fuel mode in comparison to that of diesel mode.

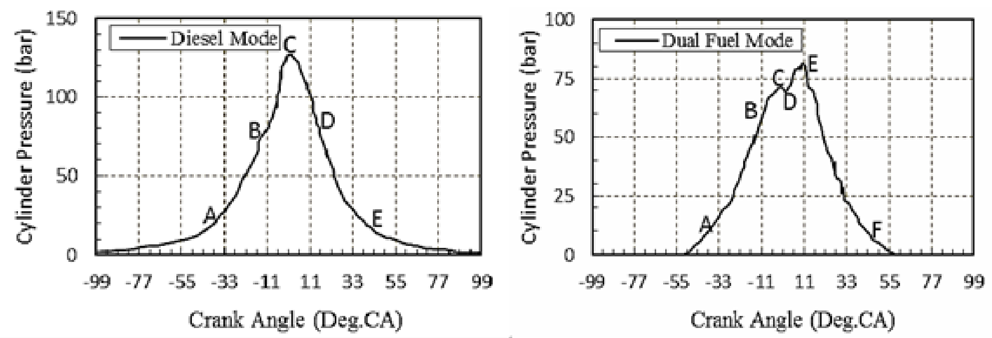

Fig. 2.1 Comparison of *P-θ* diagram for diesel and dual fuel mode (Mansour *et al.*, 2001)

The ignition delay of pilot fuel in case of dual fuel mode is longer than that of pure diesel operation. This is due to induction of large amount of gaseous fuel which reduces the amount of air and thereby, slowing down the combustion reaction.

2.6 Biogas composition

Basically, biogas contains methane (CH_4) and carbon dioxide (CO_2). The combustion value of any quality of biogas is directly related to its methane composition [30]. However, presence of CO_2 up to 30% in biogas for dual fuel application can improved the performance of the engine. Biogas containing more than 40 % of CO_2 needs scrubbing. Conversely, biogas containing more than 45% of CO_2 results in harsh and irregular running of the engine [31]. The properties of biogas used by different researchers for biogas run dual fuel applications are given in Table 2.1.

Table 2.1 Summary of the test engine, type of biogas and pilot fuel investigated for biogas run dual fuel diesel engine

Pilot Fuel	**Type of biogas**	**Type of engine**	**References**	**Composition**
Diesel, Palm oil biodiesel	Simulated biogas	2-cylinder, naturally aspirated, DI and rated power of 20kW, compression ratio 15.5:1	Bedoya *et al. [42]*	60% CH_4, 40% CO_2
Diesel, Jatropaha oil	Simulated biogas	1-cylinder With rated power of 12kW and compression ratio 17:1	Luijten and Kerkhof [46]	65% CH_4, 35% CO_2
Diesel, Jatropha biodiesel	Raw biogas	1-cylinder, Natural aspirated, DI And rated power of 5kW With compression ratio 17.5	Sahoo *et al*. [19]	85% CH_4, 15% CO_2

Pilot Fuel	Type of biogas	Type of engine	References	Composition
Diesel, Soybean biodiesel	Raw biogas	4-cylinder, Turbo charged and rated power of 46kW, compression ratio 19:1	Yoon and Lee [44]	30-73% CH_4, 20-40 CO_2, 5-40 N_2,1-3H_2
Diesel	Simulated biogas	1-cylinder DI, water cooled compression ratio 16.5:1	Mustafi *et al* [36]	39.9% CH_4, 60.1% CO_2
Diesel, Karanja biodiesel	Raw biogas	1-cylinder, DI, air cooled And rated power of 4.4kW With compression ratio 17.5	Barik and Murugan [45,48]	-

2.7 Effect of Different Parameters on the Performance of Biogas Run Dual Fuel Diesel Engines

On the performance, combustion, and emission characteristics of biogas run dual fuel diesel engines, the parameters include operating parameters, engine components, biogas quality, biogas inlet pressure, and oxygenated combustion. Studies on operating parameters like load and speed reflect the maneuvering capability of the engine. Investigations on engine components mainly include the air induction system and exhaust gas recirculation (EGR).

2.7.1 Brake Power, Torque

For both diesel and DFM, Sahoo shows that braking power (BP) and torque grow with increasing load at constant speed, demonstrating that BP and torque have a linear relationship with load [13].

2.7.2 Air-Fuel Ratio

Effect of Load:

From the standpoint of combustion and engine efficiency, the relative proportions of fuel and air are critical. This can be stated as a ratio of the fuel mass to the air mass or vice versa [32]. Equivalence ratio is the ratio of real fuel-air ratio to stoichiometric fuel-air ratio. SI engines have a 12 to 18 air-to-fuel ratio. The air fuel ratio for CI engines ranges from 18 to 80 from full load to no load. Dual fuel diesel engines, on the other hand, may operate over a wide variety of air-fuel ratios. The equivalency ratio in the DFM mode is larger than in the diesel mode, according to Yoon & Lee. This is because biogas is fed into the engine through the air intake manifold. This results in lowering of the volume of air entering into the combustion chamber [33]. Barik and Murugan found the stoichiometric air fuel ratio to decline with the rise of load. That’s because of the fact that as more amount of biogas flows needs to be supplied to the engine at higher load [34].

Effect of Air Induction System:

Bedoya *et al*. two induction systems were used to collate the performance of a biogas-fueled dual-fuel engine. The first one, SM1, comprises of a Kenics mixer and supercharger with a large mixing length (250 mm), while the second one, SM2, consists of a T-mixer and short mixing length (25 mm). The pilot used for SM1 and SM2 were palm oil biodiesel and diesel, respectively.

The tests indicated a lower air-fuel ratio found to be lower for SM1 in comparison to that of SM2[30].

Effect of Biogas Quality:

The effect of biogas quality on the performance of a dual fuel diesel engine was explored by Luiten and Kerkhof. The study inferred that air fuel ratio was hardly got affected by low the quality of biogas [35].

2.7.3 Volumetric

Efficiency Effect of Load:

For both lone and dual mode, Duc and Wattanavichien discovered that volumetric efficiency (VE) dropped as braking torque and load increased [36]. Barik and Murugan noticed a similar pattern with biogas-powered dual-fuel diesel engines employing Karanja methyl ester (KME) . This was owing to the fact that when load grows, the temperature of the exhaust gases rises, preheating the incoming air and lowering the VE. According to Barik and Murugan, the fall in VE was greater in DFM than in diesel mode. This is because under DFM, the biogas substitute displaces a greater amount of air [37]. For diesel mode, Sahoo reported that the maximum and minimum VEs found was about 85% and 81% at 0 and 100% loads, respectively. For the same loading conditions under DFM using diesel and jatropha biodiesel, these values were found to be 82% and 78%, and 82% and 77%, respectively [13].

Effect of Induction System:

Bedoya *et al*. [9] found that there was a 6% drop in VE using the combination of Kanins mixer and supercharger along with a longer mixing length as compared to that of a T-mixer for the same biogas run dual fuel diesel engine. At part load, the difference in VE between the two induction systems was observed to be less significant.

Effect of Biogas Quality:

Luiten and Kerkhof calculated VE as a function of the pilot fuel's heat release fraction and biogas quality. The VE is influenced by the quality of the biogas. Even for the lowest quality of biogas, the decline in VE was moderate, from 95 percent at zero substitution to 90–91 percent at maximal substitution.

2.7.4 Brake Thermal Efficiency

Both diesel and DFM brake thermal efficiency (BTE) was found to increase with load. In all of the test instances, however, the BTEs in DFM were shown to be lower than in diesel mode. This was primarily owing to biogas's lower calorific value when compared to diesel.

The study done by Yoon and Lee [33] and Sahoo [13] revealed that using biodiesel as pilot fuel resulted in a lower BTE than diesel under DFM. The review on BTE highlighted an important fact that it was possible for a biogas run dual fuel diesel engine to achieve a high BTE around 32% as reported by Yoon and Lee.

Effect of Air Induction System:

The test conducted by Bedoya *et al*. indicated that the combination of Kanins mix and supercharger along with a longer mixing length (SM2) produced a better BTE in comparison to that of T-mixer along with a shorter mixing length (SM1) for the same biogas run dual fuel diesel engine. The BTE increased by 8% for the SM2 system in comparison to that of SM1 system [30].

Effect of Biogas Quality:

Henham and Makkar (1998) did so many tests on diesel engine of dual fuel type using synthetic biogas of various compositions. The test indicated that the overall efficiency falls with the rise of CO2 content in the composition of synthetic biogas [38]. However, Luiten and Kerkhof found that the biogas quality has marginal effect on the BTE.

2.7.5 Brake Specific Fuel Consumption, Brake Specific Energy Consumption, Liquid Fuel Replacement and Biogas Flow Rate

Effect of Load:

According to the literature, both diesel and DFM brake specific fuel consumption (BSFC) and brake specific energy consumption (BSEC) decrease as load increases. The BSFC was found to be more under DFM in comparison to that of diesel mode for different loading conditions as observed by Yoon and Lee [33]. The BSFC of biodiesel-biogas was found to be lower than diesel-biogas under DFM in their investigation. In contrast, Sahoo [13] showed that the BSEC of diesel-biogas was lower than that of Jatropha-biogas. At 100 percent load, their investigation found a maximum liquid fuel replacement (LFR) of 69 percent and 66 percent for diesel and jatropha pilot fuel, respectively, with a biogas flow rate (BFR) of 3.66 m^3/h and 4 m^3/h. For KME-biogas run dual fuel diesel engine, Barik and Murugan found the

BSEC respectively be to be 15.9 MJ/kWh, 17.3 MJ/kWh, 19.2 MJ/kWh and 21.5 MJ/kWh for BFR of 0.3 kg/h, 0.6 kg/h, 0.9 kg/h and 1.2 kg/h at full load. The biogas energy share corresponding to these BFR values were found to be 11.8%, 21.8%, 30.2% and 37.4%, respectively [45]. In another study, Barik and Murugan found the BSFC for diesel mode to be 0.27 kg/kWh at full load. Using diesel as main fuel under DFM for same loading conditions, the BSFC is 0.32 kg/kWh, 0.37 kg/kWh, 0.43 kg/kWh and 0.49 kg/kWh for BFR of 0.3 kg/h, 0.6 kg/h, 0.9 kg/h and 1.2 kg/h, respectively as shown in Fig. 2.25. For the same loading condition and BFRs, the LFRs were found to be 0.101 kg/h, 0.145 kg/h, 0.215 kg/h and 0.223 kg/h[37].

Effect of Biogas Quality:

Bari observes that the trend of BSFC rises as CO_2 increases in the biogas composition. This is because as the CO_2 concentration in the biogas increases, CO_2 remains undissociated and acts as an inert gas. The addition of such inert gas alters the burning velocity, resulting in incomplete combustion and a higher BSFC and BFR.

In compared to diesel mode, BSFC was shown to be greater in DFM. At the same loading conditions, the BSFC increased as the quantity of CO_2 in the biogas composition was increased.

2.7.6 Exhaust Gas Temperature

Effect of Load:

For both diesel and DFM, the exhaust gas temperature (EGT) rises as the load increases. However, when comparing EGT in DFM to diesel mode, the results were quite different. Few In compared to diesel mode, studies showed a reduced EGT under DFM, whereas Sahoo observed a greater EGT under DFM [13]

Effect of Air Induction System:

Reported that the combination of Kanins mixer and supercharger along with a longer mixing length produced a lower EGT in comparison to that of T-mixer along with a shorter mixing length for the same biogas run dual fuel diesel engine. This was due to the design of SM2 system which increased the mixing turbulence and thereby, resulted in better extraction of energy in power stroke. This caused a drop in EGT [30].

Effect of Biogas Quality:

Henham and Makkar found that the EGT to be more affected by biogas substitution rather than biogas quality up to 45% replacement. At 58% replacement, EGT increased with the increase of CO_2 in biogas mixture [38].

2.7.7 Carbon Monoxide Emission

Effect of Load:

Yoon and Lee discovered that carbon monoxide (CO) emissions were higher in the DFM mode than in the diesel mode. This was due to incomplete combustion caused by the CO_2 in biogas diluting the charge and a lack of oxygen. As a result, the flame created in the pilot fuel's ignition region was generally repressed, and the biogas fuel air mixture did not advance until it reached a minimal limitation value for autoignition [33]. In comparison to diesel mode at 100 percent load, Sahoo found that biogas-diesel and biogas-jatropha biodiesel both reduced CO emissions by 50 percent and 16 percent, respectively.

At full load, Barik and Murugan found a 9 percent, 16.6%, 28.5 percent, and 37.5 percent increase in brake specific carbon monoxide (BSCO) emissions compared to KME operation for biogas energy shares of 11.8 percent, 21.8 percent, 30.2 percent, and 37.4 percent, respectively [45]. Furthermore, Barik and Murugan discovered that when biogas is used at a flow rate of 1.2 kg/h, CO emissions are around 24% greater than when diesel is used at full load [48].

Effect of Air Induction System:

Bedoya *et al*. found that the CO emissions were produced more in the exhaust by using SM2 in comparison to that of SM1. CO emissions with the SM2 system increased at 40% load due to an rise in partial oxidation events of CH4, a reduced main fuel quantity, and a greater total fuel–air equivalency ratio. This happened because the SM2 technology rise the reactivity of the biogas–air combination during the compression stroke while lowering the pilot fuel quantity. Because the reduction of unburned CH_4 and lower oxygen availability at higher equivalence ratios, CO emissions rose for both the SM1 and SM2 systems at higher loads. However, when compared to the SM1 system, the SM2 system allowed for a near-40% reduction in CO emissions at full load. Higher burning rates and oxygen availability in later stages of combustion were to blame [30].

Effect of Biogas Quality:

Mustafi *et al.,* observed a slower increase of CO emissions as the % of CO2 content rise in the fuel (Fig. 2.58). This may be due lowering of combustion efficiency as the amount of CH4 got reduced with the decrease of biogas quality [24].

2.7.8 Carbon Dioxide Emission

Effect of Load:

CO_2 emissions for both diesel and DFM are observed to rise as load increases. However, CO_2 emissions are higher in the DFM mode than in the diesel mode. This is due to the fact that biogas mostly comprises methane and CO_2. Yoon and Lee found that the combination of biogas-biodiesel emitted more CO_2 in comparison to biogas-biodiesel [33]. Conversely, Sahoo reported that the combination of biogas-diesel emitted more CO_2 emission than biogas–biodiesel especially at higher loads.

In another study, Barik and Murugan used KME as main fuel under DFM. The test revealed that as the biogas energy share increased, the brake specific carbon dioxide ($BSCO_2$) declined. Biogas energy shares of 11.7 percent, 21.7 percent, 30.01 percent, and 37.04 percent result in $BSCO_2$ drops of 25percent, 33percent, 52 percent, and 60%, respectively, as compared to KME at full load. The poorer volumetric efficiency and increased CO_2 in biogas were blamed for the reduction in $BSCO_2$ emissions under DFM. The same group of authors observed that the CO_2 emission rises with the rise in biogas flow rate for the same loading conditions using diesel as pilot fuel [37].

2.7.9 Hydrocarbon Emission

Effect of Load:

For both diesel and DFM, Yoon and Lee discovered that as load increased, HC emissions decreased. However, because the combustion chamber temperature was low, Sahoo noticed that HC emissions were higher at low loads. This diminishes gradually at normal loads before increasing at high loads . Barik and Murugan reported an increase of HC emission increases with load for high biogas flow rate.

Effect of Air Induction System:

Bedoya *et al.* reported that the methane emission dropped by 34% using the SM2 system in comparison to that of SM1 system at 40% load. Because the equivalency ratio was closer to the stoichiometric value for both the SM1 and SM2 systems at full load, the effect of dual fuel systems on methane emissions was less than at part load.

Effect of Biogas Quality:

This is because, as the CO_2 content of the biogas increases, the turbulent flame propagation from the main fuel's igniting zones is suppressed even more due to the low combustion chamber temperature and low air-fuel ratio.

2.7.10 Oxides of Nitrogen Emission

Effect of Load:

NO_X emissions are primarily determined by combustion chamber temperature, which is in turn determined by the applied load. As a result, both diesel and DFM emissions increase as load increases. In the case of dual fuel operations, however, NO_X emissions are lower. This is because of the fact the existence of carbon dioxide in biogas lowers the combustion temperature [11]. Yoon and Lee reported that diesel-biogas produced lower NO_X emission than soybean biodiesel- biogas for the same dual fuel engine [33]. In a different pilot fuel study, Sahoo set-up that diesel-biogas produced higher NO_X emission than jatropha biodiesel-biogas [38].

Barik and Murugan carried out a comparison of brake specific nitrous oxide (BSNO) emission for a biogas run dual fuel diesel engine for different biogas energy share. In comparison to KME in lone mode at full load, the results showed a reduction in BSNO emission of 24 percent, 29 percent, 34 percent, and 39.5 percent under DFM with a biogas energy contribution of 11.8 percent, 21.8 percent, 30.2 percent, and 37.4 percent, respectively. In another study, the same group of authors observed the NO emission to be lowers by 42.8% as compared to that of diesel for a BFR of 1.22 kg/h lowers at maximum load .

Effect of Biogas Quality:

As the amount of CO_2 in the biogas composition increased, NO_X emissions dropped by Mustafi *et al.* more gaseous fuel escaped the combustion process as CO_2 levels in biogas increased, affecting oxygen concentration and lowering overall cycle temperature. Therefore, NO_X formation reduced with the degradation of the biogas quality [24].

2.8 Scope of Work

The literature review suggests that there has been considerable research on biogas run dual fuel engines especially in the area of power production applications. Still, there are areas that need to explore to make biogas run dual fuel diesel engine more efficient. The following gaps are found out from exhaustive literature survey:

The standardization of the operational characteristics of a biogas run dual fuel diesel engine (such as compression ratio and pilot fuel injection timing) has not been reported. This is critical because the chemical and physical properties of diesel fuel are used to standardize the operating parameters of a diesel engine. Therefore, running biogas under dual fuel mode at standard setting of diesel will not be offer its top performance. Hence, the operating parameters need to be adjusted in order to obtain efficiency comparable to that diesel fuel.

No archival material mentions the use of intermix fuel as a main fuel for a biogas-powered dual-fuel diesel engine. This is important as the use of emulsified fuel as pilot fuel may further lower the NO_X and soot emissions.

It is unknown what influence compression ratio and injection timing have on the energy and exergy distribution of a biogas-fueled dual-fuel diesel engine. This is critical because the second law analysis calculates the highest achievable thermodynamic performance of the system by determining the availability drop or destruction in different areas of an engine.

2.9 Justification of Present Study

According to the preceding review, researchers have focused on the use both are liquid and gaseous bio-fuels in IC engines in an attempt to discover answers to environmental issues and the depletion of petroleum resources. Researchers have also tried using hydrogen, acetylene, and compressed natural gas (CNG) as a substitute to diesel and petrol in the IC engine. However, high fuel costs and increased NO_X emissions associated with these fuels continue to be a problem. Furthermore, due to their great flammability, some gases, such as acetylene, require specific treatment despite their excellent combustion qualities. Because of its environmentally favourable character, biogas, which is created by anaerobic fermentation of organic waste, has been reorganized as a potential spotless green energy resource for the CI engine.

Due of their potential to minimise smoke emissions while improving performance, dual fuel engines have piqued interest. This can be accomplished by properly designing the mixing device for air and gaseous fuel, resulting in increased thermal efficiency. Due to inadequate mixing of air and fuel gas for absolute combustion, the hurdles involved in converting a diesel engine to a dual fuel engine result in emissions of particulates and CO, as well as a loss of engine output.

It is clear from this review that more research is needed to find the best performance of a CI engine when running on biogas while reducing exhaust emissions as much as possible. Some studies have employed simulated biogas, which involves combining methane and carbon dioxide in specific amounts, and this may not accurately reflect engine performance on biogas. Though the output of nitrogen oxides from a dual fuel engine running on biogas is lower than that of a pure diesel engine, Exhaust Gas Recirculation (EGR) can be used to further reduce NO_X emissions. Most researchers haven't taken this into account. It's also crucial to look into how EGR affects engine performance characteristics like braking power, BSFC, and BTE.

Due to the problem of compact storage, using biogas as an alternative fuel for automobiles residue a barrier. However, as researchers work to solve this problem, stationary IC engines can be employed in power generation, construction site hauling, and pumps where biogas can be created and utilized at the place of production. The lack of locally available dual fuel engines that run on biogas and diesel provided a framework for converting a CI engine to a dual fuel engine that runs on biogas and diesel. The technique for engine modification and subsequent performance measurements is described in the next chapter.

Chapter 3

EXPERIMENTAL DESIGN AND METHODOLOGY

EXPERIMENTAL DESIGN AND METHODOLOGY

Overview

The conversion of a variable compression ratio engine to a dual fuel engine for diesel and biogas is discussed in this chapter. The engine was converted to run on biogas with the goal of preserving excellent thermal efficiency while lowering damaging emissions. The technique for designing and fabricating numerous parts, as well as the setup of the assessment displaying the engine and its additional part, are all displayed. Following that, the current chapter delves into the specifics of the engine, as well as the devices and tools utilize to conduct these trials. The discussion includes air and fuel flow measurement, P- θ measurement, temperature measurement, compression ratio variation control injection timing variation control and performance measurement. Further, the specifications along with the working principle of the flue gas analyzer are also described. Thereafter, the modifications carried out for converting the existing setup to biogas run dual fuel engine are briefly explained. Finally, the experimental procedure used for both diesel and dual-fuel mode is elaborately discussed.

Chapter Outline:

3.1 The VCR Engine Test Setup

As specified, the experimental arrangement includes a 3.50 kW single cylinder, 4-stroke, direct injection (DI), naturally aspirated (NA), water-cooled, variable compression ratio (VCR) diesel engine in Fig. 3.1. It is linked to an eddy current and water-cooled dynamometer for stacking on the crankshaft using electromagnetic driving. To vary the CR without put a stop to the motor or changing the burning chamber geometry, a tilting cylinder block positioning is used. The fuel injector in the motor features three circular holes with a diameter of 0.30 mm that spray gasoline with a 120o splash edge. The engine's piston top is of the bowl type. As a result, when the piston hits TDC, the combustion chamber is hemispherical. Gravity transports liquid gasoline from the fuel tank to the engine fuel pump. The electric supply for load variation is be in control by a regulator mounted on the panel box. The dynamometer's load sensor transmits the load signal to the digital display in kilogram. Along with the arrangement, there are instruments for measuring combustion pressure and crank angle. For each of the 360° rotations of the crank, the signals are interfaced to the computer via an engine indicator for pressure-crank angle (P-) and pressure volume (P-V) diagrams. Airflow, fuel flow, and temperature measurements are all possible. Rotameters are used to measure the amount of cooling water and calorimeter water flow. Cooling water circulates through the engine block and cylinder head jackets to remove excess heat generated during combustion. The pressure of the fuel injection can be adjusted between 200 and 220 bar. The engine's specifications are listed in Table 3.1. Table 3.1 includes the engine's specifications.

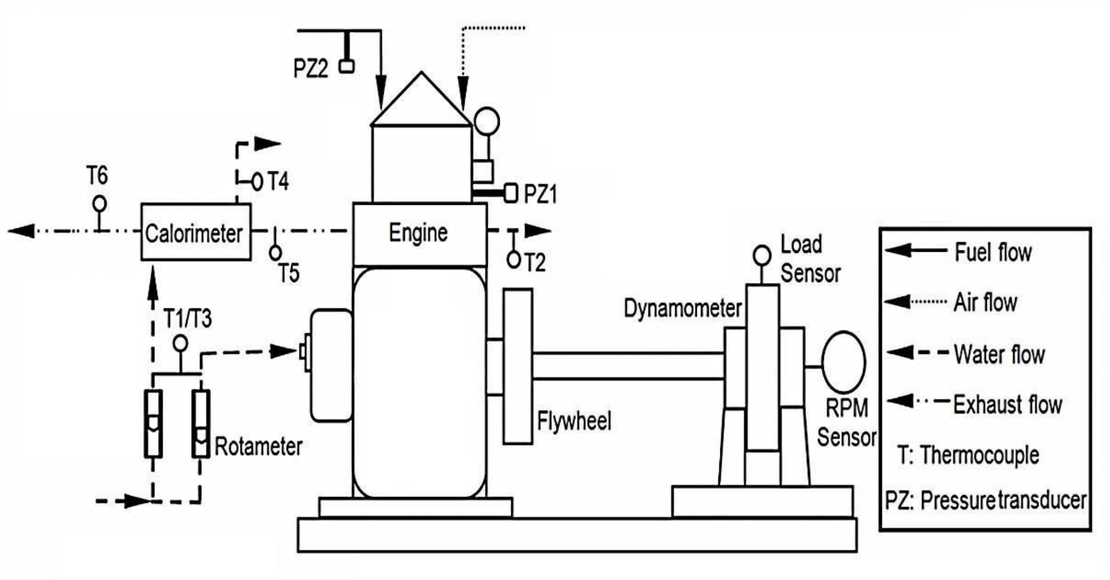

Fig. 3.1 The schematic diagram of the VCR diesel engine setup

Table 3.1 Specification of VCR Diesel Engine

System specifications	
Parameter	Specification
Make and model	Kirloskar, Model TV1
Product	VCR Engine test setup , Code 234
Type	Single cylinder, four stroke,VCR diesel engine
Power	3.5 kW (@ 1500 ± 50 rpm)
Type of cooling	Water cooled
CR range	12 – 18
Injection	23º BTDC
Combustion chamber	Hemispherical bowl in piston type
Dynamometer	Eddy-current (Make: Saj, Model: AG10)
Air box	MS fabricated with orifice meter and manometer (100 - 0 - 100)
Type of air induction	Naturally aspirated
Fuel tank Capacity	15 lit with measuring tube
Calorimeter	Pipe in pipe type
Rotameters	Engine cooling 40-400 lph, calorimeter 25-250 lph
Data acquisition Software	‘Enginesoft’ engine performance analysis software
Transmitters, sensors and indicators	
Fuel flow transmitter	Edu-tek, range 0-500 mm WC
Air flow transmitter	Pressure transmitter (-) 250 mm WC
Pressure sensors	PCB Piezotronics, range 5000 psi
Temperature sensors and Transmitters	PT100 (RTD) type, range 0-100º C, output 4-20 mA (4 nos), K (ungrounded) type, range 0-1200º C, output 4-20 mA (2 nos)
Load sensor and indicator	‘S’ Beam, Universal type load cell with digital indicator, range 0-50 kg
Data acquisition device	NI USB-6210, 16-bit, 250 kS/s
Setup constants	
Pulse per revolution	360º
No. of cycles	10
Fuel measuring interval	60 s
Speed scanning intervals	2000 ms
Bore × Stroke	87.5 mm × 110 mm
Orifice diameter	20 mm
Dynamometer arm length	185 mm
Connecting rod length	234 mm
Theoretical constants	
Orifice coefficient of discharge	0.6
Specific heat of exhaust gas	1.00 – 1.25 kJ/kg-K
Specific heat of water	4.186 kJ/kg-K

3.2 Instrumentations for Measurements

A variety of sensors, transmitters, and indicators make up the VCR research diesel engine configuration. These are attached to the data acquisition device (DAD) and used to measure practically all of the direct and indirect performance metrics automatically.

Air and Fuel Flow Measurement

Measurement of air and fuel flow can be done manually or mechanically. The variation in height of the water column in the manometer is used to manually measure airflow. It is linked across the orifice meter, which allows air to enter the engine panel box before exiting to the engine manifold. Manual fuel measurement is done by transferring fuel from the tank to the measuring tube for a set amount of time.

P-θ Measurement

Two dynamic pressure sensors developed by PCB Piezotronics are installed on the cylinder head and fuel injector. Both have the same specifications and can identify compression, combustion, explosion, hydraulic, fluidic, and other pressures. With a TDC pulse, an optical crank angle sensor (made by Kubler) is utilized to measure each degree rotation of the crank.

Temperature Measurement

The inlet and exit temperatures of engine cooling water flow and calorimeter water flow are measured using four PT100 temperature sensors. Two K type thermocouples measure the inlet and outlet temperatures of the exhaust gas to the calorimeter. The response time of the thermocouples employed in this study is greater than o.o8 seconds (for the constant speed engine around 1500 rpm). As a result, they are unable to demonstrate the pulsating behavior of the wear out gas in the form of temperature data and are achieved to be practically constant after a specific amount of time (about 5 minutes) at a given load.

Compression Ratio Variation Control

The VCR diesel engine has eight CR step variations ranging from 12 to 18. This is accomplished by using a locknut and adjustment mechanism to tilt the cylinder head. Six socket-headed vertical Allen bolts were installed on two supporting blocks on either side of the cylinder and needed to be loosen for CR modification.

Performance Measurement

The measurement of the performance parameters is done manually by collecting the data from the various indicators after setting the running engine at particular load level. The basic correlations used for estimating the performance parameters are included in the following chapters.

Emission Measurement

Emission analysis was performed using an AVL DiGas 444 analyzer. The AVL DiGas 444 analyzer determined the relative quantities of specific gaseous elements in the exhaust gases of motor vehicles. Carbon monoxide (CO), carbon dioxide (CO_2), hydrocarbons (HC), oxygen (O_2), and nitric oxide (NO) are the gases in question (Optional) Table 3.2 shows the resolution, accuracy, and range of various emission characteristics. Electrochemical measurement cells are used to detect carbon monoxide (CO) and nitrogen oxides (NOX), whereas infrared and Pellistor Heat Affect Detectors are used to monitor carbon dioxide (CO_2) and hydrocarbon (HC). The analyzer uses the ASTM-D6522 standard to monitor emissions. The gas analyzer's working concept is as follows: During steady engine running, flue gas is allowed to surge past a probe and is dried off by a condensation trap. After assessing each of the CO_2, CO, HC, and NO_X emission attentiveness present in the flue gas, separate sensors and measurements are then shown on the control unit's screen.

Table 3.2 The specifications of AVL DiGas 444 Analyzer

Sl. No.	Measured gas	Resolution	Accuracy	Range
1	Oxygen	0.1%vol	< 2% vol: ±0.01% vol ≥ 2% vol: ± 5% of vol	0 – 22%vol
2	Carbon monoxide	0.01%vol	< 0.6% vol: ±0.03% vol ≥ 0.6% vol: ± 5% of initial value	0-10% vol
3	Carbon dioxide	0.01% vol. < 25% 0.1% vol. > 25%	< 10% vol: ±0.05% vol ≥ 10% vol: ± 5% of vol	0 – 20%vol
4	Nitric oxide	1 ppm vol	< 500 ppm vol: ± 50 ppm vol ≥ 500 ppm vol: ± 10% of initial value	0 – 5000 ppm
6	Hydrocarbon	≤ 2000: 1 ppm vol, > 2000: 10 ppm vol	< 200 ppm vol: ± 10 ppm vol ≥ 200 ppm vol: ± 5% of initial value	0–20000ppm

3.3 Dual Fuel Modifications

By adding a venturi gas mixer to the inlet manifold of the VCR engine test setup, it was converted to a biogas run dual fuel system as shown in Fig. 3.2. The existing setup and the components of the modified setup are shown in Fig. 3.3.

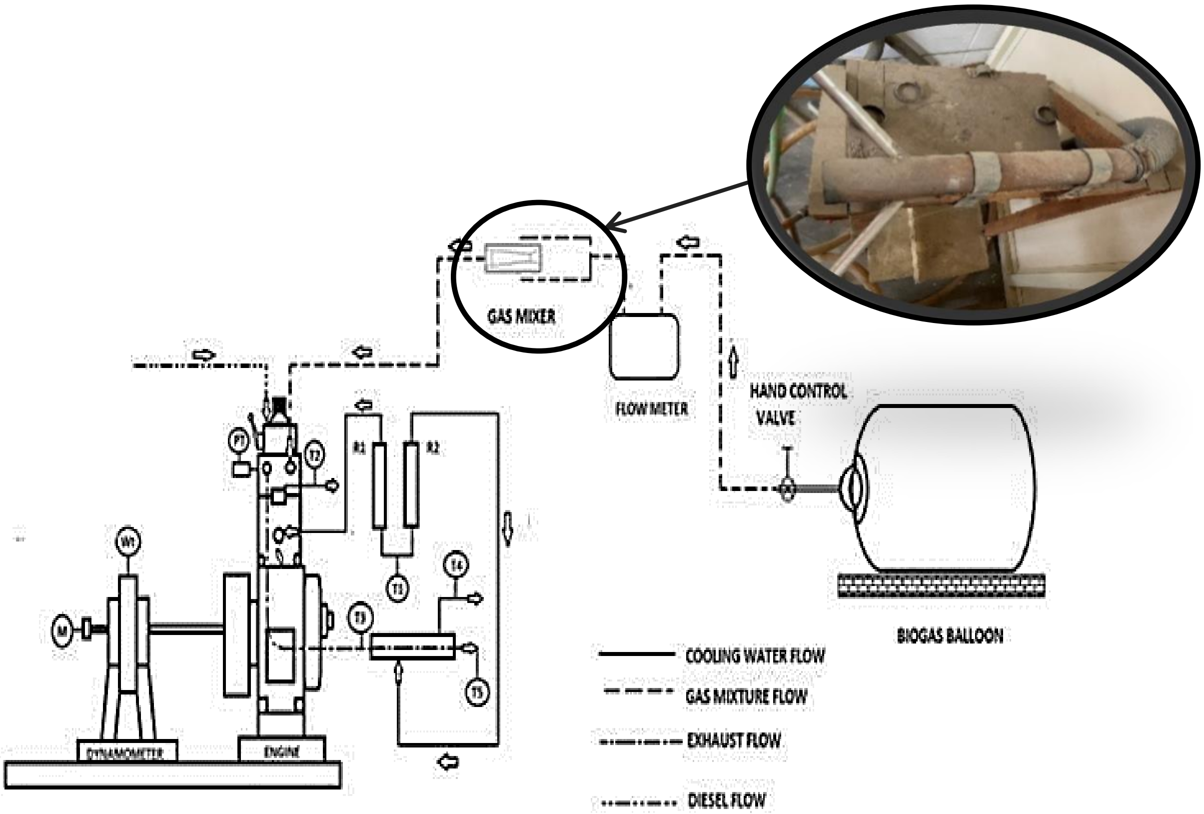

Fig. 3.2 The schematic diagram of the modified dual fuel VCR diesel engine setup

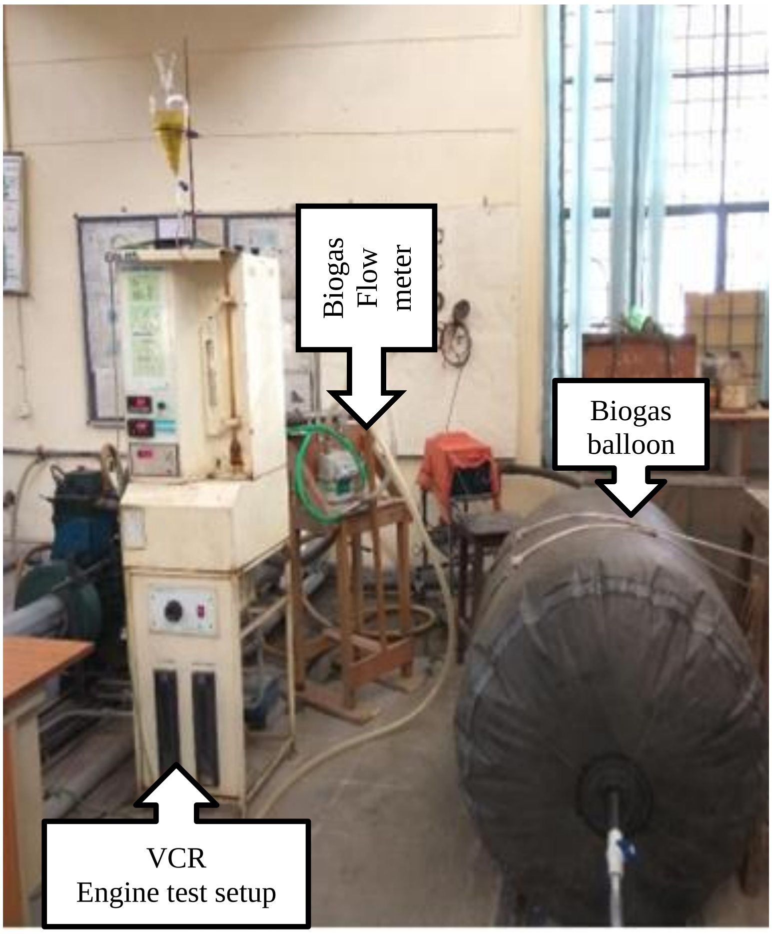

Fig.3.4 Modified test setup

3.3.1 The Fuel air mixing chamber

The only actual change needed to convert a diesel engine to dual fuel mode is to connect a fuel air mixing chamber to the input manifold [19]. In dual fuel mode, the fuel air mixing chamber is critical because it provides a burnable combination of fuel gas and air in the correct amount and quality for efficient engine operation under all situations [51]. The flow of fuel gas can be varied depending on the desired performance. The maximum air to fuel proportion ought not be under 1.5 keeping in mind the end goal to guarantee combustion even for the pilot fuel. For the construction of a fuel air mixing chamber for a certain engine, the volumetric efficiency, rated power, speed, specific fuel consumption, swept volume, and manifold connection diameter were all taken into consideration [18].

Table 3.3 Engine Specifications

Bore Diameter	D_B	87.4 mm
Stroke Length	L	111 mm
Speed	N	1500 rpm
Power	P	5.3 KW
Specific Fuel Consumption	Sfc	0.678m^3/KW-Hr
Volumetric Efficiency	H_{vol}	90%
Substitution Percent of diesel by biogas	S	80%
Diameter of Inlet Manifold	D_i	40 mm (measured)
Type of Engine	4-Stroke	
No. of cylinders	K	1

The design of the fuel air mixing chamber used the concept of venturi meter and is known as venturi-gas-mixer. The gas mixer has two gas inlets, one air inlet, and one air-gas exit. It has two sections: a smooth contraction and an expansion. Because of the smoothness of the contraction and expansion, the irreversible pressure loss is low. The goal of the converging portion is to increase the fluid's velocity while temporarily lowering its static pressure. Because of the maximum velocity, the neck region will have the lowest pressure. There is a pressure difference between the inlet and the throat. This flow rate is proportional to the pressure differential. Converging angle (n), diverging angle (2), nozzle angle (n), and nozzle angle (n) are the four primary design parameters (ratio between the diameters of throat and inlet manifold). As illustrated in the diagram, the values of 1, 2, n, and 0.46 are 20°, 5°, 35°, and 0.46, respectively (Fig.3.4). Past simulation experiments suggested these geometric values. The throat diameter, inlet manifold diameter, and biogas inlet diameter, respectively, are 20 mm, 40 mm, and 9 mm. The diameter of biogas inlet nozzle is found to 8 mm be based on the methodology as suggested by von Mitzlaff . Confer to Stewart *et al.*, The diverging section should be 10 times the diameter of the inlet manifold [21]. In comparison to the manifold diameter of 40 mm, the length of the same is discovered to be 250 mm while considering the diverging angle of 5°. All of the gas mixer's design specifications are based on a diesel engine with a rated power of 5.3 kW. The engine's volumetric efficiency and speed are set to 90% and 1500 rpm, respectively. The maximum diesel substitution by biogas in this engine is taken as 80% .

Calculations:

<u>Step- 1</u>

Swept Volume,

$$V_s = (\frac{\pi}{4} \times D_B{}^2 \times L) = 0.661 \times 10^{-3}\, m^3$$

Volumetric Air Intake,

$$V_{ai} = [{}^{(\eta_{vol} \times V_s \times N)} \big/ {}_{(2 \times 60)}] = 0.00744\ m^3/s$$

<u>Step- 2</u>

Cross Sectional Area of Intake,

$$A_i = (\frac{\pi}{4} \times d\ {}^2) = 1.256 \times 10^{-3}\, m^2$$

Step- 3

Intake Velocity,

$C_i = (V_{air}/A_i) = 5.92\, m/s$

Step- 4

Volume flow of fuel at rated power,

$f_c = sfc \times P = 1.15 \times 10^{-3}\, m^3/s$

Step- 5

Volume flow of biogas,

$f_{c1} = s \times f_c = 0.924 \times 10^{-3}\, m^3/s$

Step- 6

Area of nozzle,

$A_2 = (f_{c1}/C_2) = 0.046 \times 10^{-3} m^2$

Diameter of nozzle,

$d_2 = (4+A_{2/}\pi) \quad = 7.65mm$

$(d_2)\ safe = 110\%\ of\ d_2 = 8.41mm \approx 9$

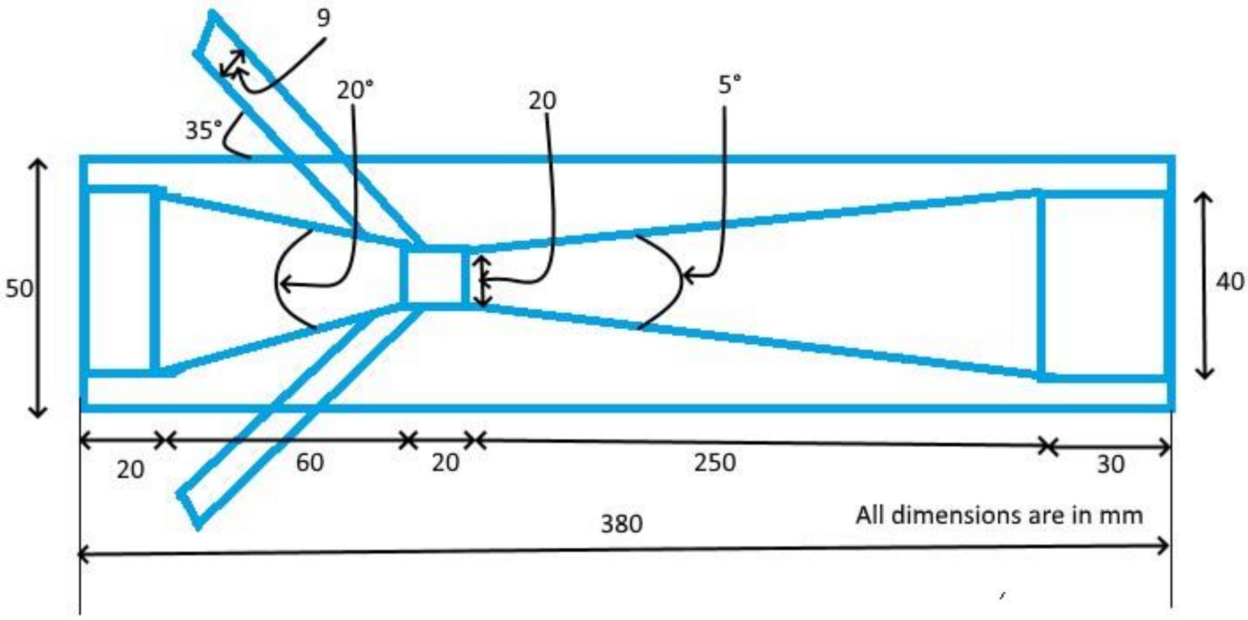

Fig.3.5 Schematic diagram of Fuel air mixing chamber

Fig.3.6 Pictorial view of Fuel air mixing chamber

The fuel air mixing chamber will be installed in between the inlet manifold line. Then after these two biogas nozzles will be connected by one “Y-Divider”. This divider will divide the biogas in equal amount.

3.3.2 Manual Regulation of Pilot Fuel Quantity

The engine stop lever was modified to allow the amount of diesel fuel delivered into the combustion chamber every cycle to be controlled. This was done to allow for a reduction in diesel fuel injection during dual fuel operation, as diesel was to be used as a main fuel to ignite biogas in modest quantities. It also helps to determine the amount of liquid fuel replacement at any given load correlate to any given speed. V-section mild steel angle line, mild steel, and M16 bolt and nut were used to make the pilot fuel regulating device. The plate was drilled with a hole and then welded to a part of the angle line. The nut was then welded into the hole to allow the bolt to be tightened or loosened to control the amount of driver fuel by modifying the position of the stop lever. With the help of nut, the angle line was welded to engine support frame located near and in line with the engine stop lever. The engine stop lever would be pushed by tightening the bolt in the nut, restricting the fuel pump output and so lowering the amount of fuel supplied to the injector. The procedure for finding out liquid fuel replacement is as follows: Initially, the rpm and fuel consumption corresponding to a particular load in diesel mode is noted down. During dual fuel operation, biogas supply is slowly open. As both diesel and biogas undergo combustion, therefore, more energy is released. This increases the rpm of the engine. After certain duration, the governor slowly reduces the rpm by reducing the liquid fuel supply. The control lever connected to the fuel shut off valve, however, limits the fuel further to perfectly match the rpm of the diesel mode. The fuel control shut valve connected to the fuel pump is slightly pushed by this control lever.

Fig.3.8 Engine stop lever

Fig.3.9 Adjustable lever arm arrangement

3.3.3 Biogas Supply and Metering System

Figure 3.12 depicts a diagram of the biogas supply and metering system. Biogas was utilized to power the modified diesel engine in dual fuel mode, and it was fed into the engine via a flexible gas bag and a flow meter to track how much gas was consumed. A flexible (collapsible) gas bag with a 1m3 capacity, flexible clear tubes, a flow meter, a Y-divider, and a flow switch comprised the system. The gas was created in a digester at the Institute of Delhi Technological University in Delhi using vegetable waste. The flow switch (shut-off valve) was used to control gas flow by turning it to 1/4, 1/2, 3/4, or fully open positions, as well as to shut off the gas supply when it wasn't in use.

Fig.3.11 Biogas Flowmeter

3.4 Experimental Procedure

The engine's performance was tested using diesel in single-fuel mode and biogas as the primary fuel and diesel as the pilot fuel in dual-fuel mode. The VCR engine is first operated with diesel, with a compression ratio of 18 and an ignition temperature of 23^{o}BTDC. To ensure effective fuel combustion, the engine is first driven at no load for a period of time to allow for optimal warm-up. During the experiments, the engine is tested at 20 %, 40 %, 60 %, 80 %, and 100 % load. The engine speed reduces with increase in load. To keep a constant BP, the engine consumes more gasoline, resulting in a higher heat release, resulting in higher temperatures inside the exhaust gas, cylinder, and cooling water outlet. The engine is permitted to run at any set load situation for a few minutes until it reaches a steady state condition. The temperature, rpm, and load values from each indicator were then recorded.

The amount of diesel fuel consumed in a minute was used to calculate the rate of consumption. Before each experiment, the fuel tank was filled while the tank outflow valve was kept closed. The fuel tank exit valve was opened at the start of the experiment to enable fuel into the burette and fill the fuel line. The gasoline tank exit valve was off, and the engine was permitted to operate on fuel from the metering burette for one minute, while recording the amount of fuel consumed. Basically, the manometer was used to found the air flow rate through the difference of height in their water column. Initially, the engine's rpm in diesel mode is recorded for a specific load in a dual fuel operation. The Biogas supply valve is gradually opened. The engine's speed improves as more biogas is fed into it. The increased energy obtained by the engine as a result of biogas burning enhances the engine's speed. The flow of biogas is gradually raised until the rpm cannot rise any higher. Meanwhile, the governor tries to reduce the engine's speed.

However, in diesel mode, the main fuel, which is diesel in this case, is gradually by the decreased by pilot fuel regulating device to perfectly match the rpm at any given load. After that, the main fuel supply is lowered until the original rpm run in diesel mode is attained. After allowing the engine to run for a few minutes, readings from the biogas flow meter are taken. The initial and end meter readings were recorded for 30 seconds for each test. The volume of gas utilized by the engine was computed by subtracting the initial from the final gas reading. The mass flow rate was determined using the volume of gas consumed, the density of the gas, and the time elapsed. For emission analysis, an AVL DiGas 444 analyzer was used.

Chapter 4
RESULTS AND DISCUSSIONS

4.1 Results of Diesel - Biogas Run Dual Fuel Engine

Diesel is the pilot fuel for this chapter, with biogas serving as the principal gaseous fuel for dual fueling. Tables 4.1 and 4.2 provide the experimental matrix and test fuel properties. Load variations are performed in increments of 2.4 kg from no-load to full-load (12 kg) (20 percent-100 percent). The diesel and dual fuel modes compared at standard diesel setting (i.e. Compression ratio =18 and Injection timing = 23º BTDC). The performance analysis is performed based on theoretical equations given in Appendix.

Table 4.1 Experimental Matrix of the Diesel-Biogas Run Dual Fuel Diesel Engine

Mode	Fuel used		CR	Loading conditions (%)
Diesel	100% Diesel		18	20,40,60.80,100
Dual	DFM1	Pilot Fuel: Diesel Primary Fuel: Biogas		

Table 4.2 Fuel Properties

Properties	Diesel	Biogas
Density(kg/m^3)	840 (Sahoo *et al.*, 2011)	0.91
Lower calorific value (MJ/kg)	42 (Sahoo *et al.*, 2011)	26.67
Cetane number	45-55 (Sahoo *et al.*, 2011)	-
Auto-ignition Temperature (K)	553 (Sahoo *et al.*, 2011b)	1087 (Sahoo *et al.*, 2011)
Stoichiometric air fuel ratio	14.92 (Sahoo *et al.*, 2011)	10 (Sahoo *et al.*, 2011)

4.1.1 Performance Analysis

Brake thermal efficiency (BTE), brake specific fuel consumption (BSFC), volumetric efficiency (VE), exhaust gas temperature (EGT), and liquid fuel replacement are among the performance evaluations considered (LFR). As demonstrated in Fig. 4.1, the BTE increases as the load rise for both diesel and dual fuel modes. However, BTEs are observed to be higher for diesel contrast to dual fuel mode because of low calorific estimation of biogas. At 100% load, the BTEs are found to 20.01%for DFM1 contrasted with 33.23% for diesel mode. There is a decline in the BSEC when the load for diesel and DFM increases. Due to the limited conversion of gaseous fuel to work, the BSEC difference between dual fuel mode and diesel is relatively considerable at low loads.

However, due to the enormous load exerted on the engine at high load (above 80%), the BSEC of dual fuel mode changes dramatically. This promotes a high conversion of biogas fuel into labor. As seen in Fig. 4.1, the VE drop as the load rise for both diesel and dual fuel modes. As the load grows, the temperature of the exhaust gases rises, preheating the incoming air and lowering the VE. With increased load, the biogas substitution dislodges a larger portion of air in dual fuel mode. As a result, DFM has a lower VE than diesel mode. The VEs for DFM1 are 64.96 percent at 100 percent load, compared to 67.19 percent in diesel mode. Both diesel and DFM EGTs have a linear bond with load. The EGT for dual fuel mode is more than the EGT for diesel mode. This is because of biogas's late combustion, which reduces the time it takes for the engine to extract power from the fuel. As a result, the combustion products emerge as gases at a higher temperature. For DFM1, LFR increases as load increases. The maximal LFR was discovered to be 76%.

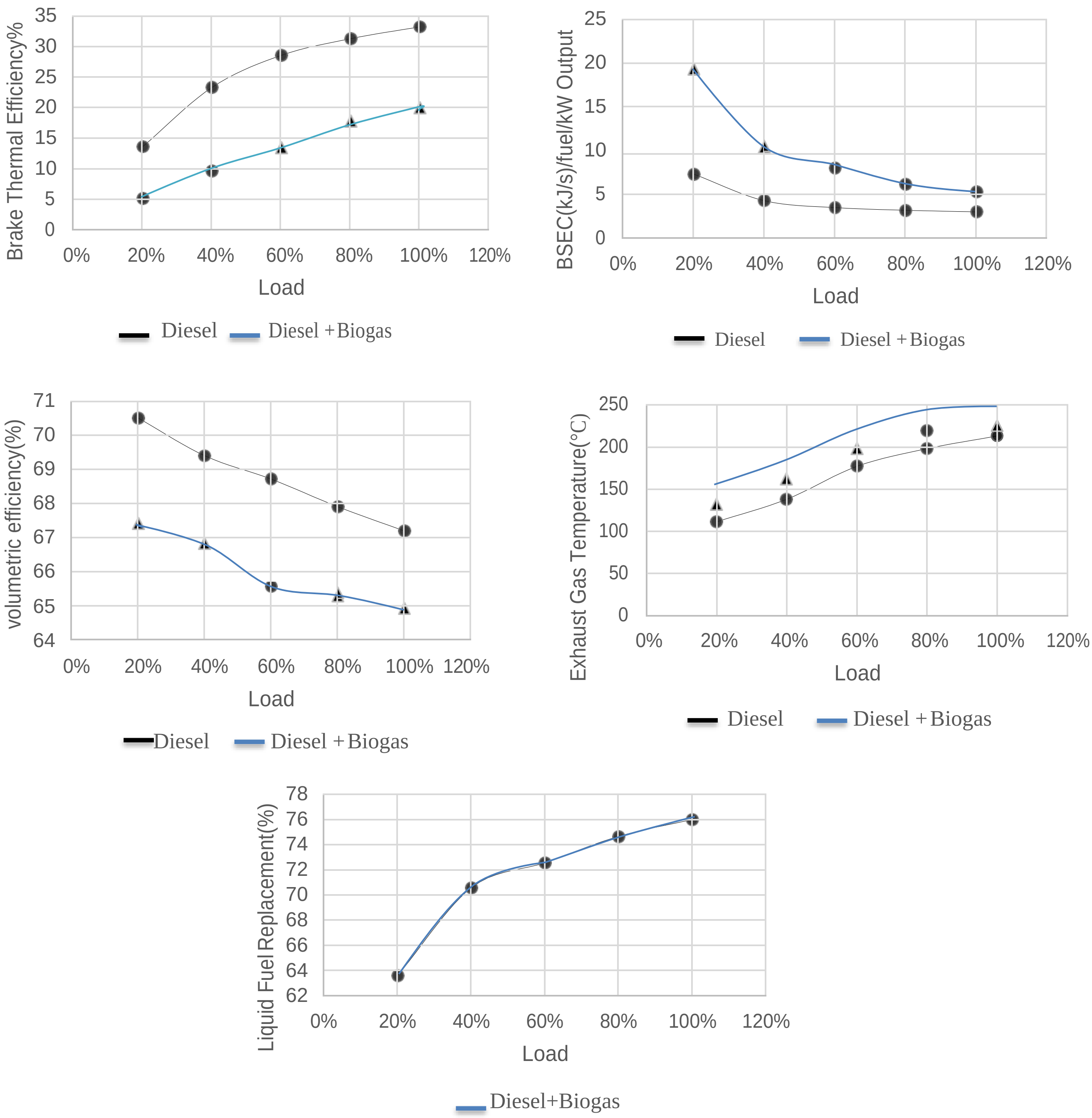

Fig. 4.1 Performance analysis for biogas run dual fuel diesel engine using diesel as pilot fuel

4.1.2 Emission Analysis

Carbon dioxide (CO_2), carbon monoxide (CO), hydrocarbon (HC), and nitrogen oxides (NOx) are all measured in emission analysis (NO_X). During engine operation, carbon dioxide is produced as a byproduct of fuel combustion. Carbon dioxide emissions for both diesel and DFM increased linearly with load, as seen in Fig. 4.2. As make the rise in engine load, the amount of fuel consumed by the engine increased in order to produce enough power to overcome the load. Because CO_2 is a product of fuel oxidation, this resulted in an rise in CO_2 emissions. However, dual fuel mode was observed to produce higher CO_2 emissions in contrast to diesel mode as Biogas constitute higher amount of CO_2. CO emissions are high at low loads, then drop at medium loads, and finally increase at high loads, as seen in Fig. 4.2. This is because the cylinder temperature is initially low at low load, resulting in improper fuel combustion. When the load is high, more fuel must be supplied. As a result, after a certain load, the fuel-air mixture becomes too rich to burn completely.

However, when compared to diesel mode, CO emissions from DFM are higher. That's because of the fact that biogas displaces air under DFM, and therefore, there is not enough oxygen is available for complete combustion. One of the regulated emissions is unburned hydrocarbon (HC), which consists of unburned fuel components and partially reacted components produced by the engine during operation. As shown in Fig. 4.2, the lower flame velocity of biogas contributes more to the creation of HC in dual fuel mode than in diesel mode. The HC emission is initially high at a low load of 20% due to inappropriate combustion due to the low combustion chamber temperature in DFM, but as the load grows to 80%, the HC emission gradually decreases. With reduction of oxygen availability at 100% load, emission of more hydrocarbons was witnessed as the fuel air mixture become rich. The formation of NO_X depends on temperature.

The oxidation of nitrogen present in the admitted air produces NO_X in the high-temperature combustion gases inside the cylinder. In this study, NOx emissions increase with load because more fuel is required to meet the increased load, resulting in an increase in combustion chamber temperature. However, as shown in Fig. 6.2, NO_X emissions in the diesel mode are significantly higher than in the DFM mode. This is because the temperature of the combustion chamber in the case of diesel is substantially higher than in the case of biogas because diesel has a higher calorific value. Furthermore, the presence of inert CO_2 in biogas lowers the combustion chamber temperature.

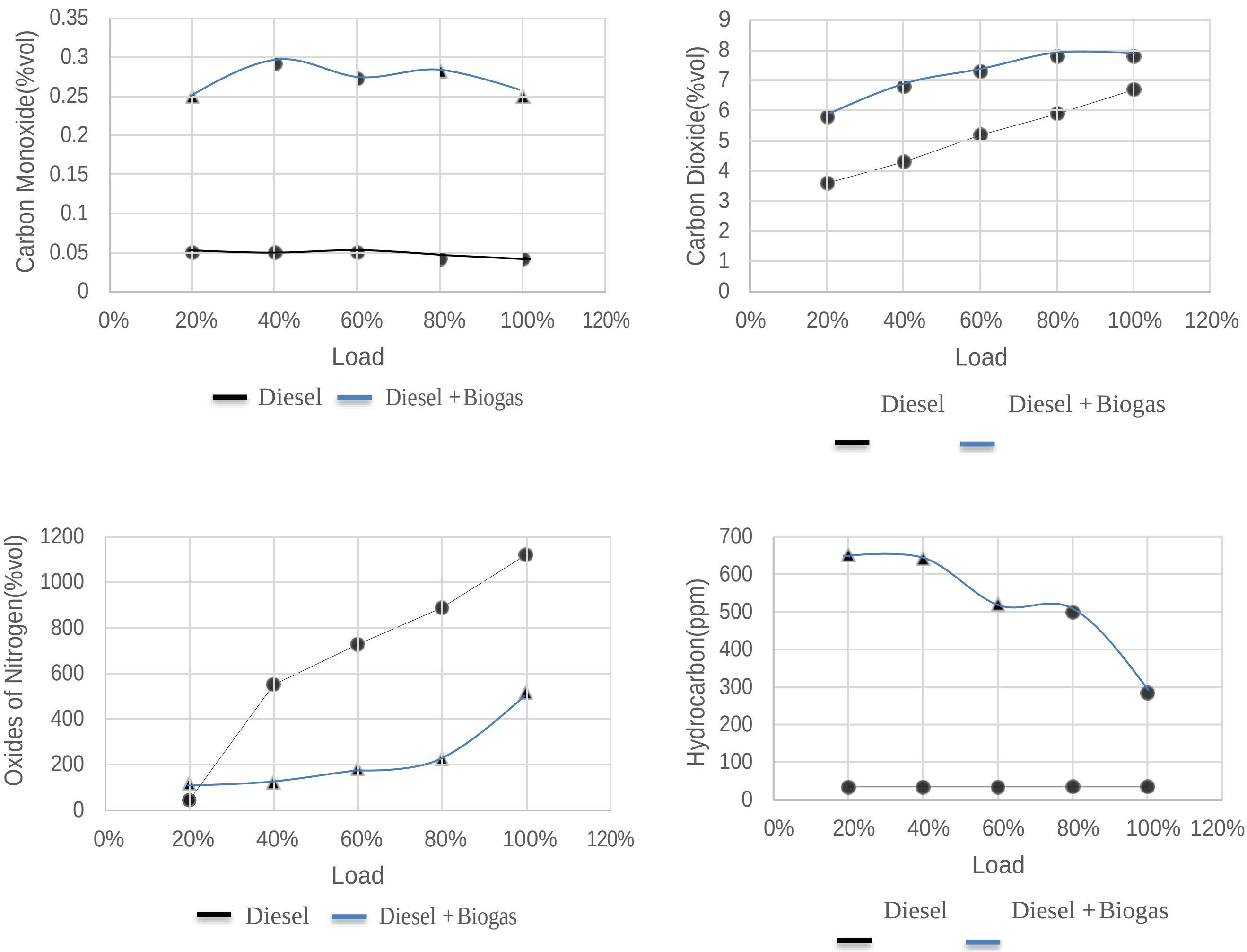

Fig. 4.2 Emission analysis for biogas run dual fuel diesel engine using diesel as pilot fuel

4.2 Results of Biodiesel - Biogas Run Dual Fuel Engine

The pilot fuels considered for this chapter is diesel and for dual fuelling biogas is considered as the primary gaseous fuel. The experimental matrix and properties of the test fuel are given in Tables 4.3 and 4.4. Load variations are performed in increments of 2.4 kg from no-load to full-load (12 kg) (20percent). The diesel and dual fuel modes compared at standard diesel setting (i.e., Compression ratio =18 and Injection timing = 23º BTDC). The performance analysis is performed based on theoretical equations given in Appendix.

Table 4.3 Experimental Matrix of the Biodiesel-Biogas Run DualzFuel Diesel Engine

<table>
<tr><th>Mode</th><th colspan="2">Fuel used</th><th>CR</th><th>Loading conditions (%)</th></tr>
<tr><td>Diesel</td><td colspan="2">SBD</td><td rowspan="4">18</td><td rowspan="4">20, 40, 60, 80, 100</td></tr>
<tr><td>Dual</td><td>DFM1</td><td>Pilot Fuel: SBD
Primary Fuel: Biogas</td></tr>
<tr><td>Diesel</td><td colspan="2">Pilot Fuel: RBB</td></tr>
<tr><td>Dual</td><td>DFM2</td><td>Pilot Fuel: RBB
Primary Fuel: Biogas</td></tr>
</table>

Table 4.4 Biodiesel Fuel Properties

Properties	SBD	RBB
Density(kg/m^3)	872.8	876.4
Lower calorific value (MJ/kg)	38.5912	39.1924
Cetane number	56.3 (Giacomo’s, 2013)	45-55 (Mittelbach,2004)

4.2.1 Performance Analysis

Brake thermal efficiency (BTE), brake specific fuel consumption (BSFC), volumetric efficiency (VE), exhaust gas temperature (EGT), and liquid fuel replacement are among the performance assessments that have been assessed (LFR). The general performance characteristics variation and their reasonings is similar to the above-mentioned diesel-biogas run dual fuel engine. At 100% load, the BTEs are found to higher for RBB and DFM2 as compared to SBD and DFM1. RBB has a little higher calorific value than SBD, resulting in a slightly higher BTE.

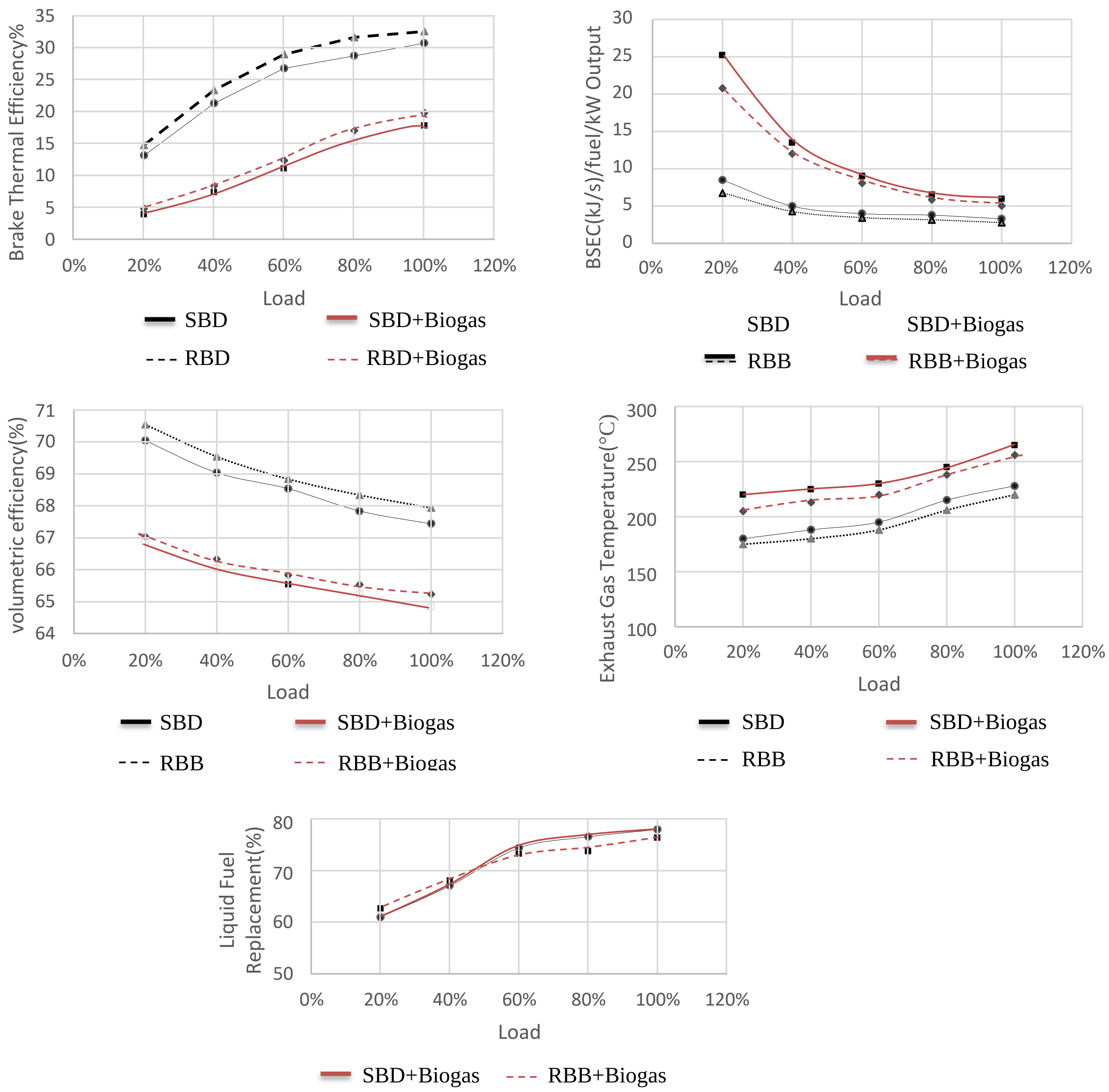

Fig. 4.3 Performance analysis for biogas dual fuel diesel engine using SBD, RBB as pilot fuel

4.2.2 Emission Analysis

Carbon dioxide (CO_2), carbon monoxide (CO), hydrocarbon (HC), and nitrogen oxides (NOx) are all measured as part of the emissions study (NO_X). The variations and their reasonings for considered parameters for emission analysis is similar to the above-mentioned diesel-biogas run dual fuel engine.

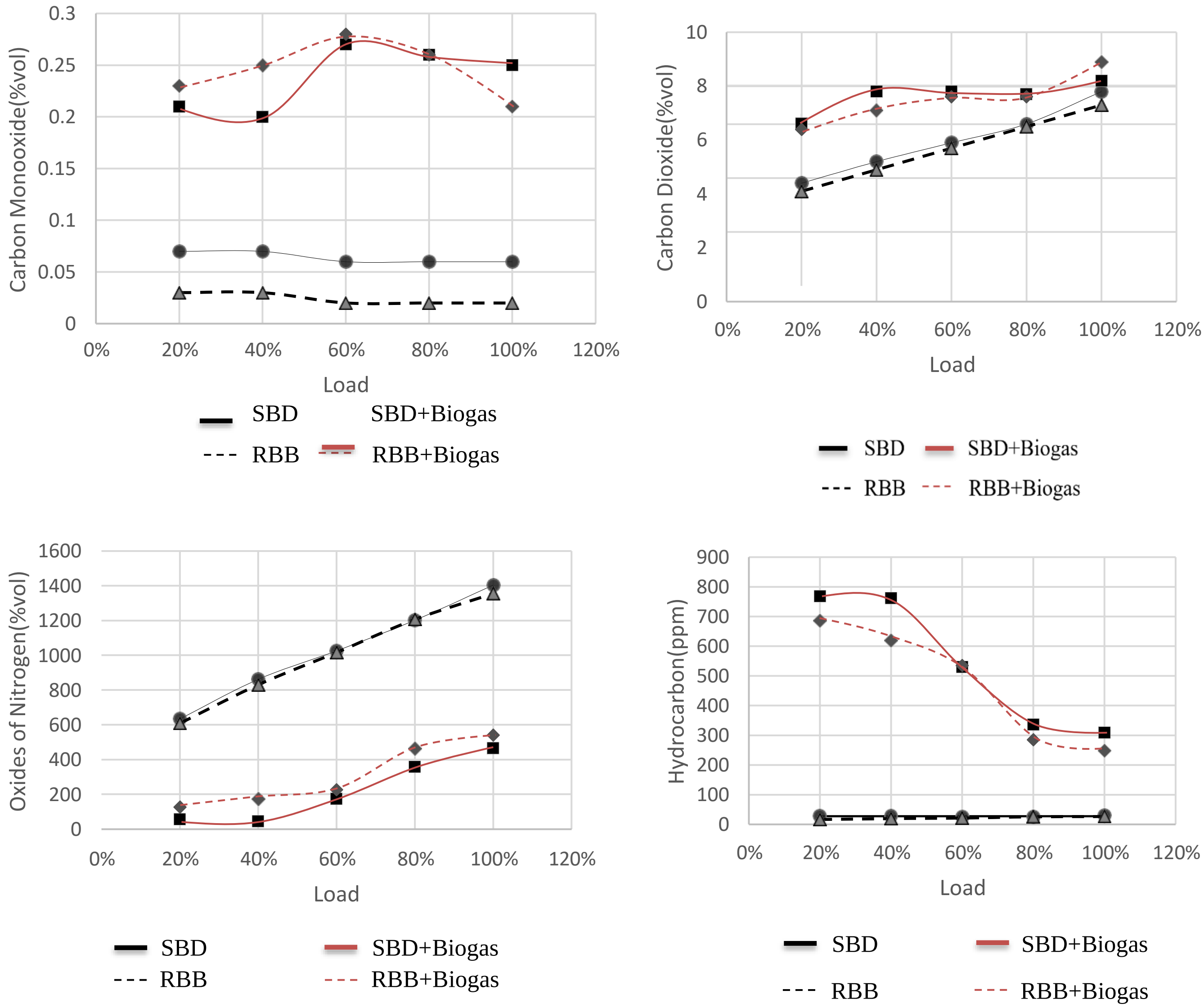

Fig. 4.4 Emission analysis for biogas run dual fuel diesel engine using SBD, RBB as pilot fuel

4.3 Results of Biodiesel and Ethanol blend Biogas Run Dual Fuel Engine

The pilot fuels considered for this chapter is SBDE and RBBE and for dual fuelling biogas is considered as the primary gaseous fuel. The experimental matrix and properties of the test fuel are given in Tables 4.5 and 4.6. Load variations are performed in increments of 2.4 kg from no-load to full-load (12 kg) (20percent). The diesel and dual fuel modes compared atstandard diesel setting (i.e., Compression ratio =18 and Injection timing = 23º BTDC). The performance analysis is performed based on theoretical equations given in Appendix.

Table 4.5 Experimental Matrix of the Biodiesel Ethanol Blend-Biogas Run DualzFuel Diesel Engine

Mode	Fuel used		CR	Loading conditions (%)
Diesel	SBDE		18	20,40,60.80,100
Dual	DFM1	Pilot Fuel: SBDE Primary Fuel: Biogas		
Diesel	Pilot Fuel: RBBE			
Dual	DFM2	Pilot Fuel: RBBE Primary Fuel: Biogas		

Table 4.6 Fuel Properties of Ethanol

Properties	Ethanol
Density(kg/m^3)	806
Lower calorific value (MJ/kg)	26.67
Boiling point	78.4^0C
Vapor pressure	5.95 kPa

4.3.1 Performance Analysis

Brake thermal efficiency (BTE), brake specific fuel consumption (BSFC), volumetric efficiency (VE), exhaust gas temperature (EGT), and liquid fuel replacement are among the performance evaluations considered (LFR). The general performance characteristics variation and their reasonings is similar to the above-mentioned diesel-biogas run dual fuel engine.

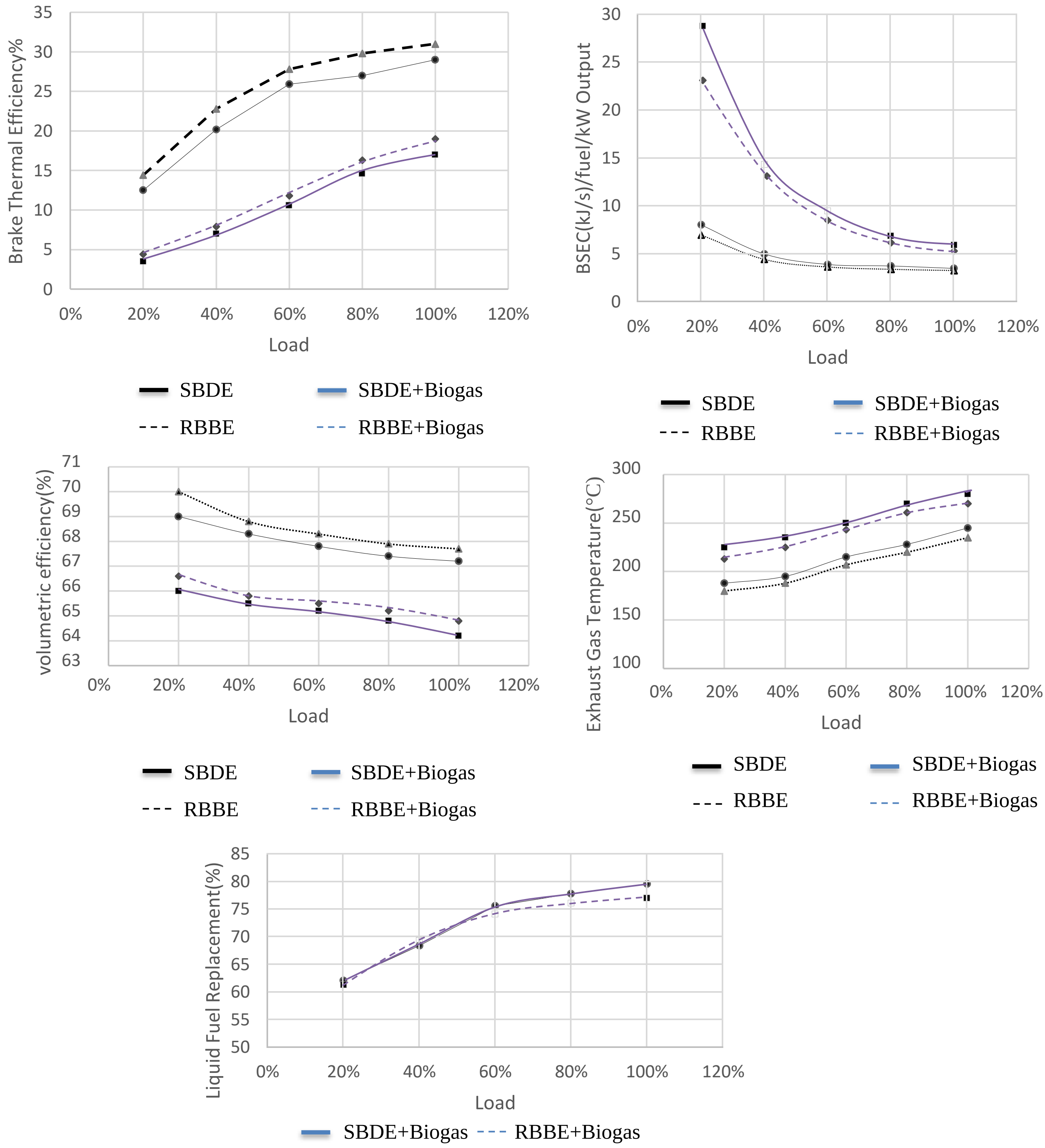

Fig. 4.5 Performance analysis for biogas run dual fuel DE using SBDE, RBBE aspilot fuel

4.3.2 Emission Analysis

Carbon dioxide (CO_2), carbon monoxide (CO), hydrocarbon (HC), and nitrogen oxides (NOx) are all measured as part of the emissions study (NO_X). The variations and their reasonings for considered parameters for emission analysis is similar to the above-mentioned diesel-biogas run dual fuel engine.

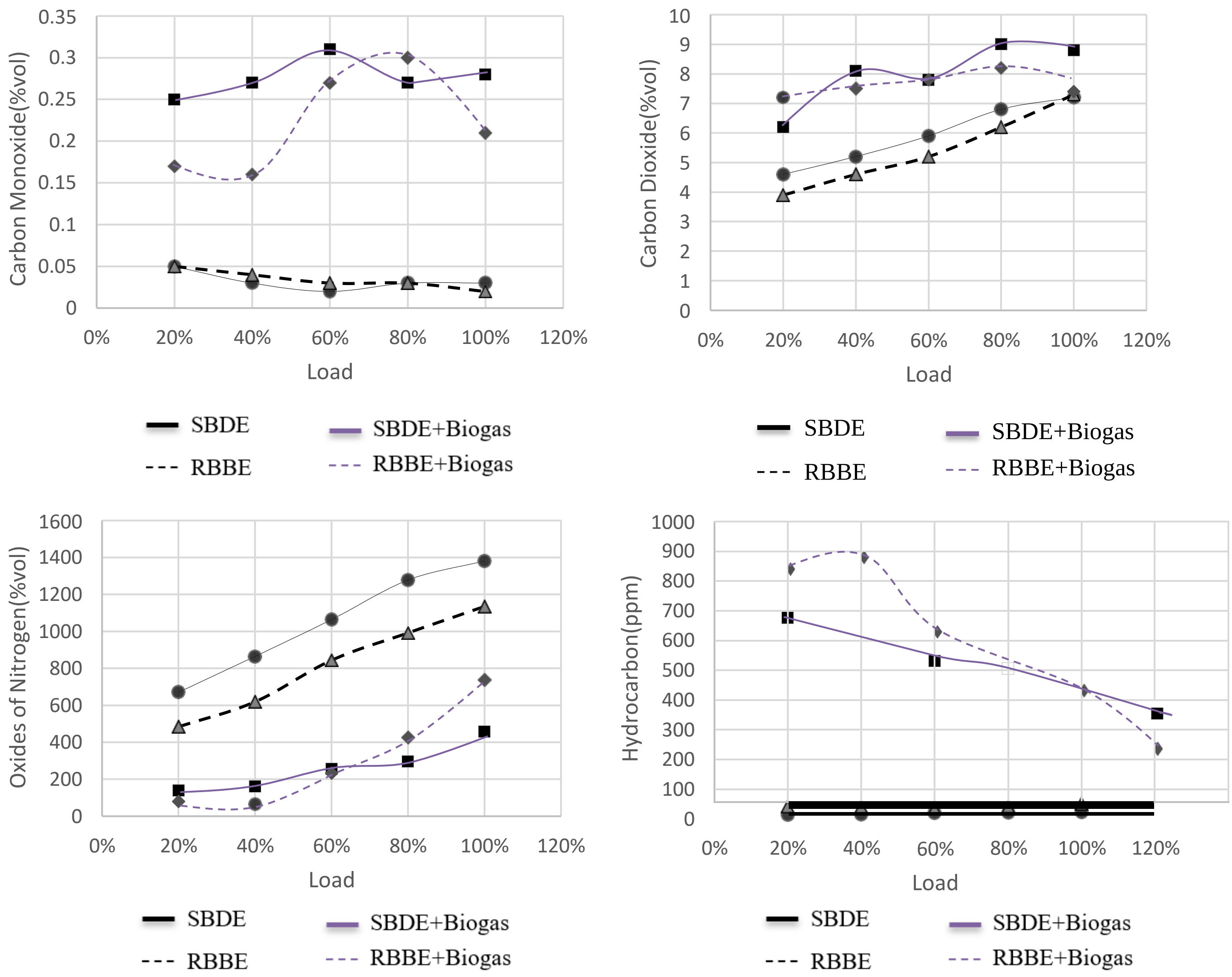

Fig. 4.6 Emission analysis for biogas run dual fuel diesel engine using SBDE, RBBE as pilotfuel

4.4 Results of Biodiesel and Octanol blend Biogas Run Dual Fuel Engine

The pilot fuels considered for this chapter is SBDO and RBBO and for dual fuelling biogas is considered as the primary gaseous fuel. The experimental matrix and properties of the test fuel are given in Tables 4.7 and 4.8. Load variations are performed in increments of 2.4 kg from no-load to full-load (12 kg) (20 percent). The diesel and dual fuel modes compared at standard diesel setting (i.e., Compression ratio =18 and Injection timing = 23° BTDC). The performance analysis is performed based on theoretical equations given in Appendix.

Table 4.7 Experimental Matrix of the Biodiesel Octanol Blend-Biogas Run Dual Fuel Diesel Engine

Mode	Fuel used		CR	Loading conditions (%)
Diesel	SBDO		18	20,40,60.80,100
Dual	DFM1	Pilot Fuel: SBDO Primary Fuel: Biogas		
Diesel	Pilot Fuel: RBBO			
Dual	DFM2	Pilot Fuel: RBBO Primary Fuel: Biogas		

Table 4.8 Fuel Properties of Octanol

Properties	Octanol
Density(kg/m^3)	824
Lower calorific value (MJ/kg)	40.6
Boiling point	195^{0}C
Vapor pressure	0.14 mm Hg

4.4.1 Performance Analysis

Brake thermal efficiency (BTE), brake specific fuel consumption (BSFC), volumetric efficiency (VE), exhaust gas temperature (EGT), and liquid fuel replacement are among the performance evaluations considered (LFR). The general performance characteristics variation and their reasonings is similar to the above-mentioned diesel-biogas run dual fuel engine.

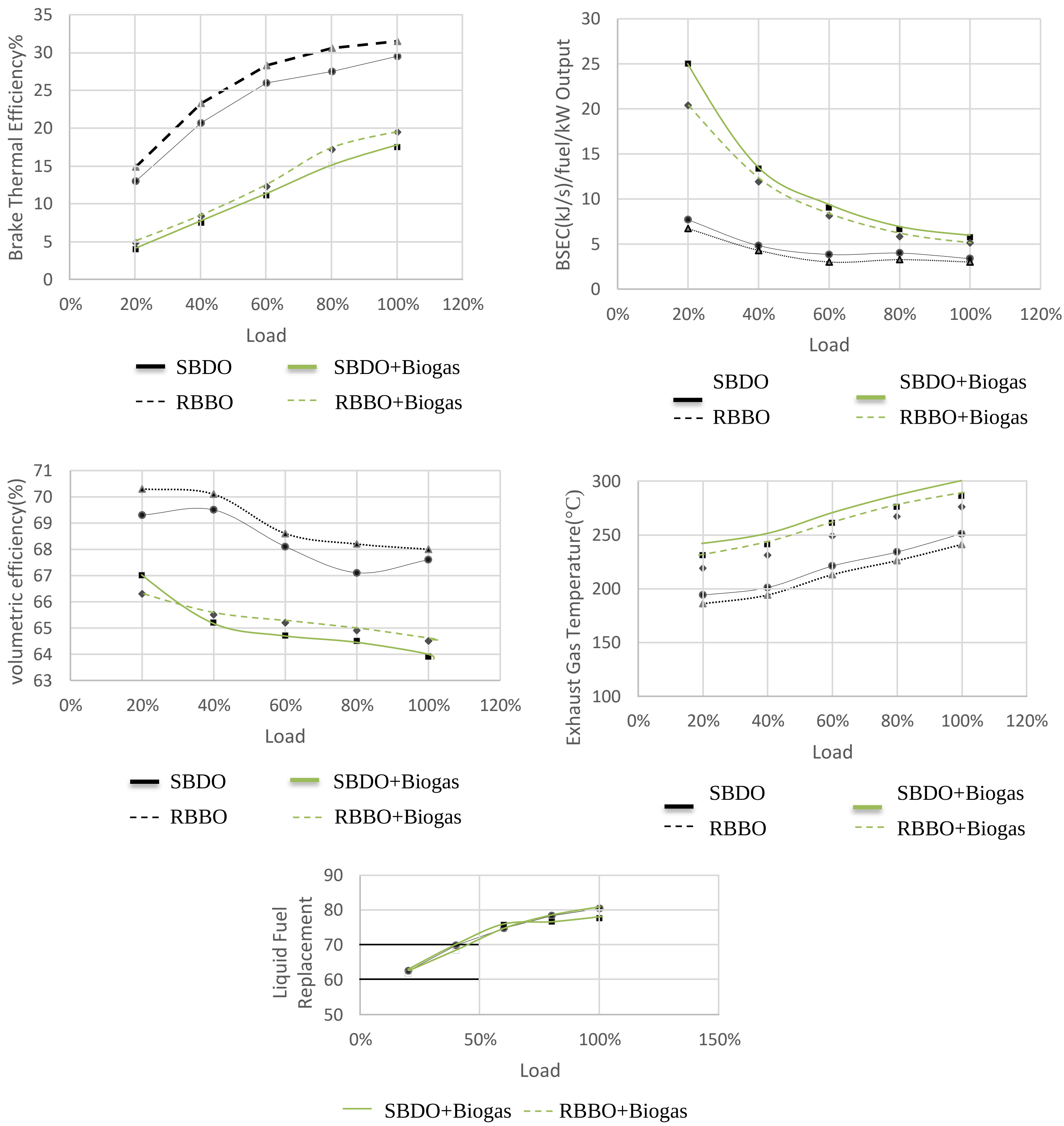

Fig. 4.7 Performance analysis for biogas run dual fuel diesel engine using SBDO, RBBO aspilot fuel

4.4.2 Emission Analysis

Carbon dioxide (CO_2), carbon monoxide (CO), hydrocarbon (HC), and nitrogen oxides (NOx) are all measured as part of the emissions study (NO_X). The variations and their reasonings for considered parameters for emission analysis is similar to the above-mentioned diesel-biogas run dual fuel engine.

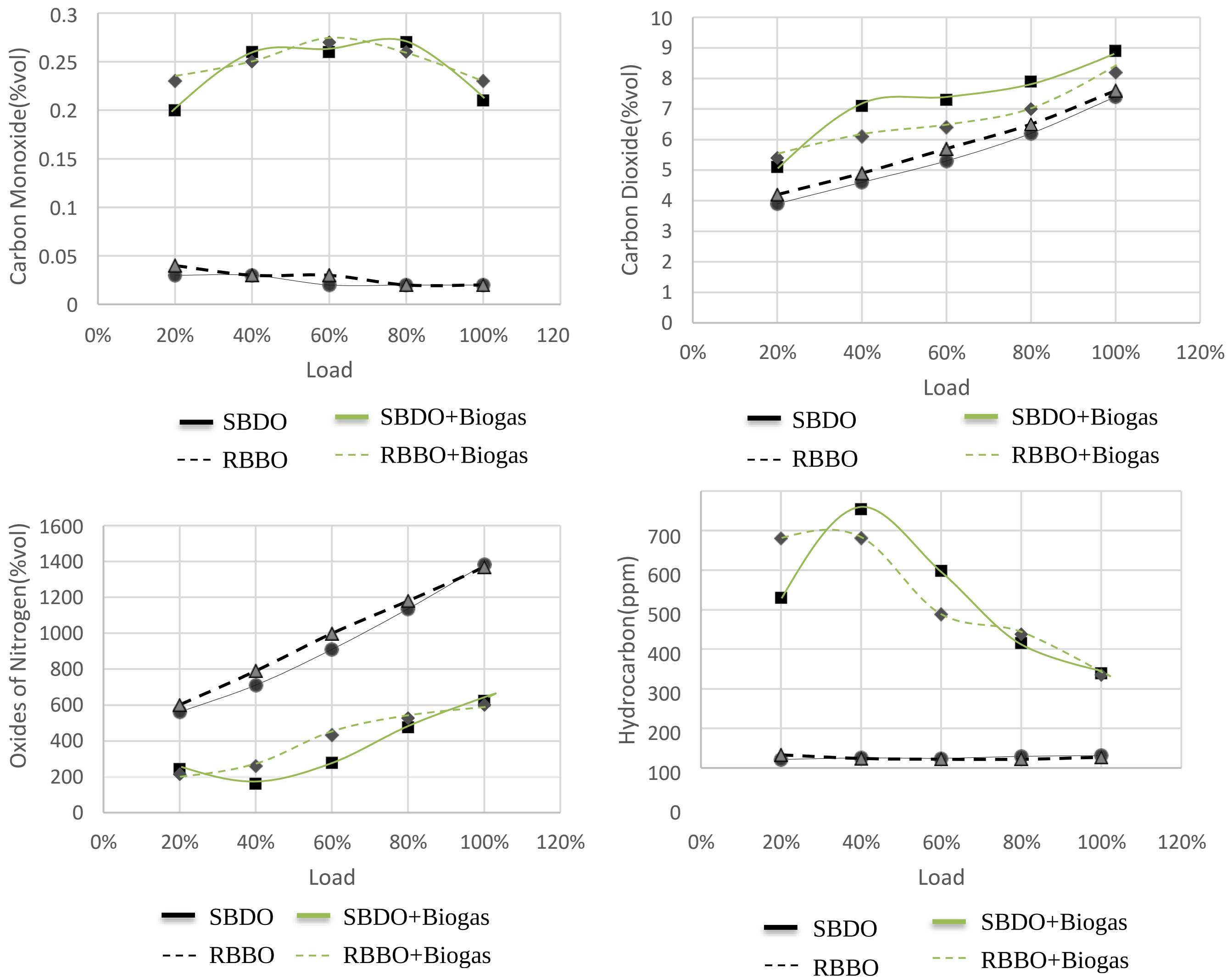

Fig. 4.8 Emission analysis for biogas run dual fuel diesel engine using SBDO, RBBO as pilotfuel

Chapter 5

CONCLUSION

The study resulted in the following conclusions: A compression ignition engine of DI type was converted to run on biogas, through the main fuels being diesel. In order to convert a diesel engine to a dual fuel engine, air-fuel mixing chamber would be installed through the air intake system to deliver an efficient technique of declaring gaseous fuel into the combustion chamber and mixing air and gaseous fuel homogeneously. A change to the engine lever arm is also required to allow for any necessary adjustments to the pilot fuel injection volumes.

It was possible to replace up to 80% of pilot gasoline with biogas, however engine operation on biogas alone was not practicable. This was owing to biogas's high autoignition temperature, which made ignition impossible without a significant volume of fuel. As a result, it was determined that using biogas in a DICI engine can assist replace diesel by up to 80%, but not fully. When compared to when diesel is used exclusively, this will reduce reliance on petroleum fuels and save money. As a result, it was determined that using biogas in a DICI engine can assist replace diesel by up to 80%, but not fully. When compared to when diesel is used exclusively, this will reduce reliance on petroleum fuels and save money.

The thermal efficiency of the brakes reduced when biogas was used instead of diesel because biogas has a lower calorific value. Due to low conversion of gaseous fuels into work in low load, the difference between the diesel mode and dual fuel mode. As higher thermal load is imposed BSEC of dual fuel mode tends to increase. For both diesel and dual fuel modes, the volumetric efficiency decreases as load increases. However, in comparison to diesel mode, dual fuel mode has a lower volumetric efficiency. In every scenario, the EGT of dual fuel mode is higher than diesel mode. The LFR is thought to grow as the load increases.

CO_2 emissions are observed to be higher in all circumstances of dual fuel mode collate to pilot fuel mode, although both modes' CO_2 emissions increase with increasing load. As for HC and CO emission it’s on the higher side at low load, decrease up to 80% load and then increase again. NOx emission was found to be increasing with increase in load, however it’s on the higher side for pilot fuel mode in contrast to dual fuel mode. Biogas is an excellent fuel for dual fuel engines and, because it is renewable and economical, it can be a good supplement to the country's fossil fuels.

Chapter 6

FUTURE SCOPE

In the dual fuel mode, biogas is the best alternate fuel for CI engines. The thermal efficiency, on the other hand, is low, and hydrocarbon emissions are significant. The presence of CO_2 in biogas is one of the reasons behind this. As a result, it is advised that the CO_2 in biogas be eliminated by promote the gas in order to improve flammability boundaries and flame speed, as well as better thermal efficiency and reduce emissions.

The displacement of air required for combustion with the gaseous fuel is another factor that contributes to reduced thermal efficiency and enlarged emissions in dual fuel operation with biogas. The engine should be changed to add a turbocharger, which would boost air intake and full fuel combustion, in order to aid improve efficiency and minimize emissions.

There is no open literature on the influence of driver fuel injection pressure on performance and emission structures. The biogas run dual fuel diesel engine's efficiency may be improved by adjusting the optimum injection pressure of the pilot fuel.

To perform Exergy Analysis so that one may know where exactly energy is lost and provide way to counter such loses on specific engine system.

APPENDIX

Equations used for Performance Analysis

1. Brake power(BP): $BP = \frac{2\times3.142\times N\times W\times R}{60\times1000}, kW$
2. Fuel flow (F1): $F1=(X\times60\times\rho_{Lf}/10^6)$, kg/hr

Where, X is the cc liquid fuel consumption of engine in 1mm(cc/min) and ρ_{Lf} is the liquid fuel density in kg/m^3

3. Brake Thermal efficiency (η_{bth}) :

For a case 100% diesel fuel operation, $\eta_{bth}= \frac{BP\times3600}{\dot{m}_d\times LHV_d}\times100$,%

Where, $\dot{m}_d$ for the diesel flow rate and LHV_d is the fuel lower calorific value.

For a case 100% diesel fuel operation, $\eta_{bth}= \frac{BP\times3600}{\dot{m}_{pf}\times LHV_{pf}+\dot{m}_g\times LHV_g}\times100$,%

Where, $\dot{m}_{pf}$ is the pilot fuel flow rate and LHV_{pf} is the fuel lower calorific value of pilot fuel and $\dot{m}_g$ is the biogas flow rate and LHV_g is the fuel lower calorific value of biogas.

4. Air flow, $F2=C_d\times(\pi/4)\times d^2\times\sqrt{\frac{2gh\times W_{den}}{A_{den}}}\times3600\times A_{den}$, kg/hr

5. Air-fuel ratio (A/F): $A/F=\frac{Air\ Flow}{Fuel\ Flow}=\frac{F2}{F1}$

6. Volumetric Efficiency, $\eta_{Vol}= \frac{F2}{(\pi/4)\times D^2\times L\times\frac{N}{n}\times60\times K\times A_{den}}\times100$,%

7. Brake specific energy consumption (BSEC):

It is defined as the fuel energy required (kJ/s) to produce unit kW shaft output.

$$BSEC = \frac{\sum(\dot{m}_{fuel}\times LHV_{fuel})}{BP},\ (kJ/s)_{fuel}/kW_{output}$$

8. Liquid fuel replacement rate (LFR):

It determined by using the liquid fuel mass flow rate in liquid fuel mode $\dot{m}_{di}$ and pilot fuel mass flow rate in dual fuel mode $\dot{m}_p$(kg/s).

$$LFR=[\{(\dot{m}_{di}-\dot{m}_p)\times100\}/\dot{m}_{di}],\%$$

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