Experimental Analysis of biogas run dual-fuelled diesel engine performance

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By

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DECLARATION OF ORIGINALITY

I hereby declare that this submission is my own work and that, to the best of my knowledge and belief, it contains no materials previously published or written by another person nor it contains no material previously published or written by another person, nor any materials presented for the award of any degree or diploma of Delhi Technological University or any other institution of higher learning. Any contribution made to this research by others is explicitly acknowledged in the dissertation.

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DEPARTMENT OF MECHANICAL ENGINEERING

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Assistant Professor
July, 2017

Supervisor Certificate

This is to certify that the work presented in this dissertation "***Experimental Analysis of biogas run dual-fuelled diesel engine performance****" by S.Lalhriatpuia* , Roll No. 2K15/THE/14, 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)*. Neither this dissertation nor any part of it has been submitted earlier for any degree or diploma to any institute or university in India or abroad.

Supervisor

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

S.LALHRIATPUIA

ABSTRACT

The price of fossil fuels is constantly increasing because of the limitations of deposit, supply and increase in demand for petroleum fuels resulting from industrialization. The regulations for emissions from diesel engines have also been strengthened, especially for particulate matter (PM) and oxides of nitrogen (NOx). There is therefore need to find alternative fuels which will reduce harmful exhaust emissions while maintaining high thermal efficiency. Biogas is one such fuel, which can be adapted for use in internal combustion (IC) engines. In this research, a Direct Injection Compression Ignition (DICI) engine was modified into a dual fuel engine which uses biogas and pilot fuel, with the focus on reduction of harmful exhaust emissions while maintaining high thermal efficiency. Diesel,biodiesel,alcohol were used as the pilot fuel wherease biogas was considered as the gaseous fuel. The performance and emission characteristics of the modified engine were compared with those of the conventional diesel engine. The air intake system of the engine was modified to allow mixing of air and biogas before supplying the mixture to the combustion chamber of the engine. The results obtained from this study showed that a DICI engine can be modified into a dual fuel engine that uses diesel and biogas. Engine performance tests were done for single fuel mode and dual fuel modes. Exhaust emissions of HC, CO, NOx and CO2 were measured for the various operating conditions. The results showed that at maximum substitution of pilot fuel with biogas (Biogas:SBDO in the ratio 80:20) occur in the case where SBDO is considered as pilot fuel. The brake thermal efficiency however, decreased from 33.23% in diesel mode to 18.86% in diesel dual fuel mode due to the lower calorific value of biogas compared to diesel. The performance characteristics of the modified engine showed that diesel can be substituted partly with biogas and the engine used for a range of applications. The most appropriate areas of use are stationary applications such as driving of electric power generators, hoisting in construction sites and driving of machines such as water pumps and concrete mixers. This is due to the challenge of compact storage of the gas.

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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
Ai	Cross Sectional Area of Intake
A2	Area of Nozzle

Ci	Intake Area
CFD	Computational Fluid Dynamics
D_B	Bore Diameter
di	Diameter of Inlet Manifold
d_2	Diameter of Nozzle
fc	Vol. Flow of Fuel
fc1	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
Vair	Volumetric Air Intake
ηvol (%)	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 fossil fuel shortage 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 peak oil production was presented. In spite of the fact that the pointers considered to anticipate the peak oil production have been enhanced altogether by considering numerous new elements, most researchers and associations concur that the pinnacle oil generation date will be before 2020[1].

After some time fuel costs are fundamentally unstable, as appeared in Figure 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].

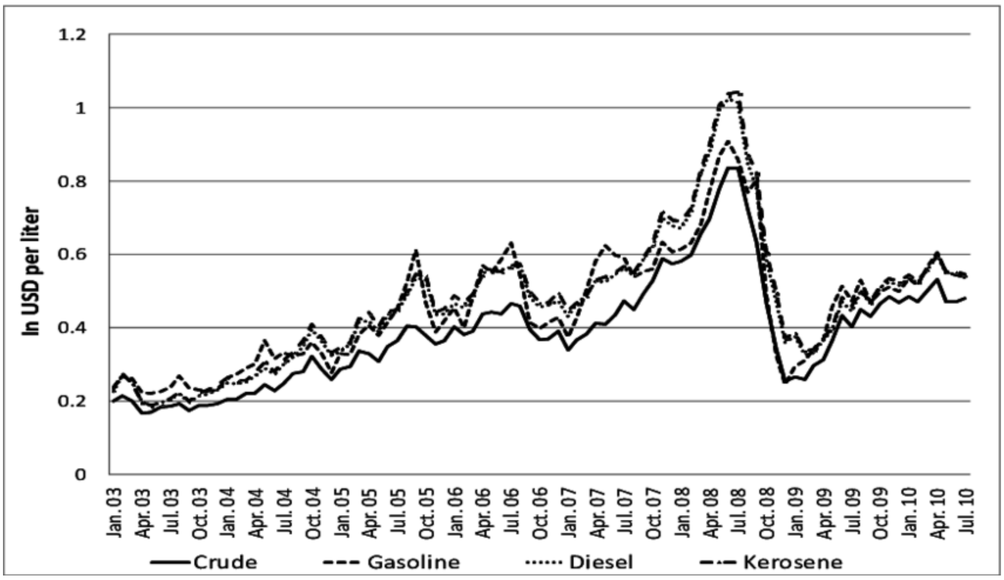

Figure 1.1 International fuel prices from 2003-2010[3]

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 CO2 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 with renewable and other alternative fuels, both gaseous and liquid fuels. Further fuel investment funds in transportation ought to be enhanced by improving motor innovation keeping in mind the end goal 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[4,5]. This is a consequence of the low energy content and low cetane number (CN) of most vaporous energizes.

To address the difficulties depicted above, researchzwork has been done which means to investigate some alternative fuels (renewable and non-reneawable), both gaseouszand liquid fuels, to supplant part or all of regular fuels to be utilized in dieselzengines.

1.2 Alternative Fuels for IC Engines

Petrol and diesel are common liquid fuels for internal combustion engines, while methanol, ethanol and biodiesel are alternative liquid fuels. Use of Liquified Petroleum Gas (LPG) in IC engines is also common, especially in food processing and manufacturing industries while alternative gaseous fuels for IC engines include hydrogen, syngas and biogas[6].

1.2.1 Biogas

Biogas is a gaseous fuel produced by fermenting organic material, that can be extracted from varied 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[7]. The process is known as anaerobic fermentation. The anaerobic process consists of four main steps: hydrolysis, acidogenesis, acetogenesis and methanogesis. It is regarded as an alternative clean energy resource for CI engines in view of its environmental friendly nature[8].

Sustainable power source and its transformation to warmth and power are key variables for maintainable advancement in nations with low or no oil fuel saves. A standout amongst the most critical inexhaustible energizes is biogas, which is made mostly out of methane (30-70%, by vol.) and carbon dioxide (20-40%, by vol.) and is a promising alternative fuel for internal combustion engines since it is renewable and environmental friendly[9].

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 it in a gas engine for power generation units 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].

1.2.2 Biodiesel

Biodiesel is generally 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 considered as biodiesel despite the fact that endeavors have been made to utilize these also in the CI engines. Biodiesel is normally created 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 obtain as a by-product, which is removed and can be used in personal care products or a variety of chemical applications[10]. Feed stocks are categorized for biodiesel into crops; including soybean and rapeseed among others. Oil trees such as palm oil were also included in their findings. They noted that these types of oils come from vegetables or animal fat, making them biodegradable and nontoxic. Given the fact that feedstock of biodiesel depends greatly on climate and local soil conditions, biodiesel may not be a very reliable fuel considering climatic change and soil degradation. Some of the plants which form the feedstock for biodiesel also take a long time to mature, leaving biogas as a more reliable bio-fuel compared to biodiesel[11]. The straight mineral oils have high viscosity that makes stream of fuel troublesome even at room temperatures and presence of glycerin in the vegetable oil causes development of heavy carbon stores on the injector nozzle holes. The oil and fats need to be converted to biodiesel them compatible for existing Compression Ignition (CI) engines. Biodiesel has insignificant sulfur and fragrant substance and higher lubricity, Cetane number,flash point, biodegradability and non toxicity [12]. It offers enhanced lubricity over certain low-sulfur petro-diesels and hence, can decrease wear of motor segments. Biodiesel is likewise simple to deal with because of its higher boiling point than that of diesel[13].

1.2.3 Bioalcohols

Bioalcohol fuel is an alternative fuel that can be utilized for driving apparatus and vehicles. Bioalcohol fuel is currently produced in four main types. These four are methanol, propanol, ethanol, and butanol. Ethanol or ethyl alcohol (CH_3) is a chemical compound derived from the fermentation of sugars, or even grapes, whose can be used as fuel, alone or mixed with gasoline in varying amounts. [14]. This fuel is also increasingly used as an additive for standard gasoline oxygenate, replacing methyl tert-butyl ether (MTBE). The last is responsible for considerable contamination of soil and groundwater. For ethanol production in the world is mainly used as a biomass source. This ethanol is called, by origin, bio-ethanol [14]. Bio-ethanol has the same characteristics and chemical composition than ethanol because it is the same compound. The difference only lies in the production process. The ethanol must be obtained from biomass, not from petroleum [14] .In the past little attention has been given to the utilization of alcohol fuels like ethanol or methanol in CI engines, this is because replacing dieselzfuel entirely by alcohols is very difficult or even almost impossible. Nevertheless, an increased interest has emerged for the use of it with different amounts and different techniques in light-duty dieselzengines as dual fuelzoperation during recent years [14,15,16].

Methanol, like in the same case of ethanol, from recent years is investigated about the option of using it like primary fuel in light-duty diesel dual fuel engines giving very good results. Many techniques are being investigated, but the option of replace diesel in a totally way from now is not an option feasible. Nevertheless, ethanol is better option than methanol since the toxicity of the last one [14,16].

1.3 Dual Fuel Concept

The dual fuel system is a retrofit to the CI engine. The dual fuel engine works using a CI engine but operates on a combustion process with characteristics from both SI and CI engines. In dual fuel diesel engine, a carburetted mixer of air and high-octane index fuel is compressed like a traditional diesel engine. The compressed mixture of air and vaporous fuel does not auto ignite because of its high auto ignition temperature. Therefore, it is let go by a jet of liquid fuel which spontaneously ignites the charge toward the end of compression. The gas–air mixture in the region of the injected spray ignites at number of points initiatiating various flame fronts[17]. Hence, the combustion process starts smoothly and rapidly. The amount of liquid fuel needed for sufficient ignition is between 10% and 20% of the amount needed for operation on liquid fuel alone[18]. 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 disadvantage is the necessity to have liquid fuel available for the dual fuel engine operation[19]. 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

One major influences of the dual fuel gas-diesel engine are reduced PM and NOx emissions, also decreased fuel cost due to substantially lower cost of natural gas compared to diesel. Other potential influence are increased thermal efficiency, which is load dependent, and decreased fuel transport impacts if natural gas is available locally. The maintenance cost after the conversion will not increase as the majority of the engine parts remain unchanged. Since a significant amount of diesel is displaced by natural gas, up to 70%, carbonization is reduced in the dual fuel engine. Therefore, the number of times the engine needs to be de-carbonized and overhauled is reduced [20]. In the dual fuel natural gas diesel engine, the transitions between diesel and dual fuel modes can be achieved while the engine is running without any interruption to the required engine load. If natural gas is not available, the control valve will be shut off and the engine continues on diesel fuel as a conventional diesel engine.

The application of diesel engines employing natural gas ranges from on-site power production to transportation vehicles. Dual fuel technology has also been used in land-based oil, well-drilling and fracking applications itself. Dual fuel is implemented on the engines that drive the hydraulic fracturing pumps, normally run on 100% diesel. This reduces the truck traffic by requiring that less diesel be trucked in to run the hydraulic fracturing trucks. By applying dual fuel engines in tanker trucks and using natural gas near the well site, truck traffic and fuel cost can be saved tremendously. In addition, the natural gas from the formation is also useful to power the high-pressure pump and diesel engine in the fracturing trunks[21].

1.4 Effects and standards of Exhaust emission

Figure 1.2 represents a breakdown of country's percentage of global energy-related CO2 emissions from fuel combustion 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%). The effects of greenhouse and other emissions are alarming. These emissions can result in unhealthy levels of air pollution, global climate change, acid rain and respiratory problems[22].

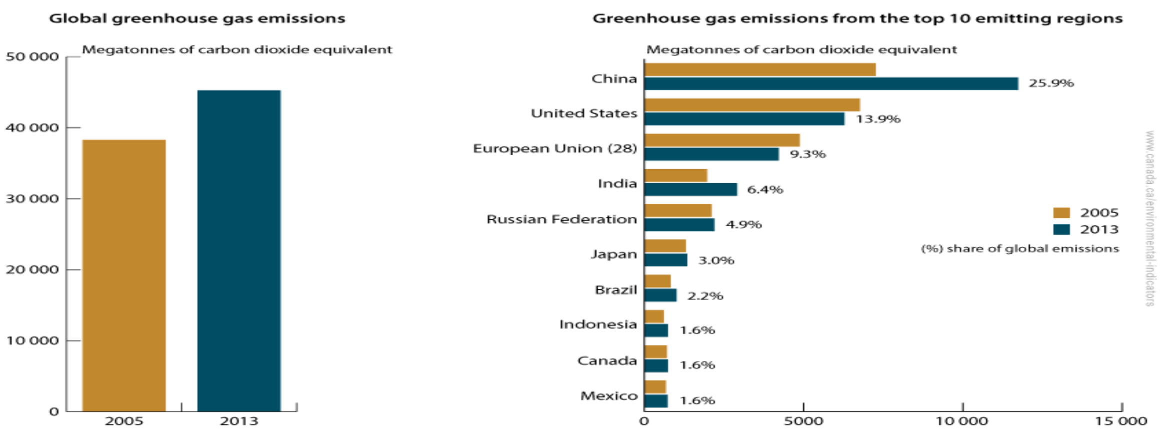

Figure 1.2: Global carbon dioxide emissions from fuel combustion (in megatons of carbon dioxide equivalent)[22]

The health effects of these hazardous emissions are both acute from short-term and chronic for long-term and repeated exposure. Therefore, it is crucial to control the production of air pollution. One way to reduce the emissions is by engine design and control of engine parameters but quite often, this will create other adverse effects. PM emission in the CI engine cannot be reduced to an acceptable level solely by engine design and control. It typically requires an after treatment system to reduce emission levels below regulator standards. Emission limits are implemented to regulate and control emissions from light-duty, heavy-duty and stationary engines.

Another concern for manufactures and operators of CI engines is the fuel cost. Diesel fuel cost is generally high and increases when the high worldwide demand for diesel fuel and other distillate oil is increased, especially in Europe, China and the USA[23]. In the USA, on-highway diesel fuel prices have been continuously higher than gasoline since September 2004 [24]. Since alternative fuel such as natural gas is considerably lower, utilizing natural gas in diesel operations may promote fuel cost savings.

In the effort to reduce undesirable emissions and expensive diesel fuel cost, many have proposed alternative ways to combine cleaner and lower cost gaseous fuel with diesel fuel as a diesel fuel supplement.

Emission standards are legal requirements governing air pollutants released into the atmosphere. Emission standards set quantitative limits on the permissible amount of specific air pollutants that may be released from specific sources over specific time frames. Emission may be divided into two categories: visible emissions and invisible emissions[17]. Smoke and particulate matter is classified as visible emissions whereas carbon dioxide, carbon monoxide, oxides of nitrogen, unburnt hydrocarbons, carbon monoxide and aldehydes are classified as invisible emissions. They are basically construct to meet air quality standards and thererby protect human health. The primary measure to control vehicular air contamination was taken in USA 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, Euro 6 emission standards are executed in the vast majority of nations over the globe as demonstrated in Table 1.1 and 1.2[25].

Table 1.1 European Standards for passenger cars, g/km [25]

Description	Vehicle type	CO	HC	NO_x	HC + NO_x	Particulate matter
1992-Euro 1	All	2.72	-	-	0.97	0.14
1996-Euro 2	Gasoline	2.20	-	-	0.50	-
Diesel	1.00	-	-	0.70	0.10	
2000-Euro 3	Gasoline	2.30	0.2	0.15	-	-
Diesel	0.64	-	0.50	0.56	0.05	
2005-Euro 4	Gasoline	1.00	0.1	0.08	-	-
Diesel	0.50	-	0.25	0.30	0.025	
2009-Euro 5	Gasoline	1.00	0.1	0.06	-	-
Diesel	0.50	-	0.18	0.23	0.025	
2014-Euro 6	Gasoline	1.00	0.1	0.06	-	-
Diesel	0.50	-	0.08	0.17	0.025	

Table 1.2 Emission Standards for diesel and gas engines, g/kWh [25]

Description	CO	Non-methane hydrocarbon	Methane(1)	NO_x	Particulate matter(2)
Euro 3. Oct. 1999	3	0.4	0.65	2	0.02
Euro 3. Oct. 2000	5.45	0.78	1.6	5	0.16
Euro 4. Oct. 2005	5.45	0.55	1.1	3.5	0.03
Euro 5. Oct. 2008	4	0.55	1.1	2	0.03
Euro 6. Jan. 2013	4	0.16(3)	0.5	0.4	0.01(4)

1.5 Problem Statement

The use of diesel fuel in CI engine releases pollutants such as particulate matter, oxides of nitrogen, carbon monoxide, unburned hydrocarbon and carbon dioxide. The quantity of pollutants released is highly dependent on the oxygen level, combustion temperature and type of fuel. These pollutants have detrimental effects on human health and the environment. Exposure to these emissions may lead to health ailments that reduce labour productivity, thereby leading to a rise in poverty level. In addition, pollutant accumulation results to rise in emission of greenhouse gases causing global warming. These impacts coupled with the challenges of escalating fuel prices due to increase in demand create a need to produce and utilize alternative fuels such as biogas[26].

The use of biogas in CI engines requires a means of admitting the gas into the engine combustion chamber and a way of regulating the pilot fuel, implying that biogas cannot be used in the diesel engine without modification. These challenges demonstrate that there is a need to develop a dual fuel engine that would utilize biogas as an alternative fuel. Higher fuel costs and increasingly stringent emission regulations are concerns for manufacturers and operators of compression ignition engines. By converting the CI unit to dual fuel operation and utilizing lower cost natural gas, both lower fuel costs and lower emissions can be achieved. However, the cost savings through diesel displacement and emission reductions need to be quantified. Thus, the main purpose of this research is to address the challenges associated with dual fuel engines in order to improve efficiency and to obtain better control of the exhaust emissions. Since dual fuel engines produce higher HC and CO emissions than regulations allow, concepts should be developed to advantageously control combustion and emissions. Therefore, the primary challenges of this research and hypotheses to the questions are as follows:

1) Does dual fuel operation save cost compared to diesel operation?
Hypothesis: Over the past five years, natural gas fuel prices have remained well below that of diesel fuel and have been more consistent compared to diesel fuel. When natural gas is introduced in diesel engine, it should displace some of the diesel fuel in CI engine, therefore reduce the fuel cost in dual fuel engine.
2) Do dual fuel engines reduce NOx and PM emission compared to diesel engines?
Hypothesis: Being a lowest member in paraffin family, the diesel combustion with natural gas in dual fuel engine is expected to produce less PM. In addition, since the premixed combustion happens, the opportunity for locally rich mixtures to form is reduced. Therefore, with lean conditions, the flame temperature should be lower, thus reducing the NOx formation in a dual fuel engine.
3) Do dual fuel engines emit excessive HC and CO?
Hypothesis: The ratio for the natural gas and air entering the cylinder is not controlled. It is speculated that at these load conditions, the natural gas air mixture is below the flammability limit in the cylinder and the penetration of the diesel fuel jet does not extend across the cylinder. Consequently, regions near the edge of the combustion chamber with natural gas and air mixtures below the flammability limit are either unburned (HC emission) or partially burned (CO emission).

1.6 Research Objectives

The principal objective of this research was to modify a diesel engine to run on biogas and diesel fuels as a dual fuel engine with diesel as the pilot fuel and biogas as the primary fuel and to achieve optimum thermal efficiency with reduced gaseous emissions.

To achieve these objectives, the following specific objectives were to be accomplished:

1. To design and fabricate an efficient mixing device for biogas and air for the dual fuel engine.
2. To evaluate the performance of the modified dual fuel engine and compare with that of the diesel engine.
3. To evaluate the biodiesel based pilot fuel and blend of biodiesel and bioalcohol based pilot fuel for the biogas run dual fuel diesel engine based on performance and emission characteristics.

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 pilot 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 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

Much of the early development of the Internal Combustion engines 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 revolutionised before World War-II in countries like Italy, Russia, Germany, Holland, the U.S.A and the U.K. During World War-II, there was much activity to utilize gaseous fuel engine applications as liquid fuels of right quality became scare especially in countries like 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 objectives drawn in the earlier chapter are justified by identifying the key scopes of works from the literature review.

Chapter Outline:

2.1 Dual Fuel Diesel Engine

During the second half of 19th century, many different configurations of internal combustion (IC) engines were built and tested. These engines operated with variable success and fidelity using different mechanical system and engine cycles. The first practical engine was invented by J. J. E. Lenoir in 1860[27]. 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 main feature of this engine was that the fuel consumption was about half that of the Lenoir engine. Around the same time, the commercial exploitation of oil well have 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. The modern compression ignition (CI) was developed from the work of two people, Alkroyd Stuart and Rudolf Diesel. Alkroyd Stuart's engine was patented in 1890 and produced in 1892. It was a basically a four stroke compression ignition engine having a compression ratio of 3. The limitation of this engine was that the compression ratio was too low to provide spontaneous ignition of the fuel upon compression. This required an external heating system. Later on, Rudolf Diesel found a way to eliminate this limitation. Diesel's concept of compressing air to such an extent that fuel would spontaneously ignite after injection was published in 1890 and patented in 1892. The prototype ran with an efficiency of 25%, about twice the efficiency of any contemporary power plant[28]. 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[29]. The earliest experiment performed on dual fuel diesel engines by Cave dates back to 1929 using hydrogen gas. In 1939, the first commercial dual fuel engine fuelled by town gas was commercialised by the National Gas and Oil Engine Co. in Great Britain[19]. The main driving force that initiated an 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.* [30] defined a dual fuel engine as an ideal multi-fuel engine that operates effectively on a wide range of fuels including the flexibility of operating as a conventional diesel engine. During dual fuel operation, a carbureted mixture of air and high octane index gaseous fuel is sucked and compressed the same way air is sucked in a conventional CI engine. The compressed mixture of air and fuel-gas does not auto-ignite due to poor ignition quality of the gaseous fuel. Hence, it is ignited by a small liquid fuel injection, known as a pilot fuel, which ignites spontaneously at the end of the compression phase. A dual fuel engine therefore uses a primary and secondary fuel. The primary fuel is the pilot fuel such as diesel and the secondary fuel may be natural gas (NG), liquified petroleum gas (LPG), hydrogen or biogas [31].

In 2006, J. Stewart *et al.* [32] modified a direct-injection CI engine into a dual fuel engine and fueled it with three different gaseous fuels: Methane, propane, and butane to investigate performance at various gaseous concentrations. They reasoned that a simple central point mixing system is the most inexpensive and straightforward method of admitting a gaseous fuel to the dual fuel engine. A simple venturi type gas mixer was therefore installed at a distance of ten times the gas pipe diameter, upstream of the inlet manifold to ensure complete mixing of air and the fuel was achieved. Engine performance data were obtained under steady state operating conditions at three loads corresponding to quarter, half and three quarters load (relative to 100 percent loading being 18.7 kW). They concluded that propane showed the most promising characteristic as a dual fuel engine fuel, with a reduction of up to 20 percent energy consumption being recorded. This was attributed to the enhanced reactivity of the fuel. Methane which has been the fuel of choice was found to increase the brake specific energy consumption for all cases considered. The study also stated that the engine clearly showed the benefits of reduced CO2 emission which must be considered if proposed global reductions are to be achieved within the transport sector. Emission of CO2 was reduced by up to 20 percent. This showed that dual fuel engines can be used to help achieve reduction in exhaust emissions. The study also recommended that better engine performance and further reduction in CO2 emission could be achieved with the employment of modern fuel injection equipment such as high pressure common-rail and multiple injection strategies. Research has shown that dual fuel engines already operate under the NO*X* and SO*X* limits set out by the International Maritime Operations Tier 3 regulation, representing an interesting alternative to diesel engines equipped with EGR or exhaust gas after- treatment [33]. According to a comprehensive simulation study by C. Christen *et al.* [33], a homogeneous mixture of fuel gas and air is crucial in order to achieve low NO*X* emission as well as reducing the risk of knocking combustion. Continuous admission of fuel gas was found to achieve best mixing results. Too much gas admission led to increased accumulation of rich fuel gas in cylinder crevices which was a main source of unburned hydrocarbons.

2.3 Combustion Characteristics

The study carried out by Mansour *et al.* [34] indicates that the dual fuel combustion is far more complex than pure diesel operation. Cylinder pressure data analysis is an important tool to diagnose the engine combustion behaviour as cylinder pressure history directly influences the engine performance and the emission characteristics. The combustion processes in CI engine comprises of four stages as shown 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. They are as follows: AB is the pilot fuel ignition delay, BC is the pilot premixed combustion phase, CD is the primary fuel ignition delay, DE is rapid combustion of primary fuel and EF is the diffusion combustion phase.The critical analysis of *P-θ* diagram diagrams reveals the following points:

1. The peak cylinder pressure of diesel mode is found to be higher than that of dual fuel mode.
2. The pressure rise rate of diesel mode is found to be lower than that of dual fuel mode.
3. The shift of peak cylinder pressure away from top dead centre (TDC) is more in case of dual fuel mode in comparison to that of diesel mode.

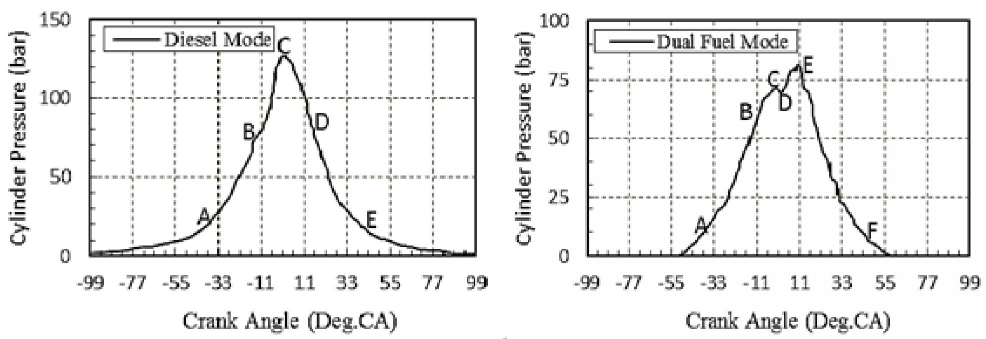

Fig. 2.1 Comparison of *P-θ* diagram for diesel and dual fuel mode (Mansour *et al.*, 2001)

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.

2.4 Engine Modifications

A simple diesel engine can be easily converted into dual fuel diesel engine carrying out the following modifications as given below:

1. Carburetor or a gas mixer needs to be connected at the inlet manifold for mixing of gaseous fuel and air[35]. The carburetor or 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[35].

2.5 Biogas Run Dual Fuel Diesel Engine

Biogas, a renewable gaseous fuel, can be successfully used in CI engines with minor modifications. This is particularly important in the case of developing countries where meeting the growing demand of fossil fuel is a major challenge[36].

A performance evaluation of a constant speed IC engine on compressed natural gas (CNG), methane enriched biogas and biogas was done by R. Chandra *et al*. [37]. A venturi type air intake and fuel gas supply system was used to supply the gaseous fuel into the engine. The observed loss in brake power due to conversion of diesel engine into spark ignition engine were 31.8%, 35.6% and 46.3% for compressed natural gas, methane enriched biogas and raw biogas, respectively. The results indicate that the power losses were quite high, with the highest loss recorded for raw biogas. Provision of a mixing chamber in the setup instead of a venturi type air intake could give a more homogenous air-fuel mixture resulting in better combustion and improved engine performance. This is because a mixing chamber provides longer retention time of air and fuel inside it leading to a homogenous mixture with better combustion ability [18].

In a study by S. Siripornakarachai *et al*. [38], a bus diesel engine was modified to use biogas as fuel for electricity production in a farm. Modifications included addition of biogas carburettor for air-fuel mixing, replacing the fuel injection system with spark ignition system and reduction of compression ratio from the original 16:1 to 8:1 using a cylinder head spacer. The results showed that there was high emission of CO, though the engine power output was 134.20 kW which is satisfactory. Literature suggests that the compression ratio should be between 10:1 and 12:1 for biogas operation [18, 39]. Since the compression ratio was lowered to 8:1, this could have led to incomplete combustion hence increase in CO emission. In another study by H.S. Sorathia *et al*. [40], the first and second laws of thermodynamics were employed to analyze the quantity and quality of energy in a single-cylinder, direct injection diesel engine using petroleum diesel oil and biogas as fuel. The maximum brake thermal efficiency for biogas was 27.50% as compared to 28.25% for diesel oil. For the diesel and diesel-biogas dual fuels, calculation results showed that 7.31% and 6.48% of the fuel exergy input was lost in heat transfer from the engine, respectively. The study concluded that diesel-biogas dual fuel mode produced lower energy conversion efficiency; which was offset by large replacement of diesel and induction air by biogas. The study concluded also, that biogas premixed charge dual fueling for the engine produced almost no performance deterioration at all test speeds. The results of this theoretical thermodynamic analysis are impressive and are worth validating through experimental methods. N.S. Ray *et al*. [41] modified a CI engine into a dual fuel engine and used biogas for partial substitution of diesel to study the effect on performance of the engine. The study indicated good performance of the modified engine in terms of power output as well as reduction of exhaust emissions. There was a reduction in both CO and HC emissions, though the brake engine power decreased slightly with increase in percentage of biogas. This was due to lower energy content of biogas compared to diesel. It was concluded that 50% substitution of diesel with biogas was the optimum ratio for dual fuel operation. The highest ratio tested was Biogas:Diesel in the ratio 50:50. There is therefore need to investigate performance of the engine at higher substitution ratios such as Biogas:Diesel in the ratio 90:10.

Due to this fact, there has been a continuous refinement of the biogas run dual fuel engines over the years. Many researchers have studied the performance and emission characteristics using different quality of biogas and different types of pilot fuel as summarised in Table 2.1

2.6 Biogas Quality Limit

Biogas mainly contains methane (CH4) and carbon dioxide (CO2). The combustion value of any quality of biogas is directly related to its methane composition[42].However, presence of CO2 up to 30% in biogas for dual fuel application can improved the performance of the engine. Biogas containing more than 40 % of CO2 needs scrubbing. Conversely, biogas containing more than 45% of CO2 results in harsh and irregular running of the engine[43]. The properties of biogas used by different researchers for biogas run dual fuel applications are given in Table 2.2.

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

Researcher(s)	Test Engine		Type of biogas	Pilot Fuel
Bedoya *et al.[42]*	Manufacturer	Lister Petter	Simulated biogas	Diesel, Palm oil biodiesel
	No. of cylinder	2		
	Stroke	4		
	Type	DI, naturally aspirated, air cooled		
	Swept volume	1550 cm^3		
	Bore × stroke	98 mm × 101 mm		
	Compression ratio	15.5:1		
	Rated power	20 kW at 3000 rpm		
Luijten and Kerkhof[46]	Type	12GF-SF	Simulated Biogas	Diesel, Jatropaha oil
	No. of cylinder	1		
	Bore × stroke	110 mm × 115 mm		
	Compression ratio	17:1		
	Rated power	12 kW		
Sahoo *et al*.[19]	Manufacturer	Kriloskar	Raw Biogas	Diesel,Jatropha biodiesel
	Model	TV1		
	Type	DI, natural aspirated, water cooled		
	No. of cylinder	1		
	Bore × stroke	87 mm × 110 mm		
	Swept volume	661mm3		
	Compression ratio	17.5		
	Rated power	5.5kW @1500 rpm		

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

Researcher(s)	Test Engine		Type of biogas	Pilot Fuel
Yoon and Lee[44]	Type	Turbo charged, water cooled	Raw Biogas	Diesel, Soybean biodiesel
	No. of cylinder	4		
	Bore × stroke	91.1 mm × 95 mm		
	Swept volume	2476 cm3		
	Compression ratio	19:1		
	Maximum power	46 kW at 4000 rpm		
Mustafi *et al*[36]	Manufacturer	Lister Petter	Simulated Biogas	Diesel
	Type	DI, water cooled		
	Compression Ratio	16.5:1		
	No. of cylinder	1		
	Bore × stroke	87.3 mm ×110 mm		
Barik and Murugan[45,48]	Manufacturer	Kriloskar	Raw biogas	Diesel, Karanja biodiesel
	Model	TAF1		
	Type	DI, air cooled, naturally aspirated		
	No. of cylinder	1		
	Bore × stroke	87 mm× 110 mm		
	Swept volume	661 mm^3		
	Compression ratio	17.5		
	Rated power	4.4 kW @1500 rpm		

Table 2.2 Summary of Biogas properties

Researchers	Type of biogas	Composition	Calorific value (MJ/kg)	Air-fuel ratio	Boiling point (°C)	Density (kg/m^3)	Octane Number	Auto ignition temperature (°C)	Woobe Index (kWh/Nm^3)
Bedoya et al. [42]	Simulated biogas	60% CH_4, 40% CO_2	23.73	6.05	-	-	-	-	6.16
Yoon and Lee [44]	Raw biogas	30-73% CH_4, 20-40 CO_2, 5-40 N_2,1-3H_2	26.17	17.2	180-330	0.65-0.91	130	632-813	-
Sahoo[19]	Raw biogas	48% CH_4, 52% CO_2	15.74	-		1.2	-	-	-
Mustafi et al. [36]	Simulated biogas-I	39.9% CH_4, 60.1% CO_2	-	-	-	-	-	-	-
	Simulated biogas-II	49.7% CH_4, 50.3% CO_2	-	-	-	-	-	-	-
	Simulated biogas-III	59.9% CH_4, 40.1% CO_2	-	-	-	-	-	-	-
Luijten and Kerkhof [46]	Simulated biogas-I	65% CH_4, 35% CO_2	-	-	-	-	-	-	-
	Simulated biogas-II	85% CH_4, 15% CO_2	-	-	-	-	-	-	-
	Simulated biogas-III	95% CH_4, 5% CO_2	-	-	-	-	-	-	-
Barik and Murugan [45,48]	Raw biogas	-	27.53	17.2	-	1.2	130	600-650	-
			17.2	15.3	-	1.31	110	640-670	-

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

The parameters include operating parameters, engine components, biogas quality, biogas inlet pressure and oxygenated combustion on the performance, combustion and emission characteristics of biogas run dual fuel diesel engines. Studies on operating parameters like load and speed reflect the manoeuvring 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

Sahoo demonstrates experimentally that brake power (BP) and torque increases with the increase of load at constant speed for both diesel and DFM indicating both BP and torque have a linear relationship with load[19].

2.7.2 Air-Fuel Ratio

Effect of Load:

The relative portion of the fuel and air are very vital from the viewpoint of combustion and efficiency of the engine. This is expressed either as a ratio of the mass of the fuel to that of the air or vice versa[17]. The ratio of actual fuel-air ratio to stoichiometric fuel air ratio is called equivalence ratio. The air fuel ratio for SI engines ranges from 12 to 18. However, the air fuel ratio for CI engines ranges from 18 to 80 from full load to no load. However, the operation of dual fuel diesel engines can take place over wide range of air-fuel ratio. Yoon and Lee observed that the equivalence ratio under DFM is found to be higher in comparison to that of diesel mode. This is due to the fact that biogas is inducted into the engine through the air inlet manifold. This results in lowering of the volume of air entering into the combustion chamber[44]. Barik and Murugan found the stoichiometric air fuel ratio to decrease with the increase of load. This is due to the fact that as more amount of biogas flows needs to be supplied to the engine at higher load[45].

Effect of Air Induction System:

Bedoya *et al*. compared the performance of a biogas run dual fuel engine using two induction systems. 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[42].

Effect of Biogas Quality:

Luijten and Kerkhof investigated the effect of biogas quality on the performance of a dual fuel diesel engine. The study inferred that air fuel ratio was hardly got affected by low the quality of biogas[46].

2.7.3 Volumetric Efficiency

Effect of Load:

Duc and Wattanavichien found that volumetric efficiency (VE) decreased with the increase of brake torque and load respectively for both lone and dual mode[47]. Similar trend was observed by Barik and Murugan for biogas run dual fuel diesel engine using karanja methyl ester (KME)[45]. This was due to the temperature of the exhaust gases increases with load which in turn preheats the incoming air and thereby, lowering the VE. The decrease in VE was more under DFM as compared to diesel mode as reported by Barik and Murugan. This was because the biogas substitution displaces a greater portion of air under DFM[48]. 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[19].

Effect of Induction System:

Bedoya *et al*. found that there was a 6% drop in VE using the combination of Kenics 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[42].

Effect of Biogas Quality:

Luijten and Kerkhof expressed VE as a function of the heat release fraction of pilot fuel and biogas quality. The quality of the biogas slightly affects the VE. Overall, the decrease in VE was quite modest from 95% at zero substitution to 90–91% for maximum substitution even for the lowest quality of biogas[46].

2.7.4 Brake Thermal Efficiency

The brake thermal efficiency (BTE) was found to increase with load for both diesel and DFM. However, the BTEs under DFM were found to be lower than diesel mode in all the test cases. This was mainly due to low calorific value of biogas as compared to that of diesel.

The study done by Yoon and Lee[44] and Sahoo[19] 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 Kenics mixe 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[42].

Effect of Biogas Quality:

Henham and Makkar (1998) have conducted tests on the dual fuel diesel engine using synthetic biogas of various compositions. The test indicated that the overall efficiency falls with the increase of CO2 content in the composition of synthetic biogas[49]. However, Luijten and Kerkhof found that the biogas quality has marginal effect on the BTE[46].

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

Effect of Load:

From reported literature, the brake specific fuel consumption (BSFC) and brake specific energy consumption (BSEC) were found to decrease with the increase of load for both diesel and DFM. 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[44]. Their study showed that the BSFC of biodiesel-biogas was found to be lower than diesel-biogas under DFM. On the other hand, Sahoo[19] reported that the BSEC of diesel-biogas was found to be lower than Jatropha-biogas. Their study indicated a maximum liquid fuel replacement (LFR) of 69% and 66 % with a biogas flow rate (BFR) 3.66 m^3/h and 4 m^3/h for diesel and jatropha pilot fuel, respectively at 100% load.

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 pilot 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[48].

Effect of Biogas Quality:

Bari observes that the trend of BSFC rises as CO2 increases in the biogas composition.This is due to the fact that as the CO2 becomes higher in the biogas, CO2 remains undissociated and acts as an inert gas. Addition of such inert gas affects the burning velocity leading to incomplete combustion, thereby increasing the BSFC and BFR[43].

Mustafi *et al.*, found BSFC to be higher under DFM in comparison to that of diesel mode. At the same loading conditions, the BSFC increased as the quantity of CO2 in the biogas composition was increased[36].

2.7.6 Exhaust Gas Temperature

Effect of Load:

The exhaust gas temperature (EGT) is found to rise with the increase of load for both diesel and DFM. However, the review on EGT under DFM in comparison to diesel mode exhibited contrasting findings. Few studies [44,45,48] reported a lower EGT under DFM whereas Sahoo reported a higher EGT under DFM in comparison to that of diesel mode[19].

Effect of Air Induction System:

Bedoya *et al.*, reported that the combination of Kenics 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[42].

Effect of Biogas Quality:

Henham and Makkar found that the EGT to be more affected by biogas substitution rather than biogas quality upto 45% replacement. At 58% replacement, EGT increased with the increase of CO2 in biogas mixture[49].

2.7.7 Carbon Monoxide Emission

Effect of Load:

Yoon and Lee found that the carbon monoxide (CO) emission to be more incase of DFM in comparison to that of diesel mode. This was due to incomplete combustion caused by dilution of charge by the CO_2 present in biogas and deficiency of oxygen. Hence, the flame formed in the ignition region of the pilot fuel was normally suppressed, and did not proceed until the biogas fuel air mixture reached a minimum limiting value for autoignition[44]. Conversely, Sahoo observed that there was a decrease of CO emission by 50% and 16% for biogas-diesel and biogas-jatropha biodiesel, respectively in comparison to diesel mode at 100% load[19].

Barik and Murugan reported an increase of brake specific carbon monoxide (BSCO) emissions by 9%, 16.6%, 28.5% and 37.5% than KME operation for biogas energy shares of 11.8%, 21.8%, 30.2% and 37.4%, respectively at full load[45]. Further, Barik and Murugan observed that the CO emission to be higher by about 24% with biogas at the flow rate of 1.2 kg/h, in comparison with diesel 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. At 40% load, CO emissions were increased with the SM2 system due to an increase in partial oxidation phenomena of methane, lower pilot fuel quantity and higher total fuel–air equivalence ratio. This occurred because the reactivity of biogas–air mixture on the compression stroke was increased, and the pilot fuel quantity was reduced with the SM2 system. At higher loads, CO emissions increased for both the SM1 and the SM2 systems due to the reduction of unburned methane and lower oxygen availability at higher equivalence ratios. However, the SM2 system allowed a reduction in CO emissions in comparison with the SM1 system, close to 40% at 100% load. It was due to higher burning rates and higher availability of oxygen in later stages of combustion[42].

Effect of Biogas Quality:

Mustafi *et al.*, observed a slower increase of CO emissions as the percentage of CO_2 content increased in the fuel (Fig. 2.58). This may be due lowering of combustion efficiency as the amount of CH_4 got reduced with the decrease of biogas quality[36].

2.7.8 Carbon Dioxide Emission

Effect of Load:

The carbon dioxide (CO2) emission is found to increase with the increase of load for both diesel and DFM. However, the CO2 emission is found more in case of DFM in comparison to that of diesel mode. This is because biogas mainly contains methane and CO2. Yoon and Lee found that the combination of biogas-biodiesel emitted more CO2 in comparison to biogas-biodiesel[44]. Conversely, Sahoo reported that the combination of biogas-diesel emitted more CO2 emission than biogas–biodiesel especially at higher loads[19].

In another study, Barik and Murugan used KME as pilot fuel under DFM. The test indicated that the brake specific carbon dioxide (BSCO2) decreased with the increase in the biogas energy share. The biogas energy share of 11.8%, 21.7%, 30.2% and 37.4% produces a drop of BSCO2 by 25%, 33%, 52% and 60% respectively, than that of KME at full load. This reduction in the BSCO2 emission under DFM was attributed to the lower volumetric efficiency and higher CO2 in biogas. The same group of authors observed that the CO2 emission rises with the increase in biogas flow rate for the same loading conditions using diesel as pilot fuel[45].

2.7.9 Hydrocarbon Emission

Effect of Load:

Yoon and Lee found a decrease of hydrocarbon (HC) emission with the increase of load for both diesel and DFM[44]. However, Sahoo observed that the HC emissions were higher at low loads as the combustion chamber temperature was low. This gradually decreases at medium loads and then increases at high loads[19]. Barik and Murugan reported an increase of HC emission increases with load for high biogas flow rate[19].

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. At full load, the effect of dual fuel system on methane emissions was less than at part load as the equivalence ratio was closer to the stoichiometric value for both the SM1 and the SM2 systems[42].

Effect of Biogas Quality:

The HC emission is found to increases with the increase of CO_2 in the biogas composition as observed by Mustafi *et al.* . This is due to the fact that with the increase of CO_2 in the biogas composition, the turbulent flame propagation from the ignition regions of the pilot fuel is suppressed further due to the low combustion chamber temperature and low air-fuel ratio[36].

2.7.10 Oxides of Nitrogen Emission

Effect of Load:

The oxides of nitrogen (NO_X) emission mainly depend upon combustion chamber temperature which in turn depends on the applied load. Therefore, there is an increase in NO_X emission with the increase of load for both diesel and DFMs. However, the NO_X emission is lower in case of dual fuel operations. This is due to the fact the presence of CO_2 in biogas lowers the combustion temperature[50]. Yoon and Lee reported that diesel-biogas produced lower NO_X emission than soybean biodiesel- biogas for the same dual fuel engine[44]. In a different pilot fuel study, Sahoo found that diesel-biogas produced higher NO_X emission than jatropha biodiesel-biogas[49].

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. The results indicated a reduction of 24%, 29%, 34% and 39.5% in the BSNO emission under DFM with the biogas energy share of 11.8%, 21.8%, 30.2% and 37.4% respectively, in comparison to KME in lone mode at full load. 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 1.2 kg/h lowers at full load[45].

Effect of Biogas Quality:

The NO_X emission decreased with the increase of CO_2 in biogas composition as observed by Mustafi *et al.* The increase of CO_2 in biogas caused more gaseous fuel to escape the combustion process, which affected oxygen concentration followed by a decrease in overall cycle temperature. Therefore, NO_X formation reduced with the degradation of the biogas quality[36].

2.8 Scope of Work

The literature review suggests that there has been a 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 operating parameters (viz. compression ratio and injection timing of pilot fuel) of a biogas run dual fuel diesel engine have not been reported. This is very important because the operating parameters of a diesel engine are standardized based on the chemical and physical properties of diesel fuel. Therefore, running biogas under dual fuel mode at standard diesel setting will not offer the best performance. Hence, the operating parameters need to be adjusted in order to obtain efficiency comparable to that diesel fuel.

The use of emulsified fuel as a pilot fuel for a biogas run dual fuel diesel engine is not found in any archival literature. This is important as the use of emulsified fuel as pilot fuel may further lower the NO_X and soot emissions.

The effect of compression ratio and injection timing on the energy and exergy distribution of a biogas run dual fuel diesel engine is not known. This is vital as the second law analysis determines the availability loss or destroyed in the various parts of an engine, and thereby estimates the maximum possible performance of a thermodynamic system.

2.9 Justification of Present Study

From the foregoing review, use of both liquid and gaseous bio-fuels in the IC engine has been the focus of researchers in an attempt to find solutions to the challenges of environmental concerns and depletion of petroleum resources. Researchers have also experimented on the use of various gaseous fuels such as hydrogen, acetylene and compressed natural gas (CNG) in the IC engine as an alternative to the conventional diesel and petrol fuels. However, the high fuel costs and increase in the NOX emissions associated with these fuels still remains a challenge. In addition, some of the gases such as acetylene, despite their remarkable combustion properties require special handling due to very high flammability. Biogas, which is produced by anaerobic fermentation of organic material has been identified as an alternative clean energy resource for the CI engine in view of its environmental friendly nature.

Dual fuel engines have been a subject of high interest due to their potential to reduce smoke emission with improved performances. This can be achieved through proper design of the mixing device for air and the gaseous fuel leading to improved thermal efficiency as well. The challenges involved in the modification of a diesel engine into a dual fuel engine results to;

emission of particulates and carbon monoxide and loss of engine power due to inefficient mixing of air and the fuel gas for complete combustion.

From this review, it is evident that sufficient research has not yet been done to determine the optimum performance of a CI engine when run on biogas while keeping the exhaust emissions as low as possible. Some of the research work have also used simulated biogas by mixing methane and carbon dioxide in certain proportions and such may not give accurate results for engine performance on biogas. Though emission of oxides of nitrogen from dual fuel engine running on biogas is relatively low compared to that for pure diesel mode, Exhaust Gas Recirculation (EGR) can be applied as a means of further reduction of NO_X emission. This has not been considered by most of the researchers. In addition, it is also important to investigate the effect of EGR on engine performance parameters such as brake power, brake specific fuel consumption and brake thermal efficiency.

Use of biogas as an alternative fuel for automobiles still remains a challenge due to the problem of compact storage. However, as researchers seek to address this challenge, application can be made to stationary IC engines such as in power generation, hoisting in construction sites and for pumps where biogas can be produced and used at the point of production. The fact that there are no locally available engines that run on biogas and diesel as dual fuel engines provided a platform to modify a CI engine into a dual fuel engine that uses biogas and diesel. The procedure adopted for the modification of the engine and thereafter its performance measurements, are presented in the next chapter.

Chapter 3

EXPERIMENTAL DESIGN AND METHODOLOGY

EXPERIMENTAL DESIGN AND METHODOLOGY

Overview

In this chapter, the modification of a Variable Compression Ratio engine into a dual fuel engine for diesel and biogas is presented. The engine was modified to use biogas, with the focus of maintaining high thermal efficiency while reducing harmful exhaust emissions. The procedure used in the design and fabrication of various components as well as the set up of the experiment showing the engine and its auxiliary components are shown. Thereafter, the present chapter discusses the details specification of the engine, devices and instruments used for conducting these experiments. 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 analyser 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

The experimental setup consists of 3.5 kW single cylinder, four stroke, direct injection (DI), naturally aspirated (NA), water-cooled, variable compression ratio (VCR) diesel engine as indicated in Fig. 3.1. It is associated with eddy current and water-cooled dynamometer for stacking on crankshaft with the assistance of electromagnetic drive. A tilting cylinder block arrangement is utilized for fluctuating the CR without halting the motor and without modifying the burning chamber geometry. The fuel injector in the motor has three circular holes having 0.3 mm measurement which shower fuel with a splash edge of 120º. The piston top of the engine is bowl type. Hence, the combustion chamber is of hemispherical type when piston reaches TDC. The liquid fuel reaches engine fuel pump from fuel tank by gravity. A regulator, fixed on the panel box, controls electric supply for load variation. The load sensor, fitted with the dynamometer, sends the load signal to the digital display in kg. Instruments for combustion pressure and crank-angle measurement are provided along with the setup. The signals are interfaced to computer through engine indicator for pressure-crank angle (P-θ) and pressure volume (P-V) diagrams for each of 360° rotation of crank. There are provisions for measuring the airflow, fuel flow and temperatures. Rotameters are used for cooling water and calorimeter water flow extent. The cooling water flows through the jackets of the engine block and cylinder head to remove excess heat produced during combustion. The fuel injection pressure can be varied between 200 to 220 bar. The specification of the engine is given in Table 3.1. The specification of the engine is added in Table 3.1.

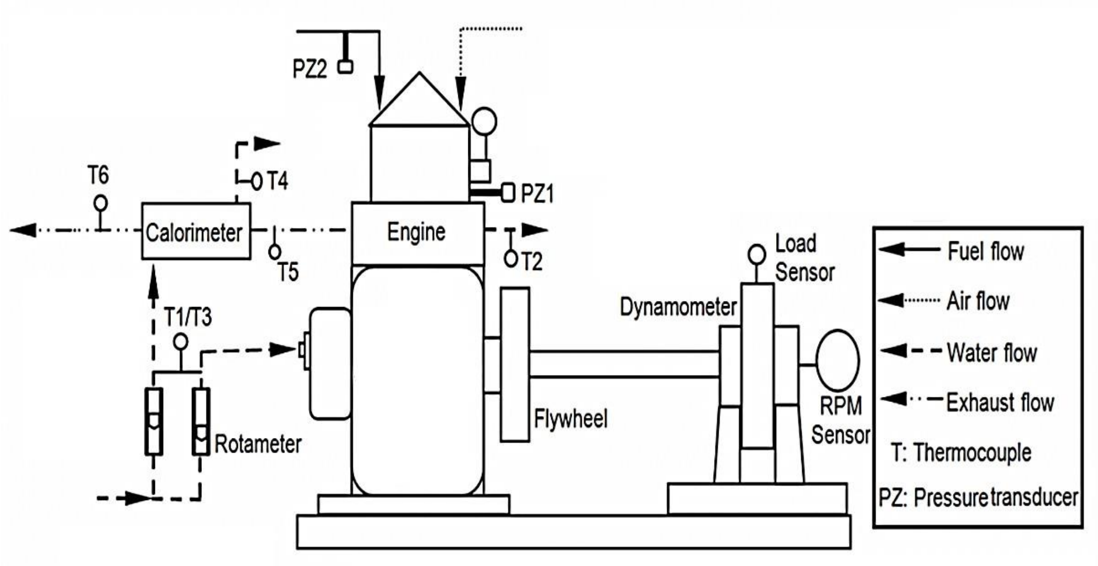

Fig. 3.1 The schematic diagram of the VCR diesel engine setup

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	Make Yokogawa, 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

The VCR research diesel engine setup is consists of a several sensors, transmitters and indicators. These are interfaced with the data acquisition device (DAD) for automatic measurement of almost all of the direct and indirect performance parameters.

Air and Fuel Flow Measurement

Both air and fuel flow measurement can be performed manually and automatically. Manual airflow measurement is carried out by recording the difference in height of water column in the manometer. It is interconnected across the orifice meter, through which air comes into the engine panel box, before leaving towards the engine manifold. Manual measurement of fuel is executed by transferring fuel from the tank through the measuring tube for known duration.

***P-θ* Measurement**

The PCB Piezotronics made two dynamic pressure sensors are fitted on the cylinder head and fuel injector. Both of them has identical specification and capable of distinguish pressure of compression, combustion, explosion, pulsation, cavitation, blast, pneumatic, hydraulic, fluidic etc. An optical crank angle sensor (Kubler make) is used to measure each degree rotation of crank with TDC pulse.

Temperature Measurement

Four PT100 temperature sensors measure the inlet and outlet temperatures of engine cooling water flow and calorimeter water flow. The inlet and outlet temperatures of exhaust gas to calorimeter are measured by two K type thermocouples. The thermocouples used in this work, have a response time more than 0.08 seconds (for the constant speed engine around 1500 rpm). Hence, they cannot show the pulsation nature of the exhaust gas in the form of temperature readings and are found almost steady after a certain time (nearly 5 minutes) at a particular load.

Compression Ratio Variation Control

The VCR diesel engine has provisions for eight step CR variation from 12 to 18. This is done by tilting cylinder head with the help of locknut and adjuster arrangement. There are six sockets headed vertical allen bolts fitted on two supporting blocks on the two sides of the cylinder, which needed to be loosen for CR variation.

Performance Measurement

The measurement of the performance parameters are 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

AVL DiGas 444 analyzer was utilized for carrying out emission analysis. Measuring the relative volumes of certain gaseous constituents in the exhaust gases of motor vehicles was the determination of AVL DiGas 444 analyzer . These gases are: carbon monoxide (CO), carbon dioxide (CO_2), hydrocarbons (HC), oxygen (O_2) and nitric oxide (NO)(Optional) The resolution, accuracy and range of these emission parameters are shown in Table 3.2. Carbon monoxide (CO) and oxides of nitrogen (NOX) are measured through electrochemical measurement cells; whereas, carbon dioxide (CO2) and hydrocarbon (HC) are measured by Infrared and Pellistor Heat Affect Detector, respectively. ASTM-D6522 standard is utilized for emission measurement by the analyzer. Working principle of the gas analyzer being as follows. The flue gas is allowed to surge through a probe and dried out by a condensation trap during steady engine operation. Thenceforth, individual sensors and readings are then displayed on the screen of the control unit after analyzing each of the CO2, CO, HC, and NOX emission concentrations present in the flue gas .

Table 3.2 The specifications of AVL DiGas 444 Analyzer

Sl. No.	Measured gas	Resolution	Accuracy	Range
1	O2	0.1%vol	< 2% vol: ±0.01% vol ≥ 2% vol: ± 5% of vol	0 – 22%vol
2	CO	0.01%vol	< 0.6% vol: ±0.03% vol ≥ 0.6% vol: ± 5% of initial value	0-10% vol
3	CO2	0.01% vol. < 25% 0.1% vol. > 25%	< 10% vol: ±0.05% vol ≥ 10% vol: ± 5% of vol	0 – 20%vol
4	NO	1 ppm vol	< 500 ppm vol: ± 50 ppm vol ≥ 500 ppm vol: ± 10% of initial value	0 – 5000 ppm
6	HC	≤ 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

The VCR engine test setup was modified to a biogas run dual fuel setup by connecting a venturi gas mixer at its in let manifold as depicted in Fig. 3.2. The existing setup and the components of the modified setup are shown in Fig. 3.3.

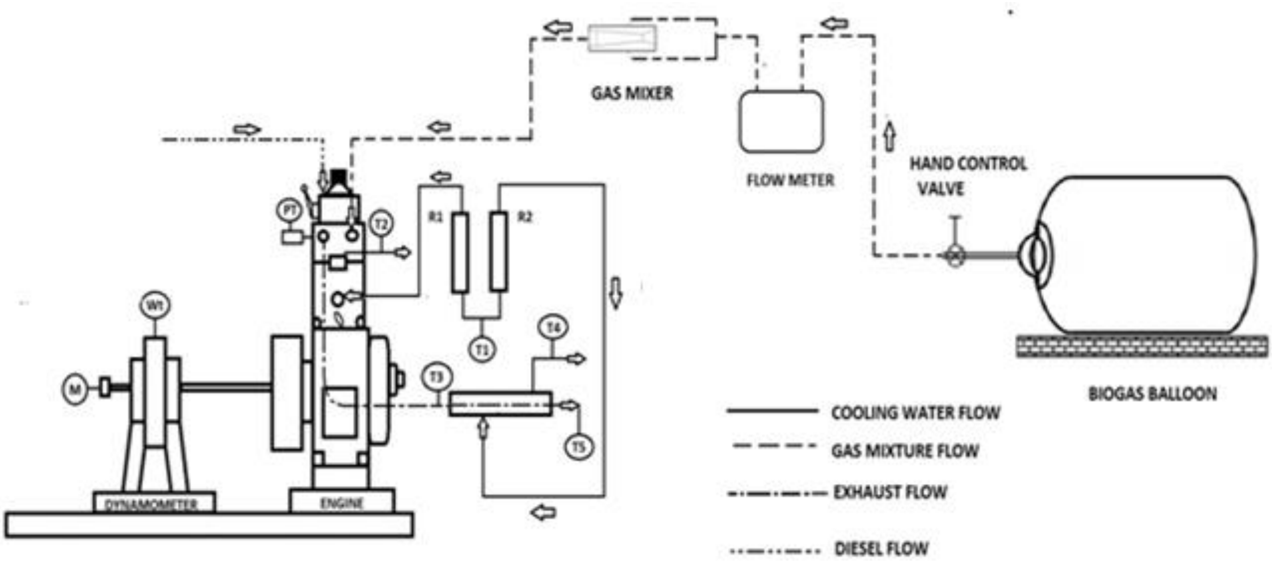

Fig. 3.2 The schematic diagram of the modified dual fuel VCR diesel engine setup

Fig.3.3 Test Setup

Fig.3.4 Modified test setup

3.3.1 The Fuel air mixing chamber

For changing over a diesel engine to dual fuel mode, the real alteration required is to associate a fuel air mixing chamber to the inlet manifold[19]. The fuel air mixing chamber is vital in dual fuel mode, as it gives a burnable blend of fuel gas and air in the required amount and quality for proficient operation of the engine under all conditions[51]. As indicated by the performance required, the flow of fuel gas can be fluctuated. It additionally empowers to supply an adequate measure of air at maximum load and speed. 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. The design of a fuel air mixing chamber for a specific engine mostly relies on upon its volumetric efficiency, rated power, speed, specific fuel consumption, swept volume and manifold connection diameter was considered for the design of fuel air mixing chamber[18].

Table 3.3 Engine Specifications

Bore Diameter	D_B	87.5 mm
Stroke Length	L	110 mm
Speed	N	1500 rpm
Power	P	5.2 KW
Specific Fuel Consumption	sfc	0.678m^3/KW-hr
Volumetric Efficiency	η_{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 venturimeter and is known as'venturi gas mixer'. Two gas inlets, one air inlet and one air-gas outlet constitute the gas mixer. It consists of a smooth contraction section and an expansion section. Because of the smoothness of the contraction and expansion, the irreversible pressure loss is low. The function of the converging portion is to increase the velocity of the fluid and temporarily lower its static pressure. Minimum pressure will be obtained at throat portion because of maximum velocity. Between inlet and throat pressure difference is developed. This rate of flow is correlated to the pressure difference. There are four main designing parameters namely converging angle (θ1), diverging angle (θ2), nozzle angle (θn) and β (ratio between the diameters of throat and inlet manifold). The values of θ1, θ2, θn and β are considered as 20°, 5°, 35°and 0.46, respectively as shown in (Fig.3.4). These geometric values are dictated by past simulation studies. The throat diameter, inlet manifold and biogas inlet diameter are 20 mm and 40 mm and 9 mm, respectively. The diameter of biogas inlet nozzle is found to 8 mm be based on the methodology as suggested by von Mitzlaff[18]. Confer to Stewart *et al.*, the length of the diverging section should be 10 times the inlet manifold diameter[52]. However, considering the diverging angle of 5° the length of the same is found to be 250 mm with respect to the manifold diameter of 40 mm. All this design parameters of the gas mixer is based on a diesel engine with rated power of 5.2 kW. The volumetric efficiency and speed of the engine are considered as 90% and 1500 rpm. The maximum diesel substitution by biogas in this engine is taken as 80%[18]

Calculations:

<u>Step- 1</u>

Swept Volume,

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

Volumetric Air Intake,

$$V_{air} = \left[{(\eta_{vol} \times V_s \times N)}/{(2 \times 60)}\right] = 0.00744\ m^3/s$$

<u>Step- 2</u>

Cross Sectional Area of Intake,

$$A_i = \left(\frac{\pi}{4} \times {d_i}^2\right) = 1.256 \times 10^{-3} m^2$$

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,

$d_2 = \left(^{4 \times A_2}/_{\pi}\right)^{0.5} = 7.65mm$

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

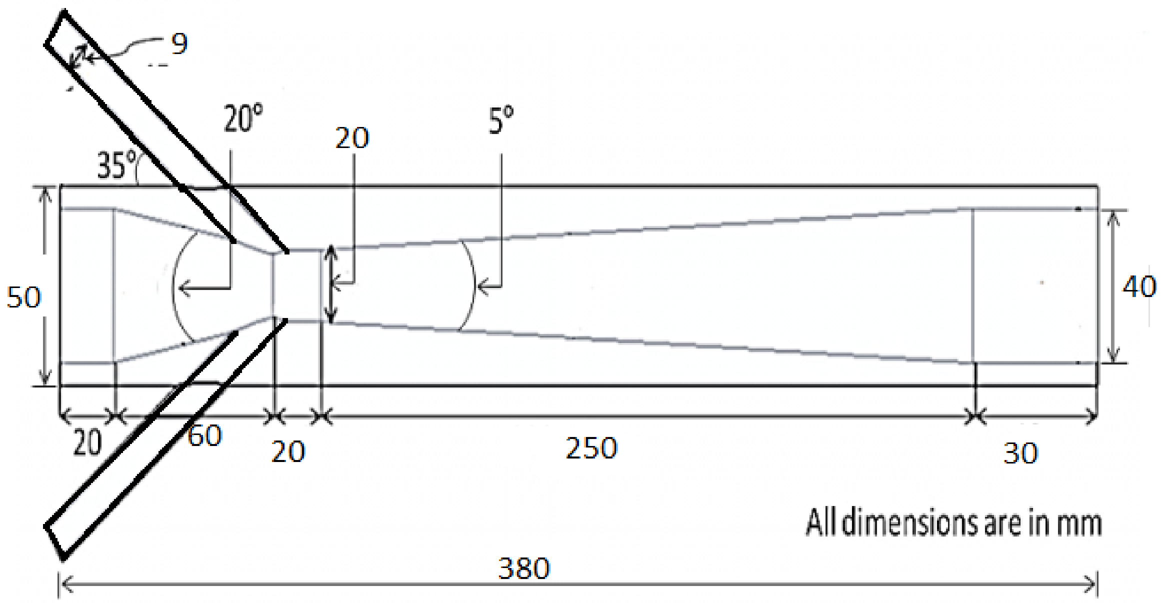

Fig.3.5 Schematic diagram of Fuel air mixing chamber

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.

Fig.3.7 Pictorial view of Y-divider

3.3.2 Manual Regulation of Pilot Fuel Quantity

The modification was done at the engine stop lever to allow for regulation of the amount of diesel fuel injected into the combustion chamber per cycle. This was to allow reduction of the diesel fuel injection quantity during dual fuel operation since diesel was to be used in small quantity as pilot fuel to ignite biogas, herby also helps to calculate the amount of the liquid fuel replacement at any specific load corresponding to any particular speed. The pilot fuel regulating device was fabricated from V-section mild steel angle line, mild steel and M16 bolt and nut. A hole was drilled in the plate which was then welded onto a section of the angle line. The nut was then welded in the hole to allow fastening or loosening of the bolt in regulating the pilot fuel quantity by adjusting the position of the stop lever. The angle line was welded onto the engine support frame with the position of the nut close to and directly in line with the engine stop lever. Fastening the bolt in the nut would push the engine stop lever, constricting the fuel pump outlet and hence reducing 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. However, to perfectly match the rpm of the diesel mode, the control lever connect to the fuel shut off valve limits the fuel further. This control lever marginally pushes the fuel control shut valve connected to the fuel pump.

Fig.3.8 Engine stop lever

Fig.3.9 Adjustable lever arm arrangement

3.3.3 Biogas Supply and Metering System

A pictorial view of the biogas supply and metering system is shown in Fig. 3.12. Biogas was used to run the modified diesel engine in dual fuel mode and was supplied to the engine from a flexible gas bag, through a flow meter to measure the amount of gas consumed by the engine. The system consisted of a flexible (collapsible) gas bag of $1m^3$ capacity, flexible clear tubes, flow meter, Y-divider and flow switch. The gas was produced from vegetable waste in a digester at the Institute of Delhi Technological University,Delhi. The flow switch (shut-off valve) was used to regulate the gas flow by turning it to quarter, half, three quarter or fully open positions and for closing the gas supply when not in use.

Fig.3.10 Biogas Balloon Storage process from plant Fig.3.11 Biogas Flowmeter

Fig.3.12 Biogas supply and metering system

3.4 Experimental Procedure

The performance of the engine was determined by utilizing diesel in single fuel mode and biogas as primary fuel and diesel as pilot fuel in dual fuel mode.The VCR engine is first run with diesel at standard diesel specification of compression ratio of 18 and IT of 23^{o}BTDC. For proper combustion of fuel the engine is initially run at no load condition for some time so that warm up is optimum. At 20%, 40%, 60%, 80%, and 100% load the engine is tested During the experiments. The engine speed reduces with increase in load. The engine consumes more fuel resulting a higher heat release in order to maintain a constant BP, thereby higher temperature inside exhaust gas, cylinder, outlet of the cooling water and. The engine is allowed for to run for few minutes at any particular specified load condition until it reaches a steady state condition. Then the readings of temperatures, rpm and load was recorded each from the indicator. The rate of diesel fuel consumption was done by noting the amount of fuel consumed in a minute. The fuel tank was filled before the start of every experiment, while keeping the tank outlet valve closed. At the start of the experiment, the fuel tank outlet valve was opened to allow fuel into the burette and to fill the fuel line. The fuel tank outlet valve was closed and the engine allowed to run on fuel from the metering burette while noting the the amount of fuel consumed in a minute. The air flow rate was recorded by measuring the difference in height of water column in the manometer. Initially, the rpm of the engine under diesel mode is noted for a particular load of the dual fuel operation. Gradually Biogas supply valve is then released. The speed of the engine increases with biogas inflow to the engine. The additional energy that the engine gets due to combustion of biogas increases the speed of the engine. Until the rpm does not rise any further, the flow of biogas is slowly increased . In the meantime, the governor tries to lower the speed of engine. However, to perfectly match the rpm at any particular load in diesel mode, the pilot fuel that is diesel in this case is slowly reduced by the pilot fuel regulating device. The pilot fuel supply is then reduced till original rpm run on diesel mode is achieved. The engine is allowed to run for few minutes and then readings are then noted from the biogas flow meter. For every test, the initial and final meter reading was recorded for 30 seconds. Subtracting the initial from the final gas reading, the volume of gas consumed by the engine was calculated. Using volume of gas consumed, density of the gas and the time taken, mass flow rate was calculated. AVL DiGas 444 analyzer was utilized for emission analysis.

Chapter 4
RESULTS AND DISCUSSIONS

4.1 SELECTION OF PILOT FUEL

Expanded industrialization and the developing transport areas overall face significant difficulties as far as vitality request and in addition expanded ecological concerns. The rising interest and the restricted accessibility of mineral oil give incentives to the improvement of option energizes from sustainable sources with less environmental effect. However, the across the board creation and the utilization of such option powers are ruined by its uncompetitive cost against the oil based powers. There is presently an escalating look for less expensive crude material for biodiesel creation. One of the possible alternatives is the use of fuels from plant based origin like rice bran biodiesel, soy seeds biodiesel, sunflower and canola biodiesel. The use of these types of biofuel as a renewable source combines the advantages of almost unlimited availability and ecological benefits such as integrated closed carbon cycle. The majority of the investigations on biogas have been directed with diesel fuel as the pilot fuel. The investigation of various sorts of pilot fuel for biogas run double fuel diesel motor is of enormous significance taking in light of late fuel emergency. The diverse pilot fuels considered for biogas run dual fuel engine are diesel(Henham and Makkar [49]; Yoon and Lee[44]; Sahoo[19]; Bari[43]; Cheng-qui *et al.*[8]; Duc and Wattanavichien[47]; Tippayawong *et al.*[7]; Bedoya *et al.*[42]; Mustafi *et al.*[36]; Barik and Murugan[45]) Soybean biodiesel (Yoon and Lee[44]; Jatropha biodiesel (Sahoo[19]); Palm oil biodiesel (Bedoya[42]) and Jatropha oil (Luijten and Kerkhof[46]). Thus, few studies have been reported on biogas dual fuel combustion using biodiesel as pilot fuel. The following chapter discusses the investigations of using rice bran biodiesel (RBB), soybean biodiesel (SBD) and blend of biodiesel and alcohol which are Soybean biodiesel-ethanol(SBDE), Rice Bran Biodiesel-Ethanol(RBBE), Soybean biodiesel-octanol(SBDO), Rice Bran Biodiesel-Octanol(RBBO) as pilot fuel for the biogas run dual fuel diesel engine.

4.2 Results of Diesel - 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 4.1 and 4.2. The variations of load are performed from no-load to full-load (12 kg) with an increment of 2.4 kg (20%). The diesel and dual fuel modes compared at standard diesel setting (i.e. Compression ratio =18 and Injection timing = 23º BTDC). The performance analysis are performed based on theoretical equations given in Appendix.

Table 5.1 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/m3)	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)

4.2.1 Performance Analysis

The performance analyses evaluated are brake thermal efficiency (BTE), brake specific fuel consumption (BSFC), volumetric efficiency (VE), exhaust gas temperature (EGT), liquid fuel replacement (LFR). The BTE increments with increase in load for both diesel and dual fuel mode as shown in Fig. 4.1. 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. The BSEC diminishes with the With increase of load for diesel and DFM, there is a reduction. At low loads, the distinction of BSEC between dual fuel mode and diesel is quite high due to low conversion of gaseous fuel to work. However, at high load i.e over 80%, there is an impressive change in BSEC of dual fuel mode due to high load forced on the engine. This encourages the high transformation of biogas fuel into work. The VE diminishes as load increases for both diesel and dual fuel mode as observed from Fig. 4.1. With increase in load,the temperature of exhaust gases increases which thus preheats the incoming air, and thereby decreasing the VE. In dual fuel mode, the biogas substitution dislodges a more prominent segment of air with the increase of load. Hence, the VE in DFM is lower than the diesel mode. At 100% load, the VEs are found to be 64.96% for DFM1 comparison to 67.19% in diesel mode. The EGT follows a linear relationship with load for both diesel and DFM. The EGT for dual fuel mode is observed to be higher than diesel mode. This is due to late combustion of biogas which shortens the duration of extraction of power by the engine from fuel. Subsequently, the combustion products in the form of gases come out at higher temperature. LFR increases with the increase in load for DFM1.The maximum LFR is found to be 76%.

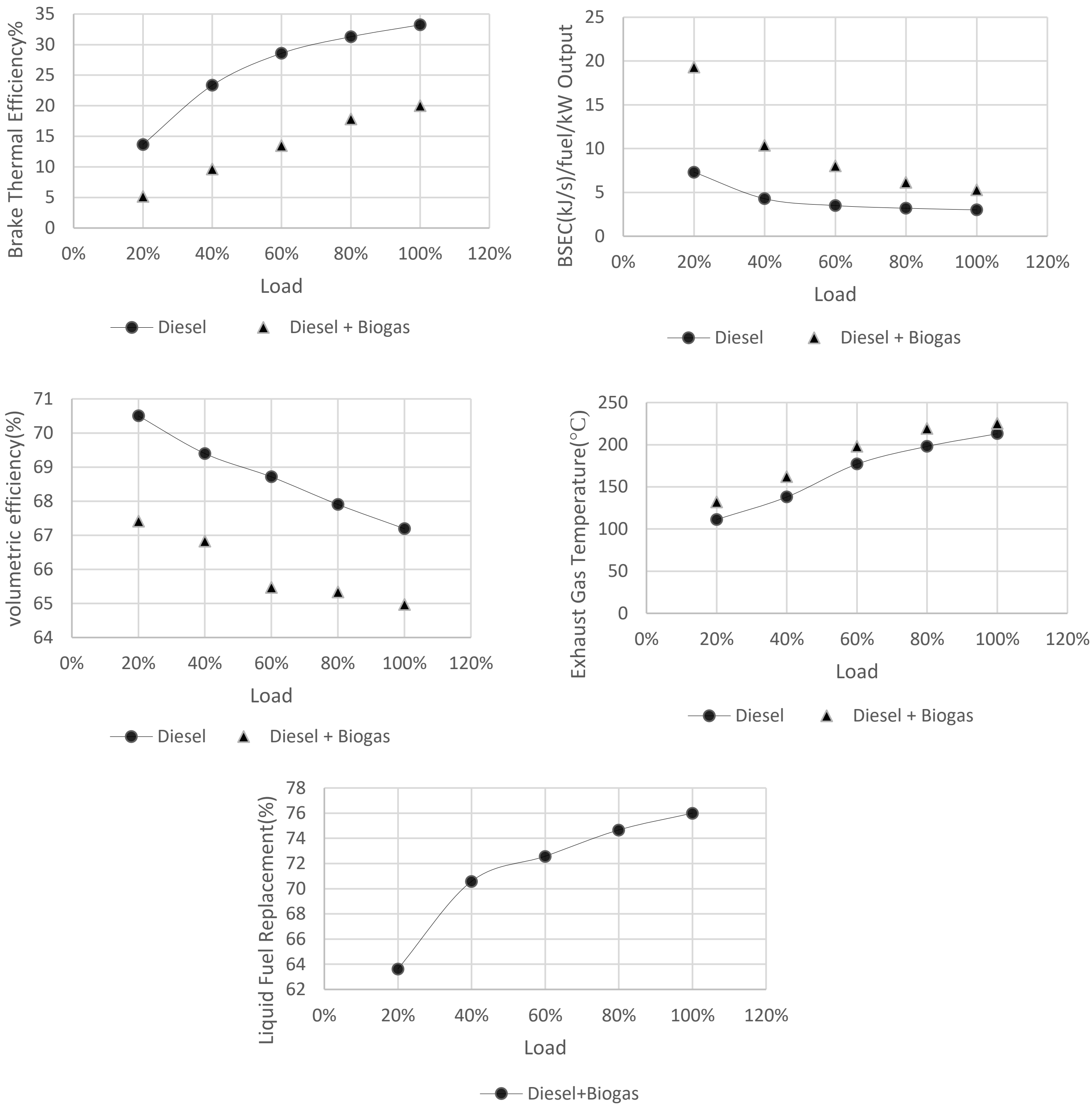

Fig. 4.1 Performance analysis for biogas run dual fuel diesel engine using diesel as pilot fuel

4.2.2 Emission Analysis

Emission analysis includes measurement of carbon dioxide (CO2), carbon monoxide (CO), hydrocarbon (HC) and oxides of nitrogen (NOX). Carbon dioxide is a product of fuel combustion during engine operation. Emission of carbon dioxide was found to increase linearly with load load for both diesel and DFM as observed in Fig. 4.2. When engine load was increased, the amount of fuel consumed by the engine went up so as to provide sufficient power to overcome the load. This resulted in increase in CO2 emission since it is a product of fuel oxidation. However, dual fuel mode was observed to produce higher CO2 emissions in contrast to diesel mode as Biogas constitute higher amount of CO2. At low loads, CO emissions are high which then decreases at medium load, and again increases at high load as indicated in Fig. 4.2. This is because initially, at low load, the cylinder temperature is low, and as a result the combustion of fuel is not proper. At high load, more amount of fuel needs to be supplied. Hence, after a particular load, the fuel air mixture becomes too rich to undergo complete combustion. However, the CO emissions are higher for DFM in comparison to diesel mode. This is due to the fact that biogas displaces air under DFM, and therefore, there is not enough oxygen is available for complete combustion. Unburned hydrocarbon (HC) is one of the regulated emissions, consisting of unburned fuel components and partially reacted components produced from the engine during operation. In this study, the lower flame velocity of biogas adds to the creation of HC more in dual fuel mode in contrast to diesel mode as observed in Fig. 4.2. Initially, at low load of 20%, the HC emission is high due to improper combustion as combustion chamber temperature is low in DFM but at load increases to 80% the HC emission gradually diminishes . With reduction of oxygen availability at 100% load, emission of more hydrocarbons was witnessed as the fuel air mixture become rich . The formation of NOX depends on temperature. NOX is formed in the high temperature combustion gases inside the cylinder mainly through oxidation of nitrogen present in the inducted air. In this investigation, the NOx emission increases with load as more amount of fuel needs to be supplied with increase of load which results in increase of temperature of combustion chamber. However, the NOX emission in the diesel mode is much higher in comparison to DFM as depicted in Fig. 6.2. This is because the temperature of the combustion chamber in case of diesel is much more as diesel has a high calorific value as compared to biogas. Moreover, the presence of inert CO2 in biogas results in lowering the temperature of combustion chamber.

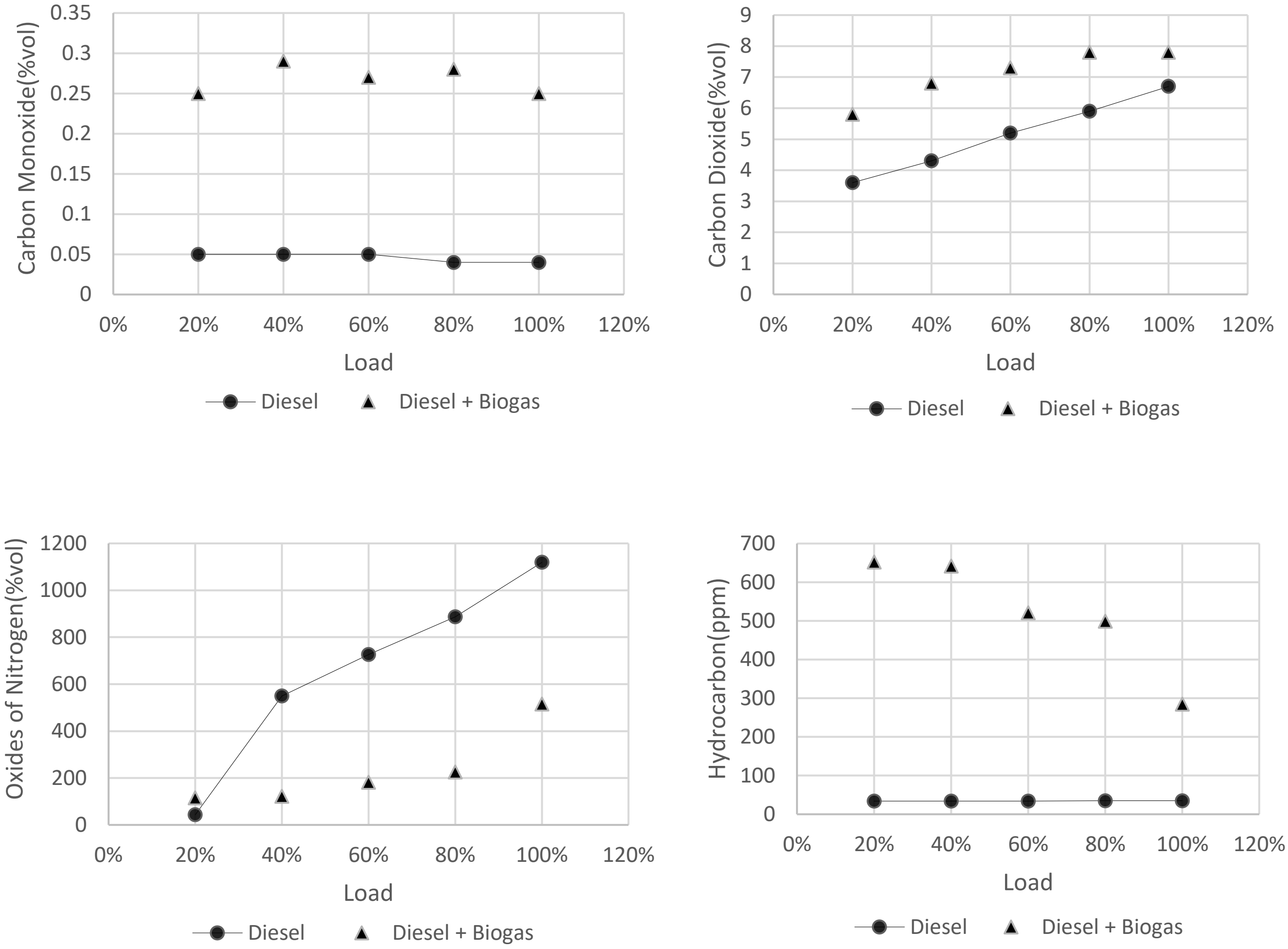

Fig. 6.2 Emission analysis for biogas run dual fuel diesel engine using diesel as pilot fuel

4.3 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. The variations of load are performed from no-load to full-load (12 kg) with an increment of 2.4 kg (20%). The diesel and dual fuel modes compared at standard diesel setting (i.e. Compression ratio =18 and Injection timing = 23^{o} BTDC). The performance analysis are performed based on theoretical equations given in Appendix.

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

<table>
<tr><td>Mode</td><td colspan="2">Fuel used</td><td>CR</td><td>Loading conditions (%)</td></tr>
<tr><td>Diesel</td><td colspan="2">SBD</td><td rowspan="4">18</td><td rowspan="2">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><td></td></tr>
<tr><td>Dual</td><td>DFM2</td><td>Pilot Fuel : SBD
Primary Fuel: Biogas</td><td></td></tr>
</table>

Table 4.4 Biodiesel Fuel Properties

Properties	SBD	RBB
Density(kg/m3)	872.8	876.4
Lower calorific value (MJ/kg)	38.5912	39.1924
Cetane number	56.3 (Giakoumis, 2013)	45-55 (Mittelbach,2004)

4.3.1 Performance Analysis

The performance analyses evaluated are brake thermal efficiency (BTE), brake specific fuel consumption (BSFC), volumetric efficiency (VE), exhaust gas temperature (EGT), liquid fuel replacement (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. The marginal higher calorific value of RBB than SBD results in obtaining a slightly more BTE.

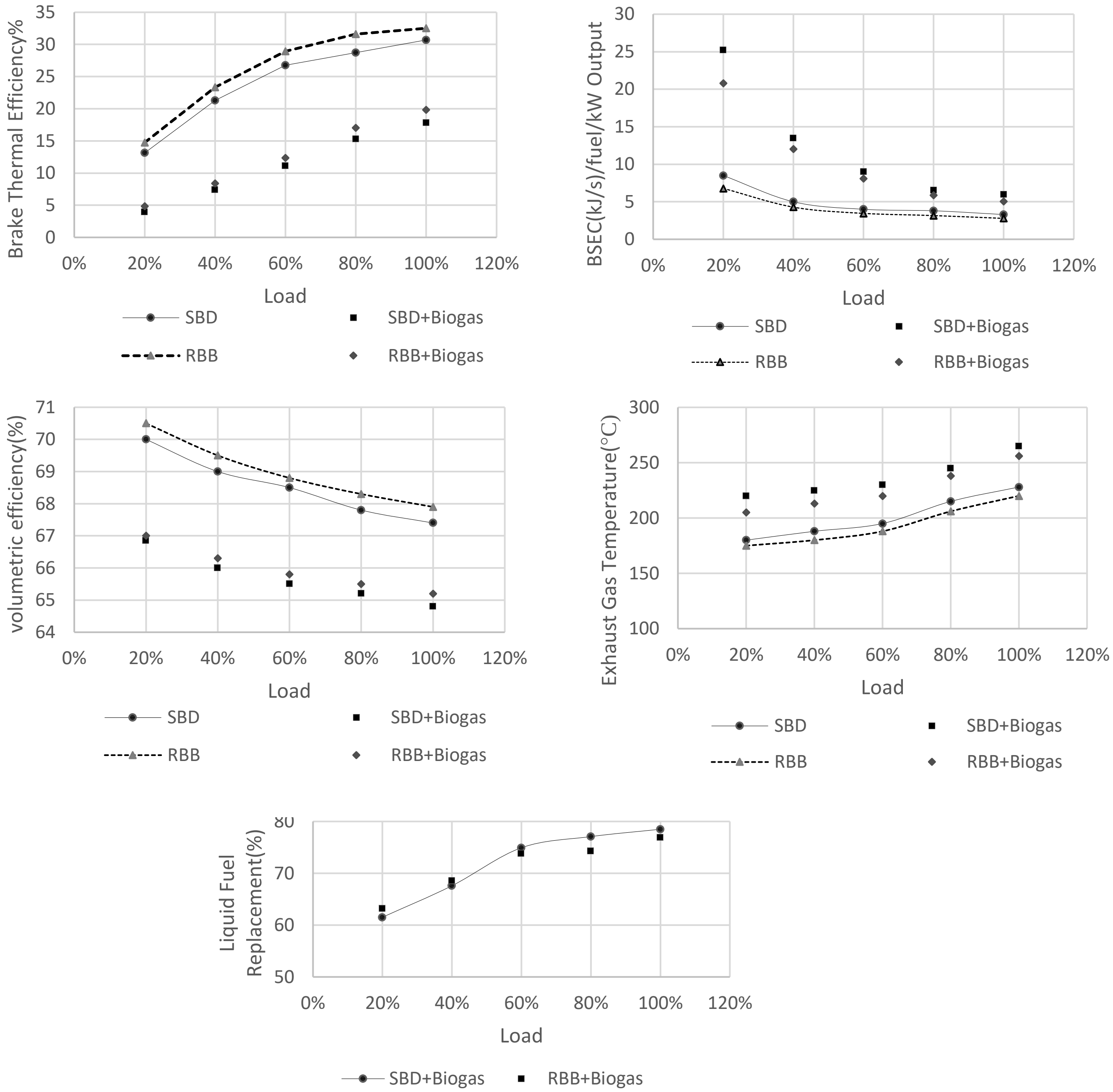

Fig. 4.3 Performance analysis for biogas run dual fuel diesel engine using SBD,RBB as pilot fuel

4.3.2 Emission Analysis

Emission analysis includes measurement of carbon dioxide (CO2), carbon monoxide (CO), hydrocarbon (HC) and oxides of nitrogen (NOX).The variations and their reasonings for considered parameters for emission analysis is similar to the above mentioned diesel-biogas run dual fuel engine.

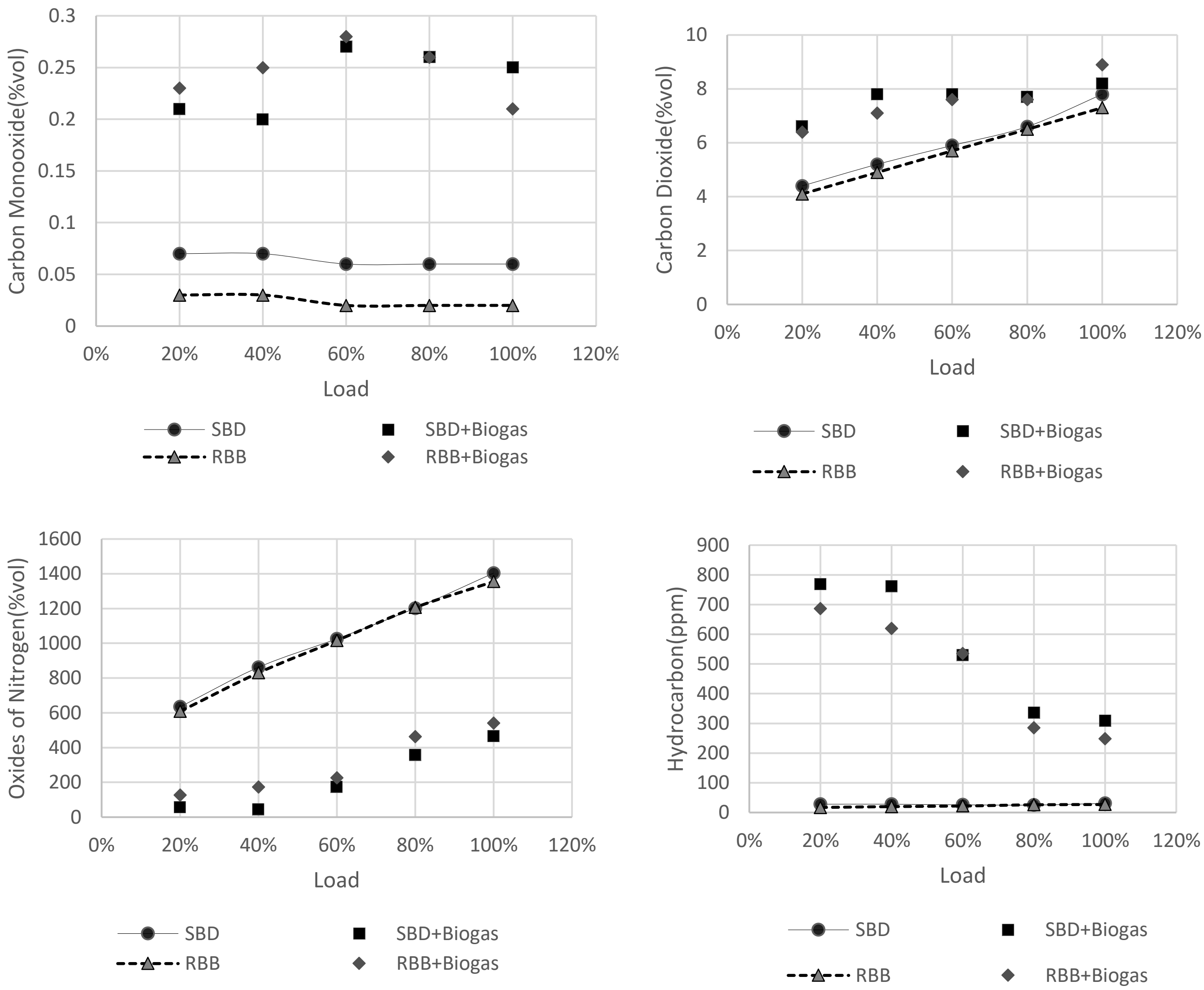

Fig. 4.4 Emission analysis for biogas run dual fuel diesel engine using SBD,RBB as pilot fuel

4.4 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. The variations of load are performed from no-load to full-load (12 kg) with an increment of 2.4 kg (20%). The diesel and dual fuel modes compared at standard diesel setting (i.e. Compression ratio =18 and Injection timing = 23° BTDC). The performance analysis are 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			20,40,60.80,100
Dual	DFM1	Pilot Fuel : SBDE Primary Fuel: Biogas	18	
Diesel	Pilot Fuel : RBBE			
Dual	DFM2	Pilot Fuel : SBDE Primary Fuel: Biogas		

Table 4.6 Fuel Properties of Ethanol

Properties	Ethanol
Density(kg/m3)	806
Lower calorific value (MJ/kg)	26.67*

4.4.1 Performance Analysis

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

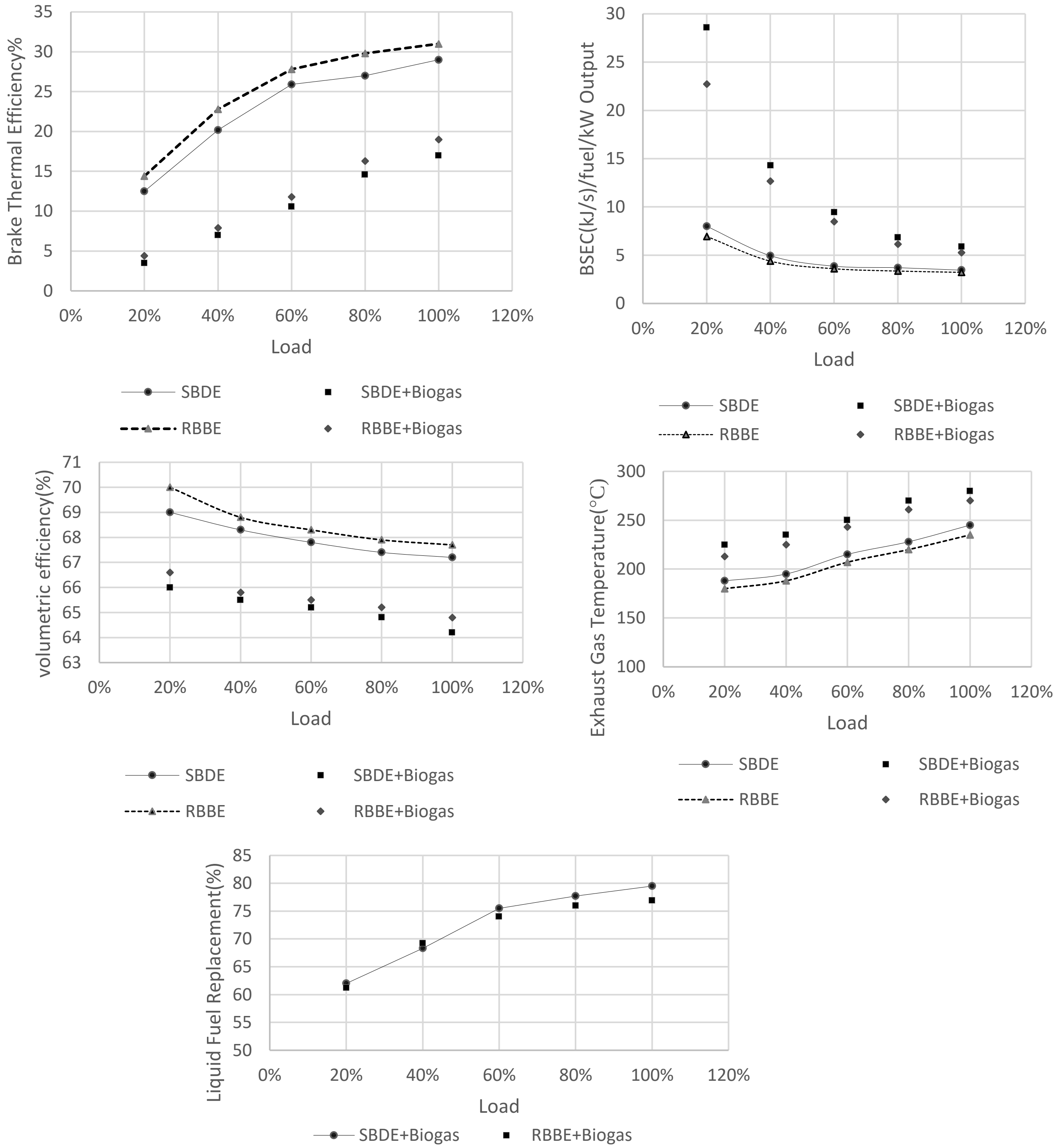

Fig. 4.5 Performance analysis for biogas run dual fuel diesel engine using SBDE,RBBE as pilot fuel

4.4.2 Emission Analysis

Emission analysis includes measurement of carbon dioxide (CO2), carbon monoxide (CO), hydrocarbon (HC) and oxides of nitrogen (NOX). The variations and their reasonings for considered parameters for emission analysis is similar to the above mentioned diesel-biogas run dual fuel engine.

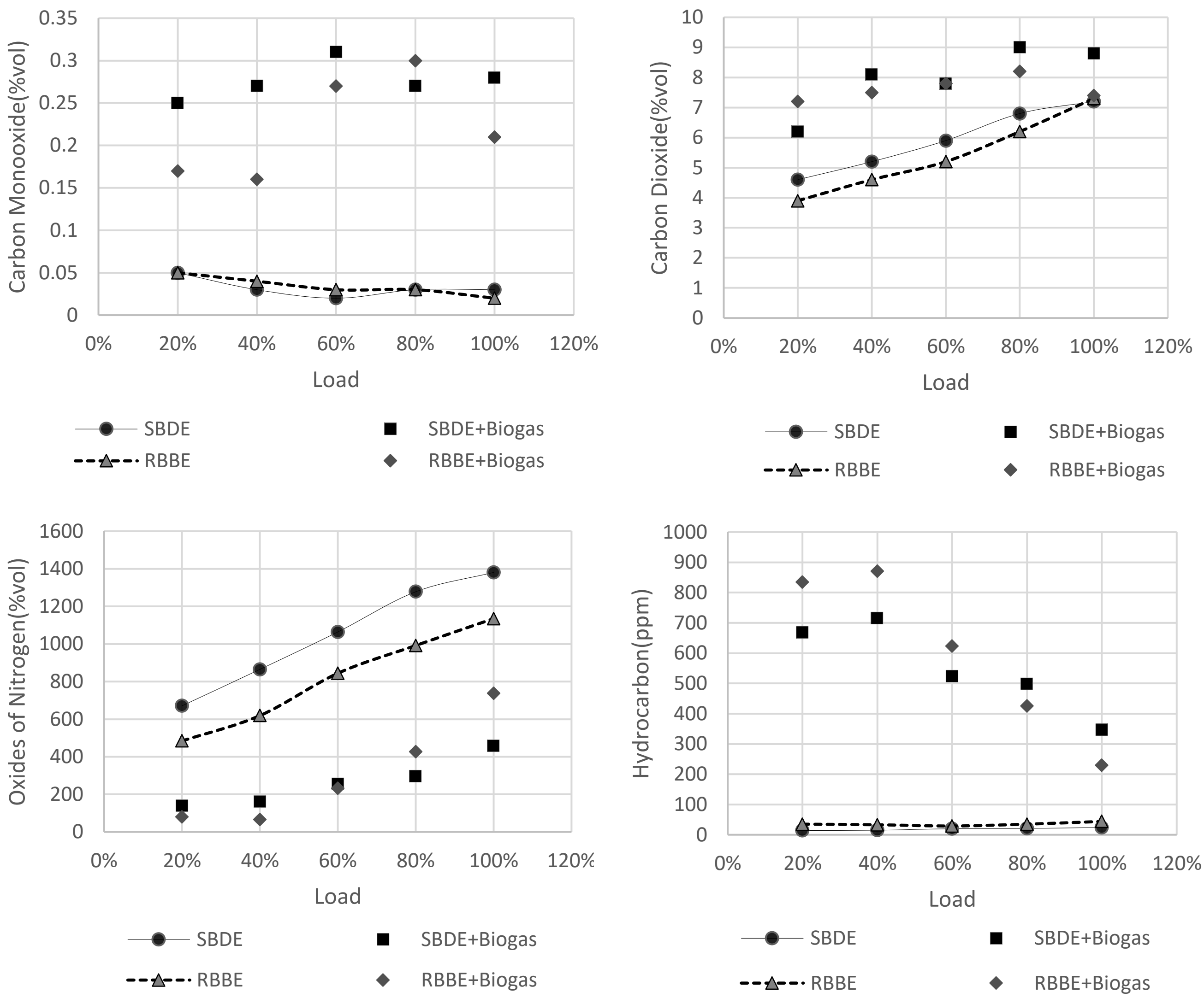

Fig. 4.6 Emission analysis for biogas run dual fuel diesel engine using SBDE,RBBE as pilot fuel

4.5 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. The variations of load are performed from no-load to full-load (12 kg) with an increment of 2.4 kg (20%). The diesel and dual fuel modes compared at standard diesel setting (i.e. Compression ratio =18 and Injection timing = 23º BTDC). The performance analysis are 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 : SBDO Primary Fuel: Biogas		

Table 4.8 Fuel Properties of Octanol

Properties	Ethanol
Density(kg/m3)	824
Lower calorific value (MJ/kg)	40.6

4.5.1 Performance Analysis

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

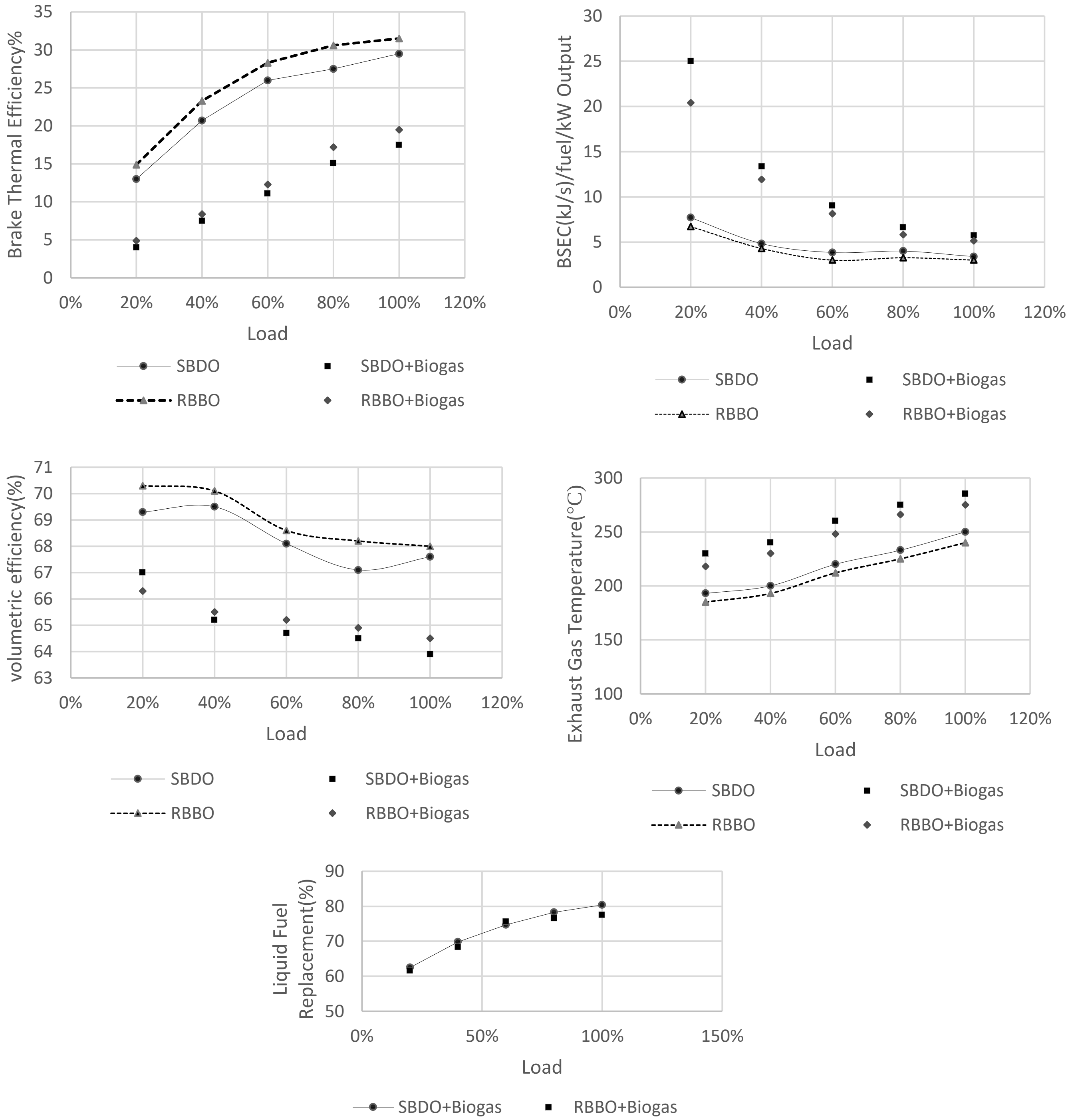

Fig. 4.7 Performance analysis for biogas run dual fuel diesel engine using SBDO,RBBO as pilot fuel

4.5.2 Emission Analysis

Emission analysis includes measurement of carbon dioxide (CO2), carbon monoxide (CO), hydrocarbon (HC) and oxides of nitrogen (NOX). The variations and their reasonings for considered parameters for emission analysis is similar to the above mentioned diesel-biogas run dual fuel engine.

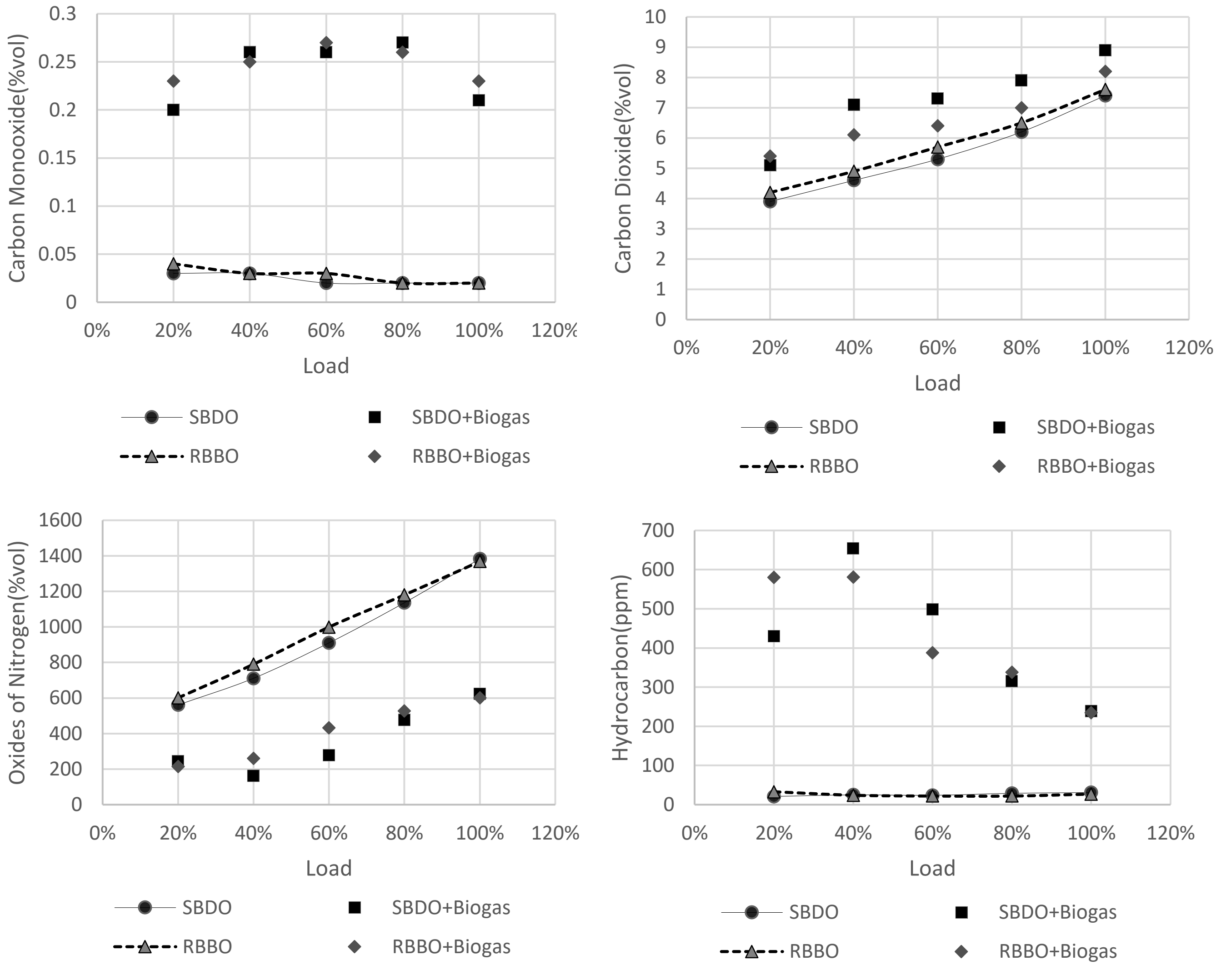

Fig. 4.8 Emission analysis for biogas run dual fuel diesel engine using SBDO,RBBO as pilot fuel

Chapter 5

CONCLUSION

The following conclusions have been drawn from the investigation: A direct injection compression ignition engine was modified to run on biogas, with diesel,biodiesel,alcohol as the pilot fuel. The modification of diesel engine into a dual fuel engine would involve introducing an air-fuel mixing chamber along the air intake system to provide an effective means of admitting the gaseous fuel into the combustion chamber and for homogenous mixing of air and the gaseous fuel. A modification has also to be made at the engine lever arm to allow adjustment of the injection quantities of the pilot fuel as may be necessary.

The maximum substitution of pilot fuel with biogas was possible up to 80% and engine operation on biogas alone was not possible. This was due to the high autoignition temperature of biogas which made it difficult to ignite without considerable amount of diesel. This led to the conclusion that utilization of biogas in DICI engine can help substitute use of diesel by up to 80% since it cannot fully replace it. This will help reduce dependence on petroleum fuels and save on finances compared to when diesel is relied on fully.

The brake thermal efficiency decreased with use of biogas due to the calorific value of biogas being lower compared to diesel. 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 increse. The Volumetric efficeiency is observed to decrease as load is increased for both diesel and dual fuel mode.But however the dual fuel mode has lower volumetric efficiency in contrast to diesel mode. The EGT of dual fuel mode is found to be higher than diesel mode for all cases. LFR is investigated to increase with the increase in load.

CO2 emission is found to be higher for all cases of dual fuel mode in comparison to pilot fuel mode,while at the same time it increase with increase in load for both the mode. As for HC and CO emission its on the higher side at low load, decrease upto 80% load and then increase again. Nox emission was found to be increasing with increase in load, however its on the higher side for pilot fuel mode in contrast to dual fuel mode. Biogas is a good fuel for dual fuel engines and can form a perfect supplement to the fossil fuels used in the country since it is renewable and affordable

Chapter 6

FUTURE SCOPE

Biogas is a suitable renewable fuel for compression ignition (CI) engines in the dual fuel mode. However, the thermal efficiency is low and hydrocarbon emission levels are high. One of the reasons for this is the presence of CO2 in biogas. It is therefore recommended that the CO2 present in biogas should be removed by upgrading the gas in order to enhance the flammability limits and improve flame speed; to improve thermal efficiency and lower emissions.

The other reason that leads to low thermal efficiency and increased emissions in dual fuel operation with biogas is the displacement of air needed for combustion, with the gaseous fuel. In order to help improve efficiency and reduce emissions, the engine should be modified to include a turbocharger which would enhance air intake and complete combustion of fuel.

The effect of injection pressure of pilot fuel on the performance and emission characteristic has not been reported in any open literature. The adjustment of the optimum injection pressure of the pilot fuel may enhance the efficiency of the biogas run dual fuel diesel engine.

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 andLHV_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 andLHV_{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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PUBLICATIONS

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