Investigation of the Performance and Emission Characteristics of Diesel Engine Fueled with Biogas-Diesel Dual Fuel

By

Melkamu Genet Leykun

A Thesis Submitted to the Department of Mechanical Systems and Vehicle Engineering

School of Mechanical, Chemical and Materials Engineering

Presented in Partial Fulfillment of the Requirements for the award of the Degree of Masters of Science in Automotive engineering

Office of Graduate Studies

Adama Science and Technology University

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Co-Advisor: Dr. Ramesh Babu Nallamothu

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Approval of Examiners

We, the undersigned, members of the Board of Examiners of the final open defense by Melkamu Genet Leykun have read and evaluated his thesis entitled "Investigation of the Performance and Emission Characteristics of Diesel Engine Fueled with Biogas-Diesel Dual Fuel'' and examined the candidate. This is therefore to certify that the thesis has been accepted in partial fulfillment of the requirement of the Degree of Master of Science in Automotive Engineering.

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DECLARATION

I hereby declare that this MSc thesis entitled "**Investigation of the Performance and Emission Characteristics of Diesel Engine Fueled with Biogas-Diesel Dual Fuel**" in partial fulfillment of the requirements for the award of the Degree of Masters of Science in Automotive engineering is my original work carried out from February 2020 to October 2020 under the supervision of Dr. Menelik Walle, Vehicle Engineering Department, Defense Engineering college and Dr. Ramesh Babu Nallamothu, Department of Mechanical System and Vehicle Engineering, Adama Science and Technology University, Ethiopia. It has not been presented for a degree in any other university, and all sources of material used for this thesis have been duly acknowledged.

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DEDICATION

I dedicate this research to my beloved mother Asmaru Fentie and all my brothers and sisters. Also, I am pleased to memorialize my research in loving memory of my father, Genet Leykun, who died in October 2002.

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First and for most, Glory to the almighty God, for protecting and giving courage to me to complete my study and in every step of my life.

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NOMENCLATURE

ASME	American Society of Mechanical Engineer
b	Cylinder bore diameter
BDC	Bottom Dead Center
BG	Biogas
BGES	Biogas Energy Share
BTE	Brake Thermal Efficiency
BSFC	Brake Specific Fuel Consumption
C	Speed of Sound
CA	Crank Angle
CFD	Computational Fluid Dynamics
CH_4	Methane
CI	Compression Ignition
CNG	Compressed Natural Gas
CO	Carbon monoxide
CO_2	Carbon dioxide
C_P	Specific Heat
CR	Compression Ratio
^0C	Degree Centigrade
D	Diesel
DFM	Dual Fuel Mode
DI	Direct Injection
Eq.	Equation
g	Gravity
GHGs	Greenhouse Gases
g/kWh	gram per kilowatt hour

g/mol	gram per molecule
H_2	Hydrogen
HC	Hydrocarbon
HCCI	Homogeneous Charge Compression Ignition
HRR	Heat Release Rate
H_2S	Hydrogen Sulfide
IC	Internal Combustion
ID	Ignition Delay
IP	Injection Pressure
IT	Injection Timing
K	Kelvin
kg/m^3	Kilogram per Cubic Meter
kJ	Kilo Joule
kPa	Kilo Pascal
kW	Kilo Watt
LHV	Lower Heating Value
LNG	Liquefied Natural Gas
LPG	Liquefied Petroleum Gas
L/min	Liter per minute
M	Mach number
m	Meter
mm	Millimeter
m/s	Meter per second
N	Engine Speed
N_2	Nitrogen
N.m	Newton meter

NOx	Nitrogen Oxide
O_2	Oxygen
P	Pressure
P_b	Brake Power
PM	Particulate Matter
ppm	Parts per million
R	Gas constant
r	Replacement ratio
RPM	Revolution per minute
s	Cylinder Stroke
T	Temperature
T_b	Brake Torque
TDC	Top Dead Center
UHC	Unburnt Hydrocarbon
V	Velocity
V_d	Displacement Volume
Vol	Volume
β	Beta ratio
ρ	Density
γ	Universal Gas Constant
Φ	Fuel-air Equivalence Ratio
Φ_{global}	Global Fuel-air Equivalence Ratio
%	Percentage

ABSTRACT

Diesel engines are the most reliable and efficient energy conversion devices with higher popularity and wider usage. This resulted in increased utilization of fossil fuels in the transport sectors. The resources of fossil fuels are depleting and their prices are increasing day by day, negatively affecting the economy of developing countries which are dependent on the importation of petroleum fuels. However, the emission from the burning of fossil fuels harming the global environment. These challenges led to search and utilize alternative fuels, which are attractive because of their environmentally friendly nature. The present study covers the utilization of a gaseous alternative fuel, raw biogas, in a diesel engine. Biogas alone cannot run a diesel engine, because gaseous fuel cannot burn by compression. It can be supplied to the CI engines in dual fuel mode by using an air-biogas mixer device. In this work, it is aimed to investigate the performance and emission characteristics of a biogas-diesel dual fuel mode diesel engine by employing a venturi gas mixer device for providing a homogeneous mixture. The experiment was conducted on a single-cylinder, four-stroke, air-cooled and naturally aspirated direct injection diesel engine (TBMC3-02) with a maximum power output of 2.2kW by varying engine load at a constant speed of 1500RPM. The diesel was injected by conventional injector setup and the biogas was inducted through the intake manifold with air at different flow rates of 2L/min, 4L/min and 6L/min. The performance and emission characteristics of the engine operated by dual-fuel mode were experimentally investigated, and compared to diesel. The results indicated that biogas inducted at a flow rate of 2L/min was found to have better performance and lower emission, than that of the other flow rates. On the other hand, dual-fuel mode with a biogas flow rate of 2L/min showed an average reduction in brake power, brake torque and BTE of 2.81%, 3.65% and 15.94% respectively and an average increment of 11.82% in BSFC as compared to diesel. Whereas an increment in CO, CO_2 and HC by 10.18%, 12.8% and 6.01 % respectively and an average reduction in O_2 and NOx emissions by 7.01% and 19.91% respectively as compared to diesel. The diesel replacement ratio varies from 19.56% to 7.61% at zero and 80% engine load with biogas energy share of 39.6% and 16.59% respectively.

***Keywords**: Alternative Fuel, Biogas, Diesel Engine, Dual-fuel, Venturi Gas Mixer*

CHAPTER ONE

INTRODUCTION

1.1 Background of the Study

The global energy demand is growing rapidly now a day due to the increasing population and by the continuous growth of industrialization. Fossil fuels supply more than 80% energy for global energy consumption and more than 95% energy for the transport sector globally (Singh et al., 2018). Global fossil fuel reserves are diminishing, rapid depletion, and increasing prices day by day. While greenhouse gas concentrations (GHGs) in the atmosphere are rising rapidly. Another important global challenge is the certainty of energy supply because most of the known conventional oil and gas reserves are concentrated in politically unstable regions (Turco et al., 2016). Additionally, combustion of fossil fuel hurts the ecology in the form of acid rains, global warming, etc. and the noxious pollutants comprising of nitrogen oxides (NOx) and particulate matter (PM) are discharged into the atmosphere by compression ignition (CI) engines rather than others (Neeranjan and Layek, 2018).

However, compression ignition (diesel) engines are the most reliable and efficient combustion device and have steadily grown with technological development over the past few decades (Yoon and Lee, 2011). It has increased its popularity due to the higher compression ratio (CR), better fuel economy, higher power, etc. Furthermore, its exhaust emissions are part of environmental pollutants, particularly oxides of nitrogen (NOx), particulate matter (PM), unburned hydrocarbon (HC), and carbon monoxide (CO), with potential health concerns (Karim and Sulaiman, 2018). But the positive achievements in terms of efficiency associated with diesel engines are, therefore, overshadowed by its high emission drawbacks. Due to the above reasons on the demand as well as the impact of fossil fuels, the identification and utilization of alternative fuels for internal combustion (IC) engines become a prominent research area. The utilization of alternative fuels especially for diesel engines has been essential due to increasing diesel engine emissions and rising energy demand.

An alternative fuel such as biodiesel, producer gas, compressed natural gas (CNG), liquefied natural gas (LNG), liquefied petroleum gas (LPG), biogas and ethanol, are those which releases less amount of emissions when used as a fuel in IC engines as compared to conventional fuels i.e. gasoline and diesel (Bhuimbar and Kumarappa, 2015). Alternative fuels like biogas and biofuels are found more attractive given their friendly environmental nature. From this, gaseous fuels like biogas release less emission which has a positive response from researchers and end-users (Salve et al., 2016). They have become a good alternative to petroleum-derived fuels (Ayade and Latey, 2016). Among all various gaseous fuels, Biogas is economical as well as suitable to the environment (Bhuimbar and Kumarappa, 2015).

Biogas is produced by the process of anaerobic digestion through the fermentation of biodegradable materials such as biomass, manure, green waste (plants), municipal waste, house waste, crops, and plant material (Prajapati et al., 2015), with a composition of Methane CH_4 (40–75%), Carbon dioxide (15–60%), Moisture H_2O 1–5%, Nitrogen N_2 (0–5%), Hydrogen H_2 (Traces), Hydrogen sulfide H_2S (0–5000 ppm), Oxygen O_2 (< 2%), Trace gases (– < 2%) and Ammonia (0–500 ppm) (Bharathiraja et al., 2018). It can be also upgraded and purified to natural gas standards and becomes bio-methane (CH_4) (Prajapati et al., 2015). It is not only an abundant resource of renewable energy for developing countries where meeting the growing demand for fossil fuels is a major economic challenge, but also for the developed countries where greenhouse gas emission reduction is a major issue (Vanga and Minhtu, 2019).

As recent studies have shown that, there is an emerging domestic energy source (i.e. biogas) in Ethiopia (Kamp and Forn, 2016) led by the Ethiopian National Biogas Program (NBPE). It may be produced in huge amounts from municipality abattoir wastes (Solomon et al., 2018) and correspondingly which improves environmental waste management. However, a large amount of solid waste disposal from a large number of livestock resources in Ethiopia increases methane emissions (Idriss and Mekonnen, 2018 and Yonas et al., 2019). One alternative way to minimize methane emissions is using solid waste disposal (which causes methane emission) to use energy sources i.e. using it for producing biogas. This promotes the utilization of biogas for automobiles, industries, stationary CI engines like generators for different organizations beyond that of household applications.

Application of biogas as a fuel in diesel engines for transportation, irrigation, and non-grid power generation purposes is preferred due to its high thermal efficiency (Vanga and Minhtu, 2019) and higher compression ratio of diesel engine (Yoon and Lee, 2011). The use of biogas in diesel engines is environmentally favorable. Thus, it can reduce pollutant levels in exhaust gases, including emissions of solid particles and NOx. It has high self-ignition temperature enabling them to operate with lean mixtures and high compression ratios result in higher efficiencies and lower emissions (Barik and Murugan, 2014).

However it cannot be used to run a compression ignition (CI) engine directly as diesel fuel due to its high self-ignition temperature (650°C) (Prajapati et al., 2015), and also cannot be used separately from diesel, a dual-fuel supply system when biogas-diesel used as fuel is required (Salve et al., 2016).

Biogas-diesel dual-fuel engine is a CI engine (with little modification on air intake system) and consists of two different fuels. Biogas as a primary fuel charge which the engine runs primarily and diesel as a pilot fuel used for ignition) (Prajapati et al., 2015), which can reduce emissions by compromising the engine performance. The performance of these energy integrations (dual fuel mode) depends on the homogeneity of the mixture, which is provided by a properly designed air-biogas mixer (Chandekar and Debnath, 2018). To utilize biogas in a dual fuel mode diesel engine, the air-biogas mixture should be in the homogeneous form, because poor premixing is caused by improper design of mixer, direct and excessive biogas substitution may badly affect engine combustion and emission characteristics (Mahmood et al., 2016). A properly designed air-biogas mixer for a biogas premixed charge dual fuel diesel engine can result in proper mixing of biogas with air for providing a homogeneous mixture to the combustion chamber and ensures better combustion and higher efficiency due to complete combustion of the dual-fuel mode diesel engine (Bora et al., 2013), even a lean mixture (Chandekar and Debnath, 2018).

Thus, this research investigated the performance and the emission characteristics of a diesel engine operated with biogas-diesel dual fuel at various load and biogas flow rate. The experimental work was carried out by varying the amount of biogas flow rate under different load conditions at constant engine speed with factory set fixed compression ratio (CR), injection pressure (IP) and ignition timing (IT). This work firstly concerned about design analysis and manufacturing of an appropriate venturi type mixer to provide a homogeneous air-biogas mixture to the engine cylinder. Then secondly performed an experimental work using a single-cylinder Auto/manual start computer-controlled test bench compression ignition (CI) engine (which is found in AASTU, Ethiopia) for examining the effect of biogas flow rate on the performance and emission characteristics by varying engine load. This work also compared the engine performance and emissions of a biogas–diesel dual fuel mode with 100% diesel fuel.

1.2 Statement of the Problem

Increasing in population and continuous growth of industrialization led to an increase in the utilization and demand of fossil fuel (diesel fuel) for diesel engines due to their most reliable and efficient combustion devices. However, fossil fuel prices are increasing as well its sources are depleting from day to day and thus create dependency on the petroleum oil reserver countries, geopolitical influence, and economical challenge for developing countries like Ethiopia. Moreover, in the transport sectors, it plays a very significant contribution to the production of greenhouse gases and global climate change. These challenges led to a search and utilize alternative fuels. Biogas is one of a gaseous alternative fuel and an abundant resource of renewable energy which is used as a fuel for internal combustion engines, especially for CI engines. But it cannot be supplied directly and run the diesel engine separately from diesel fuel due to its high self-ignition temperature. It requires a dual fuel supply system. Furthermore, a poorly premixed charge due to directly supplied and high raw biogas flow rate badly affects the performance and emission characteristics of the dual-fuel mode diesel engine. Hence the heterogeneous air-biogas mixture caused by directly supplied biogas to the engine cylinder was improved by employing a venturi gas mixer. While for having homogeneous mixture, venturi design parameter optimization should be required. Furthermore, using of biogas fuel is also an important for environmental waste management and reducing methane emissions especially for countries which have large number of livestock resources like Ethiopia. So, from this regards, this research recovers the utilization of biogas for diesel engines by modifying it to dual fuel supply system in addition to employee an appropriate venturi air-biogas mixer device at the air intake manifold.

1.3 Objectives

1.3.1 General Objective

The general objective of this thesis work is to investigate the performance and emission characteristics of diesel engine fueled with biogas-diesel dual fuel by employing an air-biogas mixer device at the intake manifold system.

1.3.2 Specific Objectives

The main objective of this thesis work is achieved through the following specific objectives:

- ✓ Designing and fabricating venturi gas mixer device
- ✓ Investigating the effect of load variation on engine performances and emissions for pure diesel and dual fuel mode
- ✓ Examining the effect of biogas flow rate on engine performance and emissions of a dual fuel mode diesel engine
- ✓ Comparing the performance and emission characteristics of dual fuel mode with baseline diesel fuel

1.4 Significance of the Study

This study shows how and why a gaseous alternative fuel (i.e. biogas) is used for a diesel engine, which benefits vehicle owners, society and the country as a whole. Usage of this renewable alternative fuel has the following basic significance;

- ✓ Reduce our dependency on petroleum-based fuels.
- ✓ It does not require considerable modifications of the air intake system, so it is easy to modify existing diesel engine into dual fuel mode for using biogas as a fuel.
- ✓ Improves environmental waste management (i.e. wastes released from universities, cities, municipal abattoir, etc. are used for biogas production).
- ✓ Reduces environmental emissions released from IC engines especially NOx.
- ✓ Reduces methane emissions released from cattle dug, which will serve as a source for biogas production.

1.5 Delimitation (Scope)

This thesis work includes design analysis and manufacturing of the venturi mixer. The experimental work focused on investigating the performance and emission characteristics of diesel-biogas dual fuel mode diesel engines by employing venturi air-biogas mixer at constant engine speed and various engine loads with different biogas flow rate also included. Comparison between the performance and emission characteristics of dual fuel mode with baseline pure diesel fuel using a gas mixer device is also included in this work.

1.6 Limitations

While conducting the research various problems have faced like lack of purified, bottled and specified biogas. There is also a lack of a biogas compressor. So, it was too difficult to bottle the biogas by using an ordinary compressor. Furthermore, the percentage concentration of hydrogen sulfide in the given biogas was not characterized due to the absence of an analyzer. Moreover, due to the lack of bottled biogas and thereby the flammability property of biogas, it was difficult to mass-transport from one place to another for experimentation. So, in the experimental test matrix, the biogas flow rate was limited within three flow rates. Finally, the testing engine was not well installed on the ground basement, thereby vibration occurred when it operated, and this affects the performance characteristics.

1.7 Organization of the Thesis

This thesis research work was organized into five chapters. **Chapter one:** introduces the background of the study, statement of the problem, objectives, significance, scope and limitations. **Chapter two:** discuss extensive literature reviews related to the application of biogas in a diesel engine; biogas-diesel dual fuel mode diesel engine combustion mechanism, its performance and emission characteristics, and also factors that affect the performance and emission characteristics of biogas-diesel dual fuel mode diesel engine. It also covers the properties of biogas as an alternative fuel for IC engines, biogas feedstocks, production, and composition. **Chapter three:** refers to the experimentation (engine setup and its different measurement and instrumentation devices), fuel supply system, engine conversion methodology, experimental design methodology, procedure along with uncertainty analysis. It includes the design of the venturi mixer, fluent analysis, and manufacturing process of the mixer and lastly, it includes experimental setup and test procedures. **Chapter four:** discuses about the results and discussion of the designed venturi mixer, performance and emission characteristics of dual fuel mode diesel engine under different biogas flow rates and various load conditions. Finally, concluding remarks with key findings of the present investigation with a recommendation and future research scopes have been summarized in **Chapter five**.

CHAPTER TWO

2. LITERATURE REVIEW

This chapter introduces about biogas, its feedstocks, production, composition and properties as an alternative fuel for internal combustion engines. Additionally, the application of biogas in diesel engines, biogas-diesel dual fuel mode diesel engines and its combustion mechanism are included. It also includes detailed reviews of different journals related to design parameters of a venturi gas mixer device, performance, combustion and emission characteristic of biogas-diesel dual fuel mode diesel engine.

2.1 Biogas as an Alternative Fuel for IC Engines

Biogas derived from organic waste materials and biomass is a promising alternative and renewable gaseous fuel for internal combustion (IC) engines and could substitute for conventional fossil fuels. It is produced by the anaerobic digestion or fermentation of biodegradable materials such as biomass, manure, sewage, municipal and green waste, plant material, and crops. The gases (in which biogas comprises) methane, hydrogen, and carbon monoxide (CO) can be combusted or oxidized with oxygen. This energy release allows biogas to be used as fuel (Sorathia et al., 2014). It is a renewable energy source and in many cases exerts a very small carbon footprint. This renewable energy is already used for heat and electricity production, but the best upgrading solution of this clean energy is using as an alternative vehicle fuel. Biogas is worth using rather than natural gas because of its renewable sources (Sai Kanth, 2015).

Due to the fossil fuel resources of oil, gas, and coal are limited nowadays; it acts as a promising alternative to fossil fuels for internal combustion engines resulting in the advantage of lower greenhouse gas emissions (Defaria et al., 2017) and also allows exhaust nitrogen oxides (NOx) emissions to be reduced substantially. Because of its properties, biogas is mostly used as a fuel for spark-ignition engines. This way of using biogas is most often applied in sewage treatment plants and rubbish dumps, where biogas form spontaneously as a result of chemical processes occurring there. These engines mostly power electricity generators or cogeneration units (Pictak et al., 2011).

The relatively high self-ignition temperature of methane (the main combustible component of biogas) of 640 °C, limits the use of this fuel as a power feed source for diesel engines. Nevertheless, because of diesel engines' higher efficiency and lower sensitivity to fuel quality, research is carried out on the use of biogas or methane as a fuel for these engines (Wierzbicki, 2012).

Generally, biogas is one of the important alternative renewable energy sources of the age as a fuel for internal combustion engines and it offers multiple benefits:

- ✓ It is environmentally friendly.
- ✓ It is produced from wastes materials like biomass, manure and municipal and hence ensures the maximum availability.
- ✓ It facilitates a better way of waste disposal along with a provision of energy supplies.
- ✓ It possesses the potential to replace a substantial amount of fossil fuels throughout the world and thus the emissions.

2.1.1 Feedstocks and Production of Biogas

Feedstocks for biogas production may be solid, slurries (partial liquid), and liquids. The range of potential waste feedstocks is much broader including municipal wastewater, residual sludge, food waste, food processing wastewater, dairy manure, poultry manure, aquaculture wastewater, seafood processing wastewater, grass wastes, and municipal solid wastes. Food processing wastewaters may come from dairy processing, vegetable canning, potato processing, breweries, and sugar production. There are many potential energy crops, which may be suitable for biogas production including sugarcane, as well as, woody crops (tree crops) (Wilkie, 2019). Biomass is abundant in the form of agricultural waste and crop residue which can be used as feedstock for anaerobic digestion. A significant amount of waste is also produced by animals as the activity of animal breed has gained increasing recognition. Generally, mostly used animal wastes include cow dung, pig waste, poultry manure, horse dung, camel dung, elephant dung, fishery waste, and slaughterhouse wastes. Studies have found that poultry wastes are rich in organic nitrogen and relatively lower-carbon sources (Nasir et al., 2012). Figure 2.1 below shows various feedstocks used in biogas production.

Manure

Biomass

Ethanol and Biodiesel By-products

Wastewater

Figure 2.1: Some feedstocks for biogas production (https://biogas.ifas.ufl.edu/feedstocks.asp).

Biogas is produced by extracting chemical energy from organic materials in a sealed container called a digester. The generation of biogas is the concept of Anaerobic Digestion (AD), also called biological gasification. It is a naturally occurring, microbial process that converts organic matter to methane (CH_4) and carbon dioxide (CO_2). The chemical reaction takes place in the presence of methanogenic bacteria with water an essential medium in the absence of oxygen to produce methane (Ray et al., 2013). For biogas production, different substrates such as organic waste from agricultural, industrial, or food sources and the biodegradable fraction of municipal wastes are used (Mao et al., 2015) as listed in its feedstocks above. Also, in several research institutions, worked on the use of other substrates, e.g., microalgae for biogas production has been conducted (Wieczorek et al., 2015). Figure 2.2 gives the diagrammatic representation for the production process of biogas in anaerobic digestion.

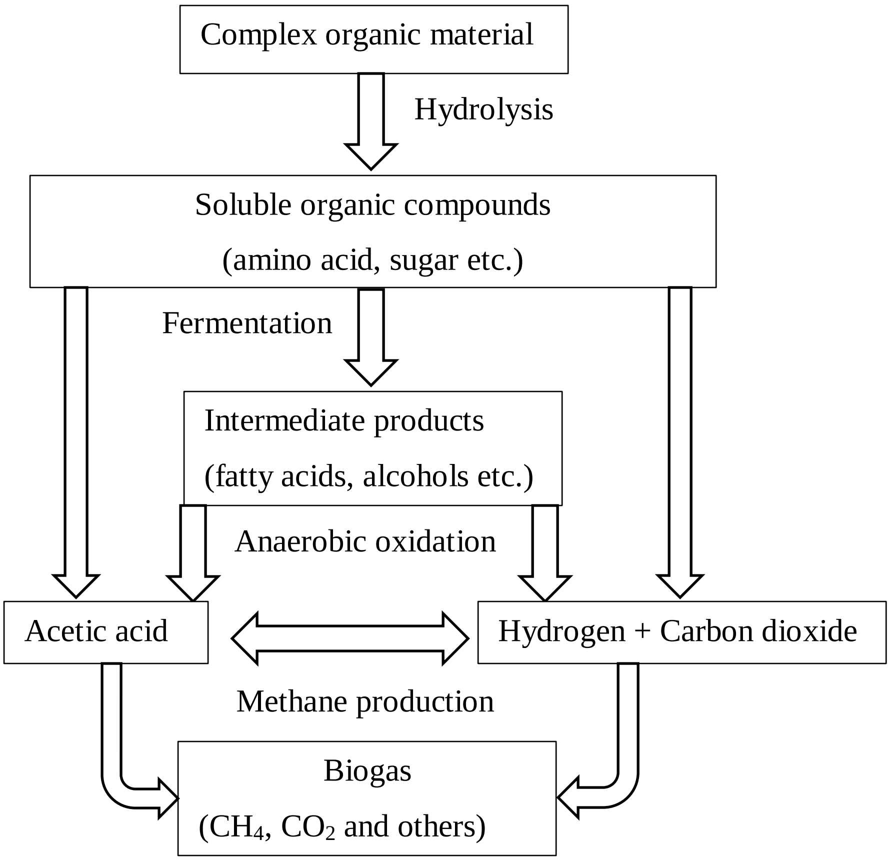


Figure 2.2: Production process of biogas in anaerobic digestion (Sai Kanth, 2015).

2.1.2 Components of Biogas

Since biogas is produced by micro-organisms from organic raw materials during the methane fermentation or anaerobic digestion process. Its main components are methane (CH_4) and carbon dioxide (CO_2) as well as small amounts of nitrogen (N_2), hydrogen (H_2), oxygen (O_2), hydrogen sulfide (H_2S), water vapor (H_2O), carbon monoxide (CO), hydrocarbons (HC), ammonia (NH_3), volatile organic compounds (VOC), and siloxanes (Ryckebosch et al., 2011).

The percentage of methane in biogas depends on the technology for obtaining biogas and ranges from 35% to 75%. Another combustible compound, hydrogen, although its percentage share is much smaller and typically amounts to 1-5%. The other biogas components are non-combustible compounds and constitute the ballast. The basic non-combustible biogas components are carbon dioxide and nitrogen. Besides the above mentioned compounds, biogas contains trace quantities of other chemical compounds whose percentage share is low. Table 2.1 shows the approximate compositions of raw biogas from different feedstocks.

Table 2.1: Composition of different feedstock's raw biogas (Wierzbicki, 2012).

Component	Composition		
	agricultural biogas	treatment plant biogas	landfill biogas
CH_4	45–75%	57–62%	37–67%
CO_2	25–55%	33–38%	24–40%
O_2	0.01–2.1%	0–0.5%	1–5%
N_2	0.01–5.0%	3.4–8.1%	10–25%
H_2S	10–30000 ppm	24–8000 ppm	15–427 ppm

Note: agricultural waste for biogas production includes animal excrement, biomass grown for energy purposes (e.g. grasses, maize, and sugar beet), municipal and organic waste (e.g. sewage treatment plants and rubbish dumps), and waste from the agricultural and food industries (plant and animal waste). Purifying raw biogas is recommended for having a high percentage of methane and thereby higher calorific value by removing impurities and separating CO_2 from it for ensuring better performance and lower emissions. Table 2.2 gives the general composition of purified biogas.

Table 2.2: Composition of purified biogas (Hogstrom et al., 2017).

CH_4	CO_2	NO_2	O_2	H_2S
≥97.3 %	≤1.6 %	≤0.7 %	≤0.4 %	0 ppm

2.1.3 Storage of biogas

Biogas can be stored at low, medium, and high pressures as shown in Table 2.3 below. The density of biogas is approximately 1.2 kg/m^3, which is approximate to air at ambient conditions. Hence, it requires a larger volume to store instead of in compressed form (Ray et al., 2013).

Table 2.3: Most commonly used biogas storage options (Ray et al., 2013).

Pressure	Storage device	Material
Low(13.8-41.4 KPa)	Water sealed gas holder	Steel
Low (42-104 KPa)	Gas bag	Rubber, Plastic, Vinyl
Medium(105-199 KPa)	Propane or Butane tanks	Steel
High (above 200 KPa)	Commercial gas Cylinders	Alloy steel

2.2 Properties of Biogas as a Fuel

Biogas is colorless, and before the removal of its hydrogen sulfide (H_2S), it smells like a bad egg. Its properties are described as follows and Table 2.4 summarizes its physical properties. Calorific Value: Methane is a high-quality gas fuel with a very high calorific value. In the standard condition, 1m^3 pure methane releases 35,822 kJ of heat through complete combustion, with the maximum temperature of 1400 ℃. As the biogas is a mixture with other ingredients, it has a lower calorific value of 20,000 to 29,000 kJ/ m^3, with a maximum temperature of 1200 ℃. The amount of CH_4 present in the biogas mainly defines the quality and energy content of the gas. As the methane (CH_4) concentration in biogas increases its calorific value also increases (Rosha et al., 2018). Specific Gravity: The specific gravity of methane is 0.55 and that of biogas is 0.94. In biogas tanks, the methane is floating in the upper space, while the heavier carbon dioxide stays at the bottom. Since biogas is lighter than air, it disperses quickly in the atmosphere, four times as rapid as the air. Solubility: The solubility of methane in water is very low: at 20 ℃ under one atmospheric pressure, only 3 unit volume of methane can be dissolved in 100 unit volume of water. That is why biogas can be generated underwater and be collected when water is let out. Critical Temperature and Pressure: The average critical temperature of standard biogas is -37 ℃ and the average critical pressure is 56.64×10^5 Pa (56.64 atmospheric pressure). This indicates that it can change its gaseous phase to the liquid phase when compressed up to the critical state and also explains why the liquefaction conditions of biogas are very demanding. Therefore biogas can only transmit through pipelines and cannot be traded as an energy commodity utilizing liquefaction and canning. Combustibility: Methane is a high-quality gas fuel. One unit volume of methane requires two units volume of oxygen to be completely combusted. As oxygen takes up about a fifth of the air and that 60% - 70% of the biogas is methane, one unit volume of biogas requires 6 to 7 unit volume of air for full combustion. These are an important reference for the development and application of biogas devices. Explosive Limit: Under normal pressure, the critical explosive mixture of standard biogas and air is 8.80% to 24.4%. Therefore, in a closure, a mixture of 1 unit biogas and 10 units air can produce a great pushing power from combustion. That is why biogas can be used as a power fuel besides cooking and lighting. An octane rating, or octane number, is a figure indicating the anti-knock properties of the fuel, based on a comparison with a mixture of iso-octane and heptane. The octane number of biogas is high. This high octane number of biogas has the potential to improve multi-cylinder partially premixed combustion (PPC). The increased ignition delay achieved for biogas with high resistance to auto-ignition will promote air-biogas premixing which suppresses smoke (Ali and Salih, 2013). Cold starting and cold weather issues: Biogas can be used for otto engines, gasoline and diesel engines. In otto engines and gasoline engines the cold starting characteristics are very good, there is no need to choke, but not in diesel engines. Hence biogas can be very good fuel for vehicles to drive a short distance in a cold climate. Corrosion: Hydrogen sulfide (H_2S) is particularly harmful when biogas is used in internal combustion engines. Its chemical reactions and those of its combustion product, sulfur dioxide, lead to corrosion and wear on engines. So for using biogas as a fuel for IC engine hydrogen sulfide should be removed by dry desulphurization.

Table 2.4: Physical properties of biogas (Khoshnevisan et al., 2017).

Properties	CH_4	CO_2
Molecular weight	16.04 g/mol	44.01 g/mol
Proportion	0.554	1.52
Boiling point	144^0C	60^0C
Freezing point	-165^0C	-39^0C

Technical parameters of biogas for engine performance are very important because of their effect on the combustion process in an engine (Ray et al., 2013). Those properties listed below and the thermodynamic properties of methane are summarized in Table 2.5 below.

- ✓ Ignitability of CH_4 in mixture with air is from 5 to15% Vol. within a temperature range of 918K to 1023 K. A compression ratio of an engine, 'CR' at which temperatures reach values high enough for self-ignition in mixture with air (CO_2 content increases possible compression ratio).
- ✓ The combustion velocity is a function of the volume percentage of the burnable component, here CH_4. The highest value of the combustion velocity of 0.38 is near

the stoichiometric air/fuel ratio, mostly at an excess air ratio of 0.8 to 0.9. It increases drastically at higher temperatures and pressures.

- ✓ Methane number, which is a standard value to specify fuel's tendency to knocking (uneven combustion and pressure development between TDC and BDC). Methane and biogas are very stable against knocking and therefore can be used in engines of higher compression ratios than petrol engines.
- ✓ Stoichiometric Air/fuel ratio on a mass basis at which the combustion of CH_4 with air is complete but without unutilized excess air.

Table 2.5: Thermodynamic properties of CH_4 (Ray et al., 2013).

Properties		Values
Specific heat(Cp)		2.165 kJ/kg.K
Density (ρ)		0.72 kg/m³
Individual gas constant (R)		0.518 kJ/kg.K
Lower calorific value	(Hu)	50000 kJ/kg
	(Hu, n)	36000 kJ/ m³n

2.3 Application of Biogas in a Diesel Engine

The utilization of biogas in diesel engines has some extra benefits over petrol engines. Diesel engines operate on the dual fuel mode with biogas fuel and they can be switched over to diesel-only mode as the biogas goes to empty. Additionally, the relatively high self-ignition temperature of methane (the main combustible component of biogas) around 650 °C limits the use of biogas as a fuel for diesel engines. Furthermore, because of diesel engines have high thermal efficiency, high compression ratio, and low sensitivity to fuel quality properties it allows the use of low energy fuels like biogas (Yoon and Lee, 2011; Wierzbicki, 2012; Prajapati et al., 2015; and Vanga and Minhtu, 2019). However, to ensure the proper operation of diesel engines using biogas, a dual-fuel system is required. Because biogas neither directly supplied nor separately provided for diesel engines (Salve et al., 2016). Not only this, the non-homogeneous air-biogas mixture badly affects the engine performance and emission characteristics (Mahmood et al., 2016), but also requires an appropriate mixer device for providing homogeneous mixture (Chandekar and Debnath, 2018).

2.4 Biogas-Diesel Dual Fuel Mode Diesel Engine

A diesel engine when converted to work under dual-fuel, i.e. gas fuel (primary fuel) is mixed with the air to form the charge primarily and a small quantity of diesel fuel (pilot fuel) is injected to ignite the charge (Karim, 2015), operation possesses certain advantages. A dual-fuel diesel engine enables the use of fuel gas (obtained from biomass) having lower calorific value, and lowers the running cost of the diesel engine. During the shortage of gas fuel, the engine can also be run with diesel fuel alone. Any contribution of gas fuel from 0 to 85 % can substitute a corresponding part of the diesel fuel without deteriorating the performance as compared to 100% diesel fuel operation (Bora et al., 2013). A dual-fuel engine can use a wide variety of primary and pilot fuels. The pilot fuel has a high cetane number. The performance of the engine depends on the amount of gaseous and the pilot fuel used. Measures like the addition of hydrogen, LPG, removal of CO_2, etc. have shown significant improvements in the performance of gaseous dual-fuel engines (Ray et al., 2013).

In dual fuel mode diesel engine operating with biogas-diesel fuel, biogas mixes with air by using a mixer device at the air intake system before entering the engine cylinder and then introduced to the combustion chamber. The mixture (air-biogas) does not auto-ignite due to the high self-ignition temperature of biogas. As a reason, at the end of compression stroke a small quantity of liquid fuel initiating gas fuel self-ignition is injected into the combustion chamber (Sahoo, 2011). Such an engine supply method is a solution that is relatively simple in design and allows simultaneous engine operation in both single (when gas fuel going to low) and dual-fuel systems (Wierzbicki, 2012).

2.5 Combustion Mechanism of Biogas-Diesel Dual Fuel Engines

The combustion of the dual-fuel (DF) engine starts similarly to a CI engine, but the propagation of a flame front occurs as similar to the SI engine (Ganesan, 2016). The combustion processes in CI engines running on pure diesel fuel can be divided into four stages as shown in Fig. 2.3. They are, A–B: period of ignition delay; B–C: premixed (rapid pressure rise) combustion; C–D: controlled (normal) combustion; and D–E: late combustion. Point ‘A’ is the start of fuel injection and ‘B’ for the start of combustion.

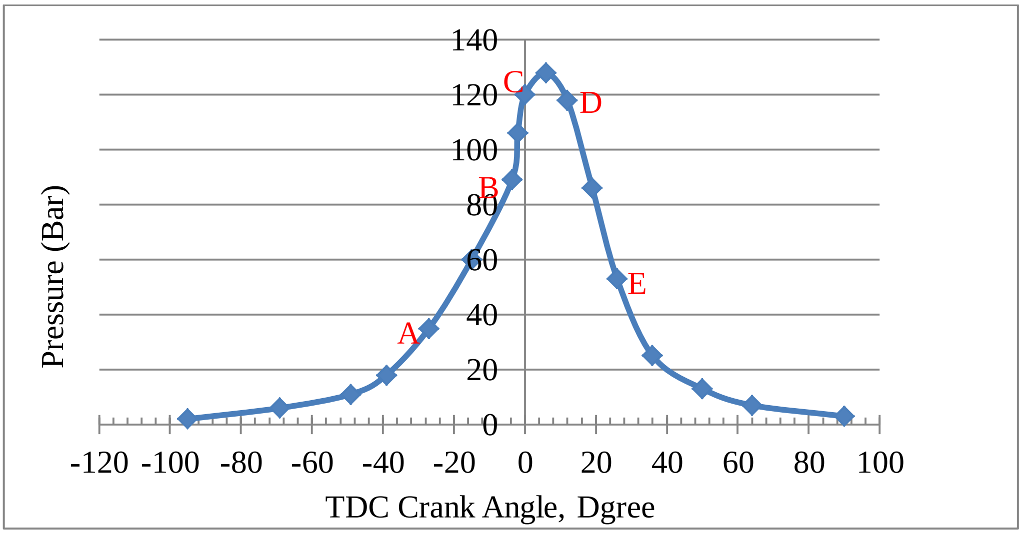


Figure 2.3: Combustion processes in a diesel engine (Sahoo et al., 2009).

However, the combustion processes in gas dual-fuel engines using pilot injection have been identified to take place in five stages as shown in Fig. 2.4. These are the pilot ignition delay (AB), pilot premixed combustion (BC), primary fuel ignition delay (CD), rapid combustion of primary fuel (DE), and the diffusion combustion stage (EF).

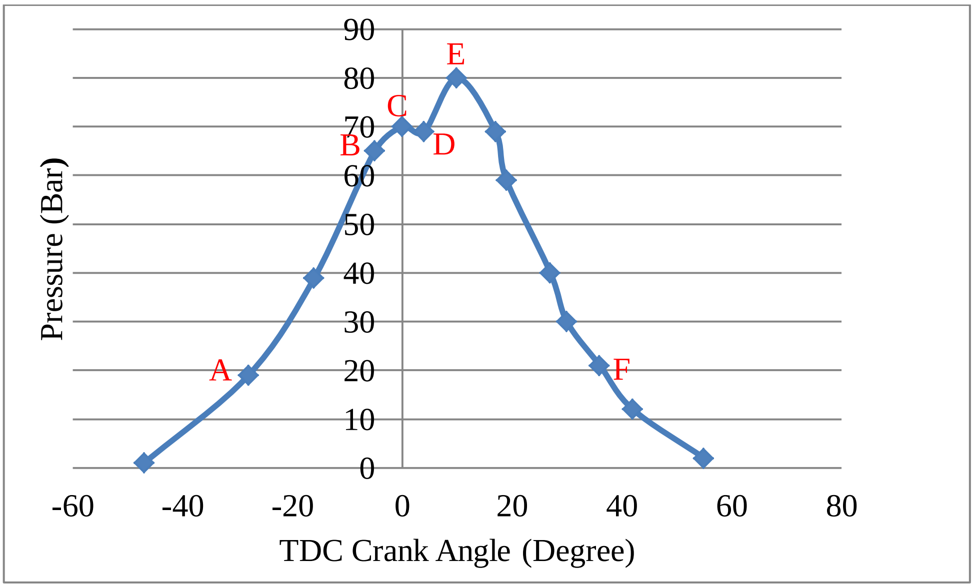


Figure 2.4: Dual-fuel pilot injection pressure–crank angle diagram (Das et al., 2020).

Ignition delay (AB) of injected pilot fuel exists longer than the pure diesel fuel operation. This is due to the reduction in oxygen concentration resulting from gaseous fuel substitution for air. The pressure rise (BC) is moderately low as compared to pure diesel fuel operation due to the ignition of a small quantity of pilot fuel. Again, there is a finite time lag between the development of the first and second pressure rises due to a longer ignition delay of a gas-air mixture, a result of the high self-ignition temperature. However, this ignition delay is short as compared with the initial delay period due to the pilot fuel injection. The pressure decreases slowly (CD) until the actual combustion of the fumigated gas starts. The phase of combustion (DE) is very unstable because it started with flame propagation that has been initiated by the spontaneous ignition of pilot fuel. The pressure rise here does not cause any operating problem since it occurs in an increasing cylinder volume. The diffusion combustion stage (EF) starts at the end of rapid pressure rise and continues well into the expansion stroke. This is due to the slower burning rate of gaseous fuel and the presence of diluents from the pilot fuel. Some gas-air mixture may escape combustion under this phase due to low oxygen concentration, valve overlap, flame quenching on the walls, or the effects of crevices (gaps). The success of this phase primarily depends on the length of the ignition delay. The heat release of diesel fuel operation is an apparent negative heat release before the main start of combustion. This is due to the cooling effect of the injected liquid fuel. This effect is not apparent with the dual-fuel combustion processes as pre-oxidation of the gaseous fuel starts before pilot fuel injection. The pilot fuel is then flattened the heat release rate within this region. The initial heat release due to pilot injection and combustion is continued well into the expansion stroke (Sahoo et al., 2009; and Das et al., 2020).

2.6 Performance and Emissions of Biogas-Diesel Dual Fuel Engine

The performance and emission characteristics of biogas-diesel dual fuel mode diesel engine may be affected by the quality of the mixture caused by the mixer device geometry and its design parameters like the diameter of the throat, the number of fuel inlet, angle of fuel inlet to the air inlet, angle of a convergent and divergent section of the mixer, etc., but also influenced by engine operating parameters like compression ratio, ignition timing, fuel-air equivalence ratio, mass flow rate of biogas and etc. Some existing related previous works are reviewed and highlighted below.

2.6.1 Air-Biogas Mixture Homogeneity

Mixture homogeneity is one of the major factors that affect combustion efficiency, performance, and emissions characteristics of the engine. The combustion process takes place inside the engine cylinder is directly related to the homogeneity of the mixture supplied through the intake manifold which affects the emissions (Dahake et al., 2016).

Therefore for having better performance and emission characteristics of biogas-diesel dual fuel mode diesel engines, the air-biogas mixture should be homogeneous.

Several studies showed that hydrocarbon (HC) emissions are produced due to incomplete combustion inside the combustion. The production of particulate matter (PM) is as a result of incomplete combustions of HCs due to heterogeneous air-fuel mixtures. Complete combustion in the internal combustion engine ensures to minimize the CO and NOx emissions. Thus, improving the combustion process and reducing emissions are directly related to the homogeneity of the air-fuel mixture induced to the engine combustion (Chintala and Subramanian, 2013). This could be achieved through employing an appropriate gas mixer device as briefly discussed in the next section below.

2.6.2 Gas Mixer Device for Biogas-Diesel Dual-Fuel Engine

A gas mixer mixes the air and fuel gas and plays a very important role in the performance of the dual-fuel operation diesel engine. It is seen that venturi type mixer is the most recommended design for gaseous fuels in the dual-fuel engines. Because the venturi type mixer meets most of the design requirements and operational phenomena. It provides optimal conditions for pressure drop, velocity, and homogeneous mixture (Chandekar and Debnath, 2018). It uses the venturi effect and provides a mixture of biogas and air to the engine cylinder in the required quantity for the efficient operation of the engine under all conditions.

When a fluid flows through the venturi throat, there is a change (decrease) in the static pressure, which creates a low-pressure zone in the throat. This increases the velocity of the flow according to the principle of mass conservation. The lower pressure at the throat drags gaseous fuel into the mixer in the proportionate quantity to the air. Besides this, the high velocity at the throat boosts the turbulence mixing of the biogas with air. The flow gets congregated at the convergent section and the same help biogas to distribute uniformly in the divergent section, covering its large volume in the downstream (Ramasamy et al., 2010 and Chandekar and Debnath, 2018).

Ramasamy et al. (2010) designed a venturi gas mixer for compressed natural gas and performed computational fluid dynamics analysis. The result showed that as the engine speed increases the pressure drop also increases at the venturi throat and the maximum engine speed of 6000rpm maximum pressure drop (i.e. 91000Pa) is noticed between 0.03 to 0.05m of a mixer within a total length of 0.08m.

Danardono et al. (2011) conducted a computational study to investigate the performance and mixing characteristic of a gas mixer and to develop an optimized mixer design. Based on the experiment and CFD results they conclude that the air-fuel ratio is mainly affected by total areas of the venturi throat, gas chamber cross-section, and gas exit. As the ratio of the throat area to the air inlet area is kept less than 0.6, the pressure loss of the venturi mixer was increased drastically and the minimum air-fuel ratio of a mixer was achieved when the ratio of the gas chamber area to the throat area is about 0.5. While increasing the gas exit area can produce a better mixing quality without significant pressure loss.

Bora et al. (2013) investigated and compared the flow behavior of the existing T-junction mixer with gas nozzles inclined at 45° and 90° with that of venturi type gas mixer designed for dual fuel diesel engines with the help of CFD based software executed based on pressure, turbulence intensity, velocity, the mass fraction of CH_4 at different cross-sectional planes. However, the T-junction gas mixer creates a large collision energy loss. Hence, a venturi gas mixer with converging angle, diverging angle, nozzle angle, and β are considered as 20°, 5°, 35°and 0.46 respectively provides a higher pressure drop at the throat region and ensures a better suction of biogas which ultimately boosts the mixing. They concluded that the venturi type gas mixer seems to provide for better mixing of biogas and air than the existing T-junction gas mixers.

Abo-Serie et al. (2015) studied computational analysis of multi-point injection methane-air venturi mixer for boiler burner application by using CFD analysis with standard venturi ASME MFC- 3M-1985 has been used with throat to pipe diameter ratio of 0.4. They optimized the values of injection holes diameter, intrusion length, and the number of holes by using CFD simulation and the Taguchi method, and was possible to achieve a uniformity index of 0.93 at a distance of one pipe diameter from the injection point. Finally, they have been concluded that: increasing the injection velocity of the fuel can lead to poor fuel distribution, increasing the intrusion length more than 0.05 times the diameter of holes can significantly lead to poor mixing and radial distribution of methane, increasing the number of holes around the venturi throat can improve the radial distribution but the effect is not that significant.

Palaniswamy et al. (2016) studied experimental work and CFD analysis of a mixer device for a supercharger CI engine by inducting biogas with air in the intake manifold by varying fuel inlet angle, fuel inlet location, and fuel injection pressure on the mixer and conclude that the supercharger enhances the performance and lowers the emission compared to natural aspiration where the gas injector is employed farther from the cylinder intake manifold and also it is observed from the simulation results that gas injector angle near to the cylinder intake manifold and 45º regarding axial axis of intake manifold are the optimum angle at maximum (i.e. 2 bar) biogas gas injection pressure and an engine speed of 750 rpm based on better mixture formation characteristics.

Arali et al. (2016) studied analytical and computational analysis of air-biogas mixer at 20^0 convergent angles with two biogas inlet for a SI engine and compared these results with standard performance curve at different engine speeds. The study has shown that the venturi mixer device is capable of increasing the engine performance, besides producing cleaner emission; the biogas operation also reduces operational cost on the vehicle.

Dahake et al. (2016) studied and optimized throat diameter, hole diameter, and the number of holes of the venturi type gas mixer for improving performance and emission of CNG engines by using five venturi configurations. The optimum designed venturi, which has a throat diameter of 23mm, hole diameter of 3mm with 8 number of holes, during actual test conditions at full load performance has found to be: lambda observed very close to 1 and equivalence ratio observed close to 0.88, desired for the smooth operation of CNG engines at its all operating points. Also, brake specific fuel consumption and exhaust emission found to be minimal compared to other ventures.

Shaikh and Shukla, (2016) performed optimization of natural gas mixture design by using a computational method for HCCI engine only by changing the convergence angle of 20^0, $30^{0,}$ and 40^0, which consequently changes throat diameter and divergence angle, by a varying number of holes 8, 10 and 12 at throat section for each design. The optimum design was obtained with a dimension of 30^0 convergence-divergence angle, inlet diameter of 35mm, throat diameter of 21 mm, 8 number of holes with 3 mm diameter for each, and outlet diameter of 30 mm within a total length of 60 mm. They concluded that increasing swirl intensity provides an improvement in BSFC and BTE but a reduction in an air-fuel ratio is noticed.

Romanczyk (2017) investigated and analyzed the influence of different gas inlet angles on venturi mixer characteristics and its performance by varying the gas inlet angle from 0^0 to 30^0 from its perpendicular junction and observed air-fuel ratio changes, pressure loss, as well as improvement of the mixing quality in the venturi mixer. He concluded that the greater the inclination of the gas inlet, the more fuel (methane) is sucked into the mixer. As a result, there would be a richer air-gas mixture at the outlet of the mixer.

Chandekar and Debnath (2018) performed a computational investigation of air-biogas mixing devices for different biogas substitutions and engine load variations. They observed that,

- ✓ As substitution of biogas increases (20% to 80%): at the throat; pressure drop (920 Pa) increases, velocities at the throat increases (from 33 m/s to 41 m/s), The turbulence kinetic energy is higher ($22m^2/s^2$) in the portions between the center and the periphery. There is a reduction in the difference between the maximum and the minimum concentrations of CH_4 across the venturi. During the initial process of mixing, as the substitution levels decrease, the high concentration region of CH_4 reduces quickly.
- ✓ As engine load increases (20% to 110%), pressure drop at the throat (620 Pa) increases, velocities at the throat increases (from 31 to 33 m/s), and turbulence kinetic energy increases (highest $19m^2/s^2$) and which provides a homogeneous mixture for CH_4 enriched biogas at different engine loads.

Carpintero Durango et al. (2019) studied air-biogas mixer device for diesel engine (dual fuel mode) application by comparing the design methodologies of Agudelo and Mitzlaff through computational fluid dynamics (CFD). They observed that in the Mitzlaff methodology there is a tendency to distribute the biogas in greater proportion over the divergence part of the mixer than that proposed by Agudelo. Likewise, the Mitzlaff design proposal tends to produce greater pressure drops in the throat section of the mixer, compared with that of Agudelo. Finally, it was found that the Agudelo-Mejia design approach tends to induce biogas in the throat of the device at pressures above 253Pa concerning the Mitzlaff design strategy and it was detected that the gradients of the surface velocity of the air in the case of Mitzlaff are greater magnitude concerning the radius of the throat than when compared with the methodology of Agudelo.

Generally as noticed from the literature above a venturi gas mixer is better from that of other mixers for gas-liquid dual-fuel engines in the case of:

- ✓ Mixing quality of gas fuel with air.
- ✓ Providing a moderate pressure drop at the throat region.
- ✓ High velocity at the throat boosts the turbulence mixing of the biogas with air.

- ✓ Ensuring a better suction of gas fuel.
- ✓ Due to a venturi effect, it provides a mixture of gas fuel and air to the engine cylinder in the required quantity for the efficient operation of the engine under all conditions.

However, for employing a venturi mixer in diesel-biogas dual-fuel engine several design parameters should be considered and which determines the mixing quality as observed from the literature above. Besides, using a venturi mixer, mixture homogeneity depends on its design parameters. Hence, homogeneous of the mixture is the major criteria for improving the performance and reduction in emission of the premixed biogas-diesel dual-engine, the current design is considered to optimize the venturi mixer only by varying the convergent angle and ratio of throat diameter to air inlet diameter of the mixer i.e. at a convergent angle of 20^0, 21^0 and 22^0 and ratio of throat diameter to air inlet diameter of 0.4, 0.5 and 0.6 which consequently changes divergent angle (from $7.5^0 \pm 0.5^0$ to 15^0), gas inlet area, etc. at a fixed entire length of the mixer because of the geometrical configuration requirement of the engine intake manifold.

2.6.3 Effect of Injection Timing and Compression Ratio

Bora et al. (2014) performed an experimental investigation for understanding the effect of CR on the performance, combustion, and emission characteristics of raw biogas run on a dual fuel diesel engine (DFDE). The experiments were conducted in a single-cylinder, direct injection (DI), water-cooled variable compression ratio (VCR) diesel engine. The CR was varied from 16 to 18 at standard injection timing (IT) 23^0 BTDC. At 100% load, the BTEs are lower than that of diesel-only mode at all CRs, the maximum pilot fuel saving was increasing with increasing CR. On the part of emission, on an average, there is a reduction in carbon monoxide as well as hydrocarbon emission as the compression ratio increases in dual fuel mode. However, there is an increase in NOx as well as CO_2 emission for the same change of CR. They concluded that a higher CR needs to be explored to arrive at an optimum performance of the biogas run a dual fuel diesel engine.

Bora and Saha (2014) studied an optimum injection timing of pilot fuel in a 3.5 kW dual fuel diesel engine run on biogas. The injection timings which were considered are 23^0, 26^0, $29^{0,}$ and 32^0 BTDC. The optimized injection timing of pilot fuel for this particular engine was found 29^0 BTDC which produced a maximum efficiency of 25.81% and a liquid fuel replacement of 75.77% at full load. The CO and HC emission were found to be least at 29^0

BTDC. They conclude that the advancement of injection timing of pilot fuel can be a potent tool for improving the performance of biogas run a dual fuel diesel engine.

Bora and Saha (2016) Experimented on a 3.5 kW single cylinder, direct injection, water-cooled, variable compression ratio biogas run dual fuel diesel engine at a combination comprising ITs of 26°, 29°, and 32° before top dead center and CRs of 18, 17.5, 17 and 16 at different loading conditions. Pilot fuel IT of 29° BTDC and CR of 18 were found to be an optimum condition. The results indicated that a combination of pilot fuel IT of 29° BTDC and CR of 18 produced a maximum brake thermal efficiency, low HC, and CO emissions whereas the NOx as well CO_2 emissions were found to be high. They concluded that the optimization of IT of pilot fuel along with the use of high CR is obligatory for effective practical utilization of raw biogas run dual fuel diesel engines.

Barik and Murugan (2016) investigated the combustion, performance, and emission characteristics of a direct injection (DI) raw biogas-diesel dual fuel diesel engine by varied injection timing from 23^0 to 27.5^0 CA BTDC. The results indicated that dual-fuel operation with the injection timing of 26^0 CA BTDC gave an overall better result. At full load, cylinder peak pressure, bsfc, CO, and HC emissions were higher, the smoke emission was lower than that of diesel. At part load, the engine performance in the dual fuel operation was improved with advanced injection timing, with a significant reduction in the exhaust emissions but NOx emission was higher.

Verma et al. (2019) performed an experimental investigation on diesel-biogas dual-fuel engines with increasing CR of 16.5, 17.5, 18.5, and 19.5 in a stepwise manner. It was revealed that the employment of higher CR improves pilot fuel substitution and energy efficiency. On the emissions side, an increase in CR showed a significant decrease in HC, CO, and smoke emissions, however, NOx emissions were increased with higher CRs.

They conclude that the higher CRs were not only advantageous to the engine performance but also the exhaust emissions.

Generally as seen from the literature reviewed above higher CR improves the pilot fuel substitution, energy efficiency, decrease NOx emissions in dual fuel mode diesel engine operation. Hence, higher CRs and advancement of injection timing of pilot fuel improves the performance of a biogas run in a dual-fuel diesel engine. Therefore advanced IT of pilot fuel along with higher CR is better for effective practical utilization of raw biogas in dual-fuel diesel engines.

2.6.4 Effect of Fuel-Air Equivalence ratio under Various Load and Speed

The fuel-air equivalence ratio in the dual-fuel engine may influence the performance and emission characteristics of the diesel engine under different load conditions.

Aklouche et al. (2017) conducted an experimental investigation to analyze dual-fuel (DF) engine operation using synthetic biogas fuel, composed of 60% methane and 40% carbon dioxide, under high load at a constant percentage energy substitution rate (60%) and compared biogas/diesel dual fuel mode performance, ignition delay, and other combustion characteristics to the conventional mode at various equivalence ratio (ϕ) (i.e. varied from 0.35 to 0.7). They conclude that; under DF mode, as ϕ increases, ignition delay becomes longer, heat release rate gives higher peak value, and BTE was improved and higher than that of conventional mode at $\phi = 0.7$. Whereas the emission concentrations HC, CO, CO_2, and NOx were reduced as ϕ was varied from 0.35 to 0.7.

Sarkar and Saha (2018) studied the effect of global fuel-air equivalence ratio (Φ_{global}) on the performance, combustion, and emission characteristics of a biogas-diesel dual fuel mode (DFM) diesel engine at different loads conditions. The Φ_{global} was varied from 0.30 to 0.89 from part to higher loads with a biogas flow rate of 0.67–3.99 kg/h. the results have shown that; at higher Φ_{global}, BTE reduces drastically, overall combustion behavior deteriorates, a higher cycle-to-cycle variation of cylinder peak pressure (CPP) was noticed, and reduction of NOx is observed with the increment of Φ_{global}. At part load with the increase of Φ_{global} from 0.30 to 0.87; drastically increase in ID, decrease in HRR, reduction of CO, an increase of both CO_2 and HC emissions was noticed.

Jagadish and Gumtapure (2019) conducted an experimental study in a dual fuel mode diesel engine using biogas, having a different percentage of methane (i.e. 20%, 30%, and 40% of CH_4 by mass basis) and diesel fuel at injection timing of 27.5° BTDC by varying the load for investigated performance, combustion and emission characteristics. They concluded that among the biogas mixtures, BG40 is optimized based on better performance and lesser emissions as compared to other biogas mixtures. The performance of the DFM diesel engine with BG40 at high loads (i.e. 75% and full load) was obtained; BTE is less than diesel and the net heat release rate higher than the diesel-only mode. Whereas the emissions; higher UBHC, higher CO emissions, lower NOx and lower soot emission than diesel-only mode were obtained.

Rahman and Ramesh (2019) conducted experimental studies on the effects of methane fraction and injection strategies in a biogas diesel common rail dual-fuel engine in two phases:

- ✓ Effect of composition of biogas on Start of injection timing, brake thermal efficiency, heat release rate, NO, CH_4, CO, and smoke emission. The results have shown that as biogas energy share (BGES) increases up to 60% and decrease in the CH_4 proportion (from 68% to 24%) in biogas the start of injection timing had to be advanced (3^0 CA) for best brake thermal efficiency (BTE) and reduces NO emission significantly. While decreasing the proportion of CH_4 in biogas did not affect the BTE and CO emission up to a BGES of about 60%. With an increase in BGES, there was an increase in heat release rate (HRR) and CO emission and a decrease in smoke.
- ✓ Effect of double-pulse injection of diesel in the biogas dual fuel mode: an early pilot injection reduces the smoke emission to a small extent at low to medium BGES without affecting other parameters. However, post-injection did not affect the dual fuel mode because of the post injected fuel undergoes diffusion combustion.

As noticed from the above for proper utilization of biogas in dual diesel engines the fuel-air equivalence ratio should be set at a higher value as seen from the literature. Within this ratio at medium load will improve the performance and emission characteristics.

2.6.5 Effect of Mass Flow Rate of Biogas

Mass flow rate of biogas during dual fuel mode diesel engine affects its performance and emission parameters. Some existing literatures are reviewed below to highlight its effect.

Sivalingam and Barik (2013) conducted an experimental work to investigate the performance and emission characteristics of a biogas fueled DI diesel engine at different flow rates of 0.15kg/h, 0.3kg/h, 0.45kg/h and 0.6kg/h. at full load with maximum mass flow rate of 0.6kg/h there is 30% reduction in brake thermal efficiency and volumetric efficiency and also higher BSEC in dual fuel mode. Higher BGES and 30% diesel fuel was saved. The dual fuel mode emission showed increment in the CO and HC emissions of 16% and 21% respectively but lower in NOx and smoke emissions of 35% and 41.3% respectively was observed. They conclude that, 0.6kg/h biogas mass flow rate is an optimum flow rate.

Barik and Murugan (2014) Investigated the combustion, performance and emission characteristics of direct injection diesel engine fueled with biogas-diesel in dual fuel mode at 0.3 kg/h, 0.6 kg/h, 0.9 kg/h and 1.2 kg/h mass flow rate of 73% methane biogas with load varaition of 0 to 100% in an increment of 25%. Biogas mass flow rate of 0.9 kg/h was found to be an optimum flow rate and therby give a better performance and lower emission, than that of the other flow rates at full load. Overall results was about 36% increase in the BSFC and 6.2% drop in BTE. The cylinder peak pressure in the dual fuel operation was found higher than diesel operation. Emissions was 17% and 30% increase in the CO and HC, 39%, 42% and 49% reduction in NO, CO2 and smoke emission with 0.215 kg/h diesel replacement at full load at 0.9 kg/h. They conclude that, the use of biogas with diesel in a CI engine seems to be desirable to reduce emissions, saving of conventional diesel and to utilize bio energy from biomass.

Satapathy et al. (2015) studied the combustion analysis of the diesel-biogas dual fuel direct injection diesel engine at 0.3 kg/h to 1.2 kg/h in steps of 0.3 kg/h 70% methane biogas flow rate with varation of load from zero to full load. The results showed that as mass flow rate of biogas increases its combustion characteristics also increases. But the optimum biogas flow rate was obtained at 0.9 kg/h flow rate and at this flow rate the combusstion characteristics results were found to be; higher rate of pressure rise and peak cylinder pressure of 2.2–3.9 bar/°CA and 6–11 bar, higher heat release rate of 3–10 J/°CA, longer ignition delay of 2–3 °CA and longer combustion duration of 1.6–4.7 °CA than that of diesel at full load.

Feroskhan et al. (2016) Investigated the effects of 75% methane biogas flow rate and cerium oxide addition on the performance of a dual fuel CI engine. They conducted an experimental work at biogas flow rate of 4L/min, 8L/min, 12L/min and 16L/min with cerium oxide addition for examining an optimum flow rate which yields better dual fuel engine performance. The result shown that as increase in biogas flow rate decreases BTE, volumetric efficiency, increases BSEC and itwas observed to contribute up to 80% of the total energy release. It was found to be; 4 L/min biogas flow rate provides the highest brake thermal efficiency than diesel operation.

Ambarita (2017) studied the engine characteristics of diesel-biogas dual-fuel compression ignition (CI) engine at a rated power of 4.41 kW with different speed and load conditions with diesel fuel as a baseline test. The result has shown that the output power and specific fuel consumption are higher than the baseline test data. Brake thermal efficiency in dual-fuel mode is higher at low biogas flow rate but lower at a higher biogas flow rate. So, it is affected by the biogas flow rate and methane concentration. An optimum biogas flow rate yields maximum brake thermal efficiency. As the biogas flow rate increases CO and HC content increases. The biogas can reduce diesel fuel consumption significantly from 15.3% to 87.5%.

Gnanamoorthi and Mohandoss (2018) worked experimental investigation on combustion, performance and emission analysis of dual fuel engine using tamarind seed and rice bran biogas at different flow rates such as 0.25 kg/hr, 0.50 kg/hr, 0.75 kg/hr and 1.0 kg/hr with load varation. The average biogas composition was 63% methane and 37% carbon dioxide. Results shown that; increase in mass flow rate of biogas decreases cylinder pressure, retards heat release rate, have longer duration of combustion, lower BTE, higher BSEC, lower exhaust gas temperature up to 80% load and higher above 80% load. On the other hand smoke and Nox emissions are lowered by 7.1% and 23.27%, respectively but the rest emissions were higher as compared to diesel mode.

Ambarita (2019) carried out an experimental study on diesel engine coupled with a catalytic converter run on dual-fuel mode using biogas produced from agricultural waste at a constant load (1500 kW) with different engine speed (1000 to 1500 rpm) and biogas flow rate (0 to 6 L/min). Experimental results showed that the output power is higher and does not affect by the catalytic converter and biogas flow rate. Thermal efficiency was better with a biogas flow rate of 2 L/min, but increasing the flow rate to 4 L/min lead to a decrease in it. As biogas flow rate increases, specific fuel consumption, HC number, and diesel replacement ratio (15.6% % to 74.8%) increases while decreasing CO emission.

Generally as noticed from the above literature in dual diesel engine higher biogas flow rate with more than 70% methane composition of biogas improves the combustion, performance and emission characteristics of a dual fuel engine than a diesel mode operation.

2.7 Summary of the Literature

Generally from the reviewed literature above a dual fuel supply system will be required to use biogas as a fuel for diesel engines. While the performance, combustion, and emission characteristics of a diesel-biogas dual fuel diesel engine mainly depend on two factors as seen from the literature above. These are the mixing device employed and engine operating parameters. As a venturi mixer provides a homogeneous charge into the engine cylinder, complete combustion will take place i.e. there is no chance for heterogeneous combustion. Therefore after go through from the existing literature the part including convergent angle and beta ratio associated with it of a venturi gas mixer is designed, for which differential pressure between the inlet and throat venturi sections is the least. This is part of an area understudied in the current research.

The performance and emission characteristic of a biogas-diesel dual diesel engine is not only influenced by the quality of mixture homogeneity, even if homogeneous air-biogas mixtures charge into the engine cylinder but also several engines operating parameters like load conditions, engine speed, biogas flow rate, compression ratio, injection timing and also equivalence ratio as seen in the literature. As noticed from the literature higher fuel-air equivalence ratio, higher CR, advance the injection timing of the pilot fuel with was obtained effective for practical utilization of biogas in dual fuel diesel engines. For most cases a biogas flow rate up to 0.9kg.h in which it contains higher methane composition above 70% improves its performance and emission characteristics.

From this regards, the effect of raw biogas flow rate on diesel-biogas dual fuel mode diesel engine at different engine loads with constant engine speed is the main area understudied.

2.8 Research Gaps

The facts cited in relevant published articles have been analyzed critically and the following salient features are found not being addressed properly.

- ✓ The literature reviewed shows that the study on diesel engine operated with diesel-biogas dual fuel mode using a gas mixer is very much limited. In most studies, the authors concentrate only on the effect of dual fuel on engine performance and emission parameters. The effect of gas mixer with different volume flow rates of biogas on engine performance parameters and emission characteristics of a conventional diesel engine for different loading conditions is not studied exhaustively.
- ✓ The use of numerical simulations in gas mixer flow parameter prediction is also limited. Many of the researches have used experimental analysis for the prediction of biogas efficiency for diesel engine characteristics to substitute diesel fuel partially.

CHAPTER THREE

3. MATERIALS AND METHODS

This chapter comprises the computational and experimental analyses to study the effects of mass flow rates of raw biogas on diesel engine performance and emission parameters under certain operating conditions of diesel engine operated with diesel-biogas dual fuel mode by employing a gas mixing device. This also includes venturi type gas mixture models that can be used to study the flow characteristics. The numerical analysis for designing a gas mixer was used for the computational methods that are described with the necessary illustrations. The simulation results were considered for the final fabrication of the suitable gas mixer in the house. Whereas the experimental investigation was carried out using a single-cylinder, direct-injection four-stroke diesel engine equipped with an eddy current dynamometer for loading the engine. The detailed research methodologies and required materials are explained below in the next sections.

3.1 Materials

This section describes the equipment's being utilized in this research along with detail of its characteristics and measurement capabilities. It also includes the experimental setup and procedure, device calibration and equations and measurements that were used to obtain the performance and emission data. In any experimental study, instrumentation plays an important role, as it provides the required data for analysis. Before starting an experimental investigation, it is essential to acquire the basic knowledge about the instruments required for experimentation and details about their working principles, range, accuracy, etc. In this section, different instruments used for the experimental investigation are described with the necessary illustrations. In this thesis work, the main component was fabricated (air -biogas mixing device), and employed in the modified experimental setup for experimentation. In this regard, the materials and equipment used for manufacturing the mixer device were aluminum shaft, drilling machine, drilling tool, lathe machine, etc., are listed in Table 3.1 and Table 3.2.

Table 3.1: Materials, tools, and machines used for mixer manufacturing

No.	Item	Quantity
1	Aluminum shaft	50mm diameter with 200mm length
2	Drilling tool	18mm, 8mm and 4mm diameter
3	cutting tool	2
4	Glass paper	4 piece
5	Lathe machine	1
6	Drilling machine	1

Table 3.2: List of materials and equipment used for experimental test

No.	Item	Quantity
1	Diesel fuel	10 liter
2	Biogas	4 compressed Tire tube at 1.5 bar
3	Flow control valve	2
4	Rotameter (flow meter)	1
5	Pressure gauge	1
7	Diesel engine	1
8	Exhaust gas analyzer	1
9	Dynamometer	1
10	Computer (desktop)	2
11	Tire Tube (kelemadary)	4

3.2 Methodology

To address the general and specific objectives of this work a well-designed methodology was being followed. There were combinations of procedures that were used to accomplish this study. These were design, fluent analysis and then manufacturing of the mixer device, preparing fuels (collecting biogas from digester outlets and purchasing diesel fuel), preparing the experimental setup of the test engine for experiment and analysis of experimental results. This thesis work was conducted based on the following procedures summarized as shown in Figure 3.1 below.

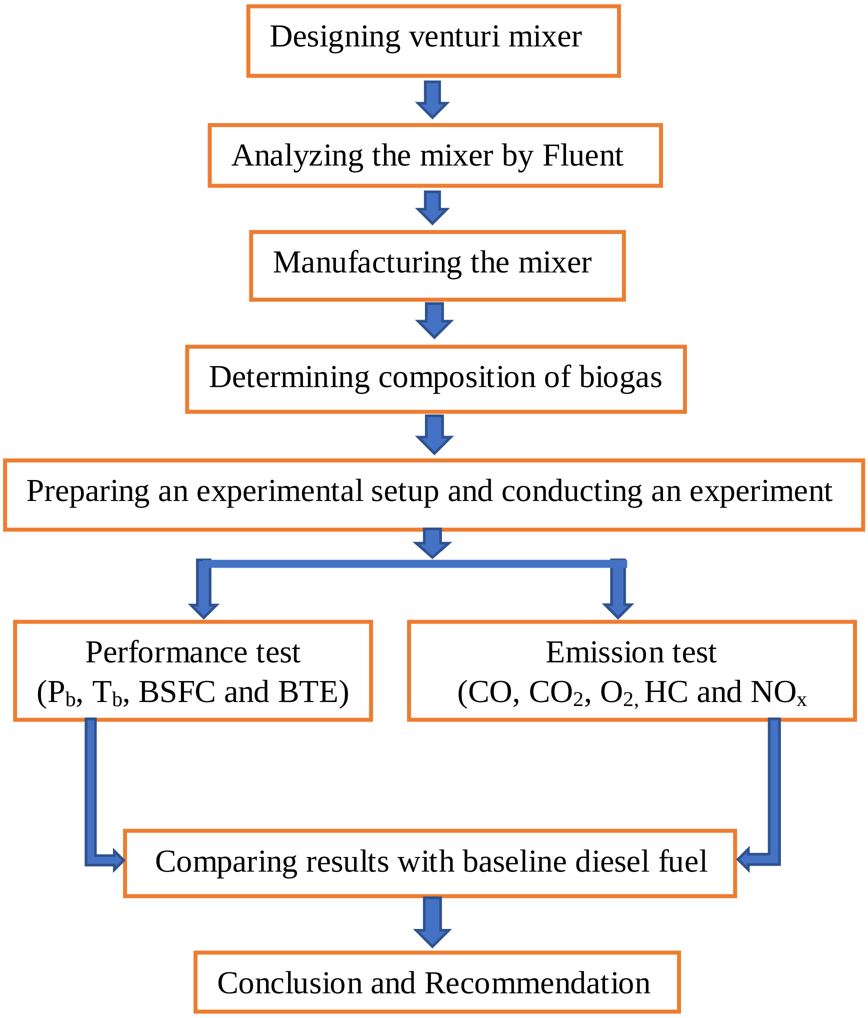


Figure 3.1: Block diagram of the procedure of this thesis work

3.3 Design of Biogas-Air Mixer Device

The biogas-air mixing device is designed according to the carburetion principle. It is a venturi having more than one biogas inlet bores at the throat circumference for mixing biogas with inlet air. Turbulent flow in the downstream of the venturi facilitates the mixing of biogas with the inlet air. The pressure drop at the venturi throat sucks biogas from the biogas cylinder (Sorathia et al., 2014). While designing the air-biogas mixer device, the parameters (which affects the proper mixing of the mixture thereby influence the performance and the emission characteristics of the engine) that should be considered are velocities at the throat and inlet of the mixer, mixer sizes (i.e. diameter at the throat of the mixer, angle of the converging and diverging section of the mixer) and biogas fuel inlet holes (i.e. number of holes, the diameter of holes and angle of fuel inlet holes to the air inlet). But for this design, the ratios of the throat to air inlet diameter and converging angles of the mixer are considered.

3.3.1 Design Calculation of the Venturi Mixer

The design calculations, which are considered to design the venturi mixer, are based on the computer-controlled Auto/manual start single-cylinder test bench engine TBMC3-02 technical specification as shown in Table 3.7, which is found in Addis Ababa Science and Technology University (AASTU), Addis Ababa, Ethiopia. For this calculation, the flow is considered as incompressible with velocity, pressure, and diameter at both the inlet and outlet of the mixer are assumed to be constant. The temperature and pressure of the air at the inlet of the mixer are taken as atmospheric conditions as listed below.

Assumption:

- ✓ The air is composed of only oxygen and nitrogen with 23% and 77% by volume respectively and its molecular mass is 28.97 g/mol
- ✓ Biogas is composed of 60% methane (CH_4) and 40% CO_2 only
- ✓ Ambient atmospheric conditions: temperature of 25^0C and pressure of 101325Pa
- ✓ The density of air and speed of sound at this temperature and pressure is 1.184 kg/m^3 and 343 m/s respectively.
- ✓ Universal gas constant (R) is 8.314 J/mol.K

Typical maximum values of volumetric efficiency (η_v) for naturally aspirated engines are in the range between 80% and 90%, and it is for diesel engines somewhat is higher than SI engines (Heywood, 1988). Whereas the volumetric efficiency for an engine at wide-open throttle is in the range between 75% and 90%, going down to lower values as the throttle is closed. Restricting airflow into the engine (closing the throttle) is the primary means of power control for a spark-ignition engine (Pulkrabek, 1996). For petrol engines, η_v is between 80% and 85 %, whereas for the diesel engine it is in the range 85% to 90% (Kumar, 2012). Therefore, it is assumed as 90% in the present work.

Then the required mass airflow rate of a four-stroke engine has to be determined by the equation (Muhssen et al., 2019):

$$\dot{m}_a = \dot{Q}_a \times \rho_a \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots. \ldots\ldots\ldots\ldots..(Eq. 3.1)$$

Where; $\dot{m}_a = \ mass\ flow\ rate\ of\ air\ in\ kg/s$

$\mathrm{Q_a} = \text{required air flow rate } (\mathrm{m^3/s})$

$\rho_a = \ density\ of\ air\ (kg/m^3)$

The required airflow rate to the engine cylinder is calculated by the following formula (Dahake et al. 2016):

$$Q_a = \frac{\eta_v \times V_d \times N}{2 \times 60} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (Eq. 3.2)$$

Where; $\eta_v =$ volumetric efficiency, assumed to be 90%

$V_d =$ Displacement volume (m^3)

N = Engine speed (rpm)

The displacement volume (V_d) for the testing engine is obtained by the formula (Feroskhan et al. 2016):

$$V_d = \frac{\pi \times b^2}{4} \times s \times N_c \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (Eq. 3.3)$$

Where; b = cylinder bore diameter

s = cylinder stroke

N_c = number of cylinder (i.e. equal to 1)

$$V_d = \frac{\pi \times (0.069\text{m})^2}{4} \times 0.06\text{m} \times 1 = 2.244 \times 10^{-4}\text{m}^3$$

Then, the required airflow rate enters into the testing engine cylinder at maximum power output in diesel-only mode is:

$$\dot{Q}_a = \frac{0.9 \times 2.244 \times 10^{-4}\text{m}^3 \times 3600\text{rpm}}{2 \times 60} = 0.00606\, {}^{\text{m}^3}/_{\text{s}}$$

Therefore, the mass airflow rate to the engine cylinder at maximum power output in diesel-only mode becomes:

$$\dot{m}_a = 0.00606\, m^3/s \times 1.184\,\text{kg/m}^3 = 0.0072\,\text{kg/s}$$

Since the inlet air velocity to the mixer is based on maximum engine speed, where maximum power output is obtained, at which the throttle is wide open (John K. Vennard, 1982).

Therefore the value of the volumetric airflow rate to the mixer inlet is equal to the product of intake air velocity and inlet area of the venturi. Thus it is given by (Kadirgama et al., 2008):

$$\dot{Q}_a = V_i \times A_i \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots \quad (Eq. 3.4)$$

Where; $V_i = \text{ inlet air velocity to the mixer}$

$A_i = \text{ inlet area of the mixer}$

To determine the inlet velocity, first, the inlet area (A_i) of the mixer should be calculated (Ramasamy et al., 2010).

$$A_i = \frac{\pi (D_i)^2}{4} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (Eq.\ 3.5)$$

Where; $D_i =$ inlet diameter of the mixer which is equal to the inlet manifold hose diameter of the indicated testing engine (i.e. 36 mm). Then the inlet area of the mixer is;

$$A_i = \frac{\pi (0.036m)^2}{4} = 0.00101787\ m^2$$

Therefore air velocity at the inlet of the mixer in diesel-only mode is:

$$V_i = \frac{\dot{Q}_a}{A_i} = \frac{0.00606\ m^3/s}{0.00101787\ m^2} = 5.95\ m/s$$

During designing a venturi gas mixer device, the venturi throat to its inlet pipe diameter ratio (beta, β, ratio) will be in the range of the American Society of Mechanical Engineers (ASME) standard which is from 0.25 to 0.75 (Grabel, 2001). For this design, the beta ratio is taken as 0.4, 0.5, and 0.6.

Hence the throat diameter is equal to the product of beta and air inlet diameter of the mixer. That is:

$$D_t = \beta \times D_i \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (Eq.\ 3.6)$$

Where; $D_t = \text{ throat diameter } (m)$

$\beta = \text{ throat to pipe diameter ratio (beta ratio)}$

Thus at $\beta = 0.4$;

$$D_t = 0.4 \times 0.036m = 0.0144m = 14.4mm$$

Then the throat area (A_t) becomes;

$$A_t = \frac{\pi(D_t)^2}{4} = A_i = \frac{\pi(0.0144m)^2}{4} = 0.000163m^2$$

From the continuity equation volume flow rate at the inlet is equal to the volume flow rate at the throat and also that of at the outlet of the venturi (John K. Vennard, 1982)

$$\dot{Q}_i = \dot{Q}_t = \dot{Q}_o = constant \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (Eq.\ 3.7)$$

Where; $Q_i =$ air flow rate at the inlet of the venturi (m^3/s)

$\dot{Q}_t =$ air flow rate at the throat of the venturi (m^3/s)

$\dot{Q}_o =$ air flow rate at the outlet of the venturi (m^3/s)

Thus;

$$\dot{Q}_t = V_t \times A_t \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (Eq.\ 3.8)$$

Where; $V_t =$ $velocity\ of\ air\ at\ the\ venturi\ throat\ in\ m/s$

Therefore the throat velocity becomes;

$$V_t = \frac{\dot{Q}_i}{A_t} = \frac{0.00606\, m^3/s}{0.000163m^2} = 37.2\, m/s$$

The throat of the venturi causes to raise the air velocity as a linear function of the change in cross-sectional area and the Mach number (M) at the point where the flow velocity is maximum (at the throat) should be less than 0.3 for considering a flow is incompressible flow (Ramasamy et al., 2010).

The Mach number at the throat of the venturi is calculated by (Muhssen et al., 2019):

$$\mathrm{V} = \mathrm{MC} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (Eq.\ 3.9)$$

Where; $V =$ $maximum\ velocity\ obserbed\ through\ the\ venturi$

M = Mach number,

C = sound speed (m/s)

$$\mathrm{M} = \frac{V}{C} = \frac{37.2\, m/s}{343\, m/s} = 0.108$$

Therefore the flow is considered as an incompressible flow. The flows of gases at low Mach number, the density of a fluid can be considered to be constant, regardless of pressure variations in the flow. Therefore, the fluid can be considered to be incompressible and these flows are called incompressible flows. A common form of Bernoulli's equation, valid at any arbitrary point along a streamline is (Dahake et al. 2016):

$$\frac{P_1}{\rho} + gZ_1 + {V_1}^2/2 = \frac{P_2}{\rho} + gZ_2 + {V_2}^2/2 \ldots\ldots\ldots\ldots\ldots\ldots..(Eq.\ 3.10)$$

Where; $\mathrm{P_1, P_2} =$ pressures at the inlet and throat respectively $(\mathrm{N/m^2})$

$\mathrm{z_1, z_2} =$ the elevations of the point above a reference plane (m)

$\mathrm{V_1, V_2} =$ velocities at the inlet and throat respectively (m/s)

$\rho =$ density of air $(\mathrm{kg/m^3})$

$\mathrm{g} =$ acceleration due to gravity in $\mathrm{m/s^2}$

But $Z_1 = Z_2$. Then the vacuum created at the throat is:

$$\Delta P = \frac{{V_t}^2 - {V_i}^2}{2} \times \rho_a = 798.275\ Pa$$

The parameter of a venturi in diesel fuel mode is summarized in Table 3.3 below.

Table 3.3: Venturi dimension parameters in diesel fuel mode

Air mass flow rate ($\dot{m}_a$) (kg/h)	Velocity of inlet air (V_a) m/s	Venturi throat velocity (V_t) m/s	Vacuum at the throat (ΔP) N/m^2
25.92	5.95	37.2	798.3

In dual fuel mode

The biogas flow control valve is to be opened after the engine startup in diesel fuel mode. As the biogas flow control valve starts to open then the vacuum created at the throat in diesel-only mode sucks biogas from its source to the mixer. Hence the amount of airflow rate to the engine cylinder decreases and substituted by biogas. Frequently the engine starts to operate in dual fuel mode. Thus the total mass flow rate of air in diesel fuel mode is equal to the sum of the mass flow rate of air and biogas in dual fuel mode due to the fixed volume of the engine cylinder.

$$(\dot{m}_a)_{diesel\ mode} = (\dot{m}_a + \dot{m}_{bg})_{dual\ fuel} \ldots\ldots\ldots\ldots\ldots\ldots. (Eq.\ 3.11)$$

Where; $(\dot{m}_{bg})_{dual\ fuel} =$ mass flow rate of biogas in dual fuel mode (kg/s)

Therefore;

$$(\dot{m}_a + \dot{m}_{bg})_{dual\ fuel} = (\dot{Q}_a\rho_a + \dot{Q}_{bg}\rho_{bg})_{dual\ fuel} = 0.0072\ kg/s$$

Where; $\rho_{bg} = density\ of\ biogas (kg/m^3)$

$\rho_a = 1.184\ kg/m^3$

The density of biogas is also calculated by the following formula (Vitazek et al., 2016);

$$\rho_{bg} = \frac{P}{\gamma \times T} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots. (Eq.\ 3.12)$$

Where; P = ambient pressure (Pa)

T = ambient temperature (K)

$\gamma = universal\ biogas\ gas\ constant\ (kJ/kg.K)$

The universal biogas gas constant is calculated (Vitazek et al., 2016) as:

$$\gamma = \frac{R}{M_{bg}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots. (Eq.\ 3.13)$$

Where; $R = universal\ gas\ constant\ (8.314 J/mol.K)$

$M_{bg} = molecular\ mass\ of\ biogas\ (g/mol)$

Therefore the density of biogas (ρ_{bg}) is equal to $1.12\ kg/m^3$.

Thus the volumetric airflow rate becomes;

$$\dot{Q}_a = 0.00608 - 0.94\dot{Q}_{bg} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (Eq.\ 3.14)$$

Since biogas contains more components but only methane is combusted. So for stoichiometric combustion of methane, one mole of methane is completely burnt by two moles of air as shown below.

$$CH_4 + 2(O_2 + 3.76\ N_2) \Rightarrow CO_2 + 2H_2O + 7.52N_2$$

From the above reaction, the stoichiometric air-fuel ratio is 17.2. Thus it becomes (Kadirgama et al., 2008):

$$\frac{\dot{m}_a}{\dot{m}_f} = \frac{\dot{Q}_a \times \rho_a}{\dot{Q}_{CH4} \times \rho_{CH4}} = 17.2 \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (Eq.\ 3.15)$$

Where; $\dot{m}_\mathrm{f} =$ mass flow rate $of\ methane$ (kg/s)

$\dot{Q}_\mathrm{CH4} =$ volume flow rate $of\ methane$ $(\mathrm{m}^3/\mathrm{s})$

$\rho_{CH4} =\ density\ of\ methane\ (= 0.656\, kg/m^3)$

Where the density of methane and universal methane gas constant is;

$$\rho_{CH4} = \frac{P}{\gamma \times T} \quad and\ \gamma = \frac{R}{M_{CH_4}}$$

Where; $M_{CH_4} =\ molecular\ mass\ of\ methane\ (g/mol)$

Then the volume flow rate of methane is:

$$\dot{Q}_\mathrm{CH4} = 0.6\dot{Q}_{bg} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots (Eq.\ 3.16)$$

When substitute (Eq. 16) in (Eq. 15) then volume airflow rate becomes:

$$\dot{Q}_a = 5.7178\dot{Q}_{bg} \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots.. (Eq.\ 3.17)$$

Also substitute (Eq. 3.17) in to (Eq. 3.14);

$$\dot{Q}_{bg} = 0.000913\,\mathrm{m}^3/s\ and\ \dot{Q}_a = 0.00522\,\mathrm{m}^3/s$$

Since $\dot{Q}_a = V_i \times A_i$; then the air inlet velocity in dual fuel mode is:

$$V_i = \frac{\dot{Q}_a}{A_i} = \frac{0.00522\,\mathrm{m}^3/s}{0.00101787\,m^2} = 5.13\,m/s$$

Initially, the biogas is sucked by the vacuum created at the throat in diesel fuel mode and the velocity inside the biogas cylinder is zero. Hence, the velocity of biogas flow is obtained by using Bernoulli's equation (Dahake et al. 2016).

$$V_{bg} = \sqrt{\frac{2\Delta P}{\rho_{bg}}} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots..\dots\dots\dots\dots (Eq.\ 3.18)$$

Where; $V_{bg} =$ biogas flow velocity (m/s)

Then the velocity of biogas flow at the exit of the fuel tube is: $V_{bg} = 37.86\,m/s$.

Since $\dot{Q}_{bg} = V_{bg} \times A_{bg}$; then the fuel inlet area is:

$$A_{bg} = \frac{\dot{Q}_{bg}}{V_{bg}} = \frac{0.000913\,\mathrm{m}^3/s}{37.86\,m/s} = 0.24\ \mathrm{cm}^2$$

The diameter of the biogas inlet depends on the number of biogas inlet holes and given as below (Ramasamy et al., 2010).

$$A_{bg} = \frac{\pi {d_{bg}}^2}{4} n \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots .. (Eq.\ 3.19)$$

Where; d_{bg} = diameter of biogas inlet (m)

n = number of biogas inlet holes which are taken as 2.

Therefore the diameter of each hole is;

$$d_{bg} = \sqrt{\frac{4A_{bg}}{2\pi}} = \sqrt{\frac{4 \times 0.000024}{2\pi}} = 3.9mm$$

Table 3.4: Venturi dimension parameters in dual fuel mode

Air mass flow rate ($\dot{m}_a$) (kg/h)	Biogas mass flow rate ($\dot{m}_{bg}$) (kg/h)	Velocity of inlet air (V_a) m/s	Gas velocity (V_{bg} m/s	Gas inlet area (A_{bg}) m^2
22.23	3.67	5.13	37.86	2.4×10^5

Table 3.5: Different parameters of a venturi mixer at a different beta ratio

Parameters	$\beta = 0.4$	$\beta = 0.5$	$\beta = 0.6$
Throat area (A_t) in cm^2	1.63	2.54	3.66
Velocity at the throat (V_t) in m/s	37.2	23.8	16.54
Biogas inlet velocity (V_{bg}) in m/s	37.86	23.77	15.9
Biogas inlet area(A_{bg}) in cm^2	0.24	0.38	0.57
Diameter of biogas inlet holes (d_{bg}) in mm	3.9	4.94	6
Vacuum at the throat (ΔP) in Pa	798.27	314.17	141
Mach number (M)	0.11	0.07	0.05

3.3.2 Fluent Analysis

In this work, ANSYS workbench was used to generate a multi-block structured mesh for the geometries, and ANSYS Fluent was used to conduct a single phase multi-species flow simulation. Auto CAD was utilized to draw the mixer models without the movable mechanical mechanism as shown in Figure 3.2.

Figure 3.2: Venturi mixer geometry

Since the inlet and outlet diameter of the venturi has to be prefixed based on an engine air intake system. That is:

Inlet manifold diameter = Inlet venturi diameter = Outlet venturi diameter = 36 mm.

To provide homogeneous mixtures of a gas fuel with air (biogas-air) inside the modeled mixer, three convergent angles (20^0, 21^0 and 22^0) and beta ratios of 0.4, 0.5, and 0.6 were analyzed to investigate their effects on the distribution of methane through the air-biogas mixture for enhancing homogeneous mixture at moderate pressure loss at the throat and thereby complete combustion. The mixer models were drawn using Auto CAD 17 software, simulated by Fluent 18.1 and their values are compared with each other.

Computational Fluid Dynamics (CFD) is based on the numerical solutions of the fundamental governing equations of fluid dynamics namely the continuity, momentum, and energy equations. In the present study, a tetrahedral structured mesh is used in all the simulations. The followings are assumptions made in the simulations: -

- Turbulent flow: - This involves the use of a turbulence model, which generally requires the solution of additional transport equations. The standard k-ε model is used in this study. One of the most prominent turbulence models, the k-ε (k-epsilon) model, has been implemented in most general-purpose CFD codes and is considered the industry standard model. It has proven to be stable and numerically robust and has a well-established regime of predictive capability. For general purpose simulations, the k-ε model offers a good compromise in terms of accuracy and robustness.

- Mixing flow without reaction: - This requires the solution of additional equations for mixture fractions of species mass fraction. In this simulation, it is assumed that there is no reaction occurs between air and biogas gas.
- Incompressible: - As the speed involved in reasonably low, it is adequate to assume that the fluids are incompressible. In the incompressible flow, the density of fluids is constant.
- Air inlet boundary: - The condition needs to specify in inlet boundaries is the fixed velocity condition. For the fixed velocity condition, the flow solver determines the mass flow rate applied to each facet of the boundary using the velocity specification, the pressure, the temperature, and the fluid property density.
- Biogas inlet boundary: - Fixed static pressure inlet boundary condition was used at the fuel inlets. By using this boundary condition type, the mass of the fuel inducted into the venturi will be part of the solution. Fuel will be inducted into the venturi because of the low pressure created at the throat.
- Outlet boundaries: - The fixed pressure outlet boundary conditions serve to anchor the system pressure and allow both inflow and outflow to satisfy continuity in the domain.

3.4 Venturi Gas Mixer Manufacturing Process

The designed air-biogas venturi mixer device is manufactured from aluminum material by using a lathe machine and drilling machine. Operations that were implemented during manufacturing the mixer are: **Turning**: turning operation is executed for minimizing the external diameter of the aluminum shaft up to 40mm diameter and also for surface finishing by using a cutting tool on the lathe machine. **Drilling**: the aluminum shaft is then drilled longitudinally at the center entirely by a drilling tool of 18mm diameter and also biogas inlet holes are drilled transversally by a diameter of 5mm drilling tool at the throat after divided the circumference of the throat into two equal places and scratched by scratcher. **Tapering**: for tapering the convergent section firstly the cutting tool is set at 22^0 on the compound head and then tapering the air inlet section. Secondly, the cutting tool is again set at 8^0 on the compound head for tapering the divergent (mixture outlet) section. Finally, **the surface finishing** of the tapered and turned surface was performed by using glass paper.

3.5 Compositions and Storage Mechanism of Biogas

3.5.1 Compositions of Biogas

The composition of raw biogas, which is obtained from cow dung, is analyzed by Geotech biogas analyzer from Addis Ababa University and all its contents are listed in Table 3.6 below. Figure 3.3 below shows the sample raw biogas.

Table 3.6: Composition of biogas

Contents	Percentage (% by Vol.)
CH_4	44.5
CO_2	10.7
O_2	8.7
H_2O	0 ppm
Traces gases (H_2S, H_2, N_2, etc.)	36.1

Figure 3.3: Sample raw biogas stored in glucose bag

3.5.2 Storage of Biogas

The raw biogas, which is initially inside the digester, is stored in the tire tube by using an ordinary compressor as shown in figure 3.4 below. At the outset, the compressor inlet was connected with the digester biogas outlet tube by a hose. Similarly, the compressor outlet was connected with the tire pressure gauge inlet and the tire pressure gauge outlet is then connected to the tire tube inlet tip after removing the one-way valve at the tire tube tip.

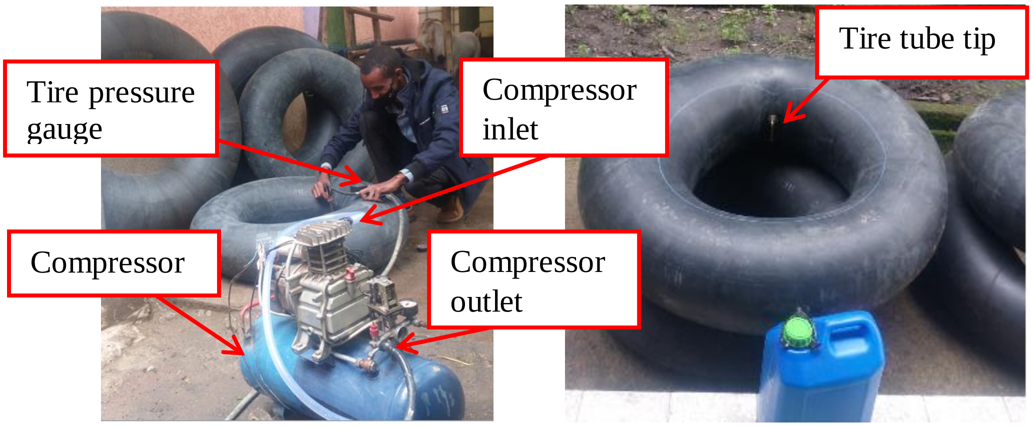


Figure 3.4: Compressed biogas

3.6 Experimental Setup and Procedure

The experimental test rig consists of a single-cylinder, four-stroke, naturally aspirated, direct injection, and computer-controlled test bench diesel engine TBMC3-02. It is equipped with an eddy current asynchronous motor dynamometer (AM-1) for loading the engine. The setup also includes a fuel supply system, water cooling, and lubrication system and exhaust emissions. The engine produced a maximum power of 2.2 kW and a maximum torque of 7.4 N-m. The schematic and photographic representation of the experimental setup is illustrated in Figure 3.5 and Figure 3.6 respectively. The setup enables the evaluation of performance parameters and emission constituents of the diesel engine. The engine performance parameters include Tb, Pb, BSFC and BTE. The technical specification of a test engine is shown in Table 3.7 below. The engine exhaust emissions levels are measured using an SV-50 automotive emission analyzer. The CO, CO2, emissions were measured in percentage volume (%Vol), and the HC and NOX emissions were measured in parts per million (ppm) during each run of the engine.

The experimental investigation was done on three different volume flow rates of biogas with diesel fuel in dual fuel mode. The timing and duration of the injection of diesel were controlled by a factory engine electronic control unit (ECU) in which none of the programmed values was modified. Besides, all the sensors located on the engine were completely wired to the ECU by following the factory setting. In the engine, providing a compression ratio of 21:1, the nozzle opening pressure (210 bar) and the standard of the injection timing 23º bTDC.

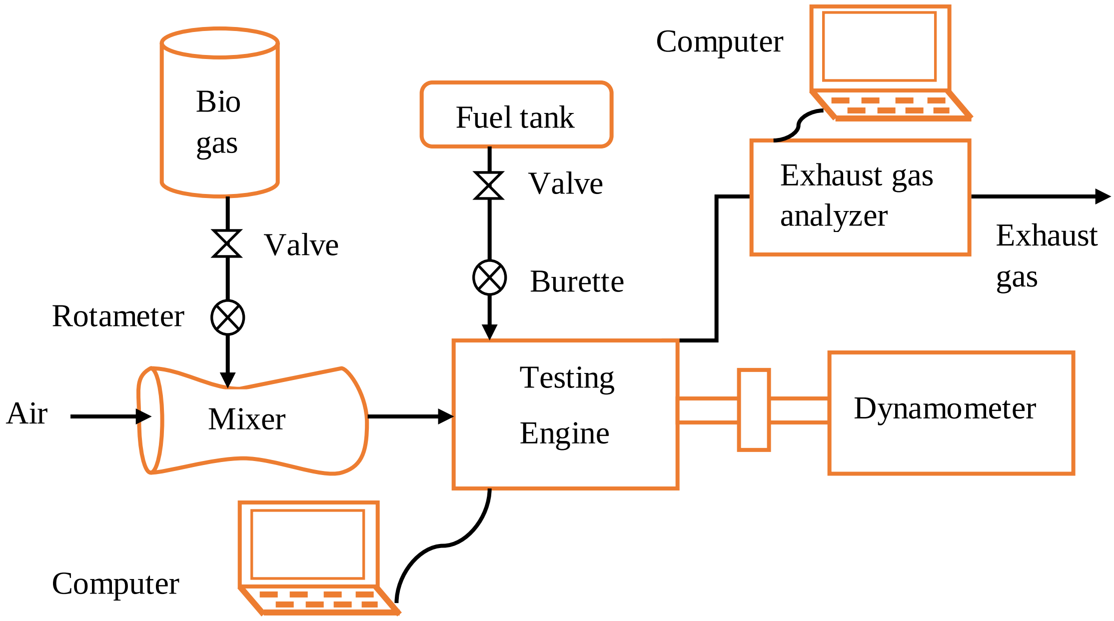


Figure 3.5: Schematic experimental setup of the test engine

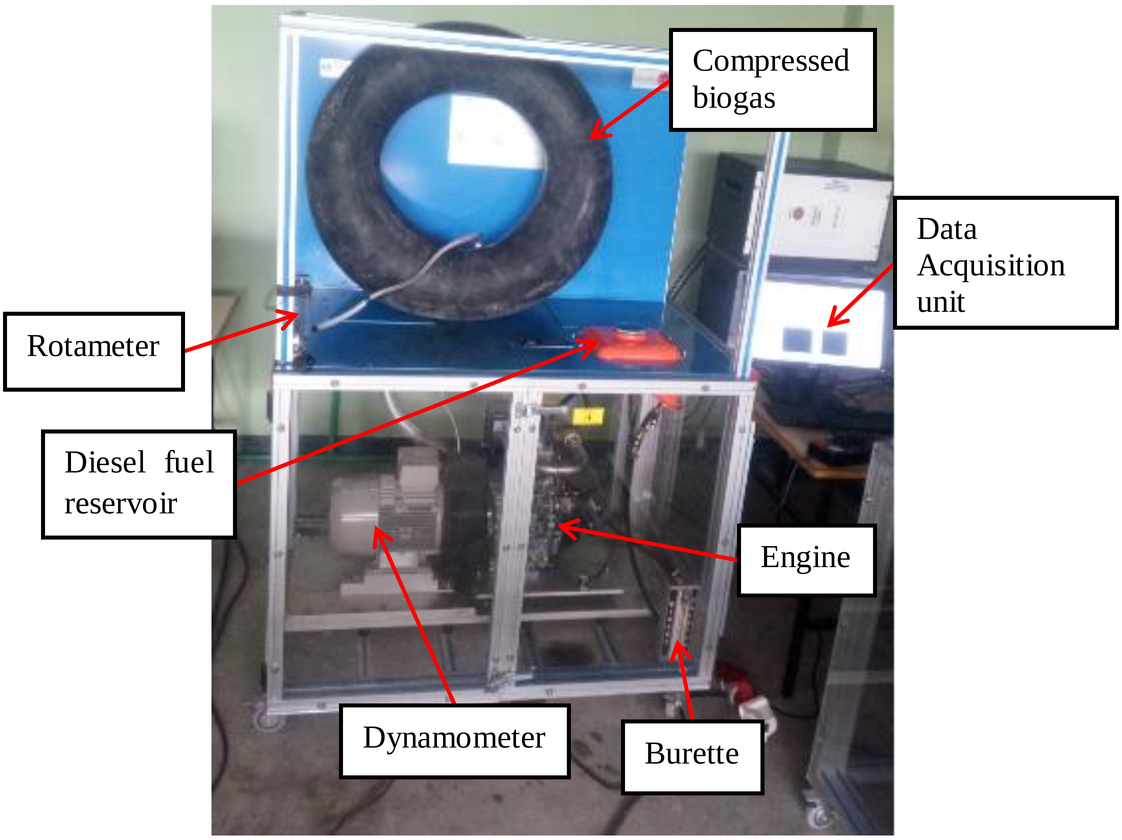
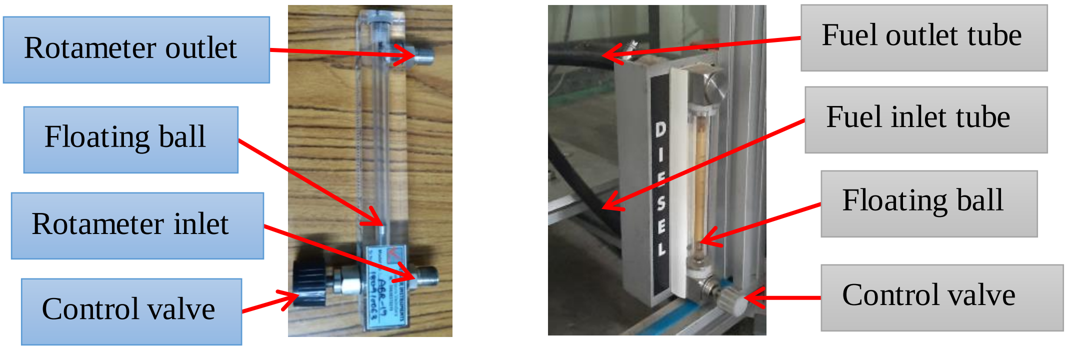


Figure 3.6: Photographic experimental setup of the test engine

The various experiments were executed once the venturi gas mixer is installed at the air intake manifold after the airflow sensor as shown above in figure 3.7b. The biogas flow control valve is to be opened after the engine startup in diesel fuel mode. As the biogas flow control valve starts to open then the vacuum created at the throat in diesel-only mode sucks biogas from its source to the mixer. Hence the amount of airflow rate to the engine cylinder decreases and substituted by biogas. Frequently the engine starts to operate in dual fuel mode. Whereas the amount of diesel fuel injected for ignition decreases due to the engine inject a pilot fuel to ignite the charge according to the amount of air entered (i.e due to a decrease in the amount of air entered). The properties of raw biogas and the commercial diesel fuel used in this study are given in Table 3.8. In order to achieve the intended main objectives of the thesis, the engine was performed at various loads. Each trial was repeated three times and on different days, enough care was taken to load the engine accurately at each step of load and to maintain ambient conditions constant. Figure 3.7 shows the position of the venturi gas mixer on the air intake manifold of the engine.

a) Before installing gas mixer b) After installing a gas mixer

Figure 3.7: Installation of venturi gas mixer at the air intake manifold



a. Rotameter b. Burette

Figure 3.8: Flow meters

Initially, the engine was started with conventional diesel fuel at zero engine loads. All the tests were taken after the engine reached its operating temperature and at a constant engine speed of 1500 RPM, which is set fixed on the computer. The diesel engine is tested initially with diesel fuel for baseline data for each load condition according to the baseline test matrix shown in Table 3.9. The load was ranging from zero to 80% in a step-up of 10% and as the load varies consequently the brake power and brake torque varied and displayed on the data acquisition computer. After restarting the engine with diesel fuel, the biogas flow control valve is to be opened slowly for supplying biogas from the tire tube and employ the engine in dual fuel mode. The flow rate was adjusted and regulated by Rotameter (which is found in Defence Engineering College, Debrezeit, Ethiopia) as shown in Figure 3.8a above. The flow rate of biogas in dual fuel mode was demonstrated with 2L/min, 4L/min, and 6L/min as the DF test matrix is shown below in Table 3.10. Then the performance and emissions of the engine were tested and the values were recorded with diesel (D) fuel, D + BG@2L/min flow rate, D + BG@4L/min flow rate, and D + BG@6L/min flow rate for each load conditions at a constant engine speed of 1500 RPM. In each experiment, the various emissions concentration such as CO, HC, CO2 and NOx emissions were measured using an exhaust gas analyzer, and the fuel flow rate was also recorded using the fuel burette as shown in Figure 3.8b above. The performance parameters of BSFC and BTE were calculated by using equation (3.20 - 3.23). Finally, the engine performance and emission parameters were plotted against varying engine loads.

Table 3.7: Test engine TBMC3-02 technical specification

Engine model	TM3-02
Engine type	Compression-Ignition
Fuel type	Diesel oil
Number of cylinders	1
Type of air intake	Naturally aspirated
Cooling type	Air
Compression ratio, CR	21:1
Bore diameter	69mm
Stroke	60mm
No. of stroke	4 stroke
Torque	7.4 N.m
Engine power	2.2 kW
Internal dia. of the air inlet manifold (D)	36mm

Table 3.8: Properties of diesel and biogas (Bouguessa et al., 2020).

Properties	Diesel	Biogas
Lower heating value (MJ/kg)	42.061	20.3958*
Density (kg/m^3)	840	1.12*
Auto ignition temperature (^0C)	280	650
Stoichiometric air-fuel ratio	14.60	17
Cetane number	49	-
Octane number	-	130
Laminar burning velocity (m/s)	0.5	0.2

* Calculated value (see section 3.3.1 and 3.8).

3.6.1 Experimental Test Matrix

An experimental test matrix is the number of tests that were being executed and guidance for experimental work. Baseline and DF mode experimental test matrix are listed below.

Table 3.9: Baseline data experimental test matrix

<table>
<tr><th rowspan="2">Mode of operation</th><th colspan="2">Engine operating parameters</th><th rowspan="2">Evaluating Engine Parameters</th></tr>
<tr><th>Fixed CR, IP and IT</th><th>Load</th></tr>
<tr><td rowspan="9">Diesel fuel mode</td><td rowspan="9">Set fixed as factories recommender</td><td>0%</td><td rowspan="9">a. Performance parameters
i. Brake Torque (T_b)
ii. Brake Power (P_b)
iii. Brake specific fuel consumption (BSFC)
iv. Brake thermal efficiency (BTE)
b. Emission Parameters
i. CO
ii. HC
iii. CO_2
iv. NOx
v. O_2</td></tr>
<tr><td>10%</td></tr>
<tr><td>20%</td></tr>
<tr><td>30%</td></tr>
<tr><td>40%</td></tr>
<tr><td>50%</td></tr>
<tr><td>60%</td></tr>
<tr><td>70%</td></tr>
<tr><td>80%</td></tr>
</table>

Table 3.10: Experimental test matrix for dual fuel mode operation

Mode of operation	Engine operating parameters		Biogas flow rate	Evaluating Engine Parameters
	Fixed CR, IP and IT	Load		
Dual fuel mode (Diesel-Biogas)	Set fixed as factories recommender	0%	2L/min, 4L/min, 6L/min	a. Performance parameters i. Brake Torque (T_b) ii. Brake power (P_b) iii. Brake specific fuel consumption (BSFC) iv. Brake thermal efficiency (BTE) b. Emission Parameters i. CO ii. HC iii. CO_2 iv. NOx v. O_2
		10%	2L/min, 4L/min, 6L/min	
		20%	2L/min, 4L/min, 6L/min	
		30%	2L/min, 4L/min, 6L/min	
		40%	2L/min, 4L/min, 6L/min	
		50%	2L/min, 4L/min, 6L/min	
		60%	2L/min, 4L/min, 6L/min	
		70%	2L/min, 4L/min, 6L/min	
		80%	2L/min, 4L/min, 6L/min	

3.7 Performance Parameter Analysis

The performance of the diesel engine such as brake power, brake torque, brake specific fuel consumption and brake thermal efficiency was evaluated by fueled the engine; diesel (D) fuel, D + BG@2L/min flow rate, D + BG@4L/min flow rate and D + BG@6L/min flow rate with different engine load conditions.

i. Brake Torque (T_b)

Brake torque is the torque available at the flywheel in N.m. The dynamometer on the TBMC3-02 test stands used to load the engine from minimum (zero loads) to the maximum. By doing this in steps, torque values are displayed on the computer which is integrated with the TBMC3-02 test stand.

ii. Brake Power (P_b)

One of the most important metrics for engine performance is output power. Brake power is the actual engine power output at the output shaft of the engine in kW. This power is related to the torque applied to the crankshaft. Similar to brake torque, the brake power is also displayed on the computer during the engine runs and its value changes as the engine loads changes.

iii. Brake thermal efficiency

Brake thermal efficiency is the ratio of output brake power to the energy supplied by the fuel. For pure diesel mode, it is calculated as (Gurung et al., 2018):

$$\eta_{bth}(\%) = \frac{3600 \times P_b}{\dot{m}_d \times LHV_d} \times 100 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (Eq.\ 3.20)$$

Where, η_{bth} = brake thermal efficiency

$\dot{m}_d$ = mass flow rate of diesel in pure diesel mode in kg/h

While in the case of dual-fuel mode, brake thermal efficiency is given by the following formula (Bouguessa et al., 2020 and Gurung et al., 2018):

$$\eta_{bth}(\%) = \frac{3600 \times P_b}{\dot{m}_{bg} \times LHV_{bg} + \dot{m}_{dd} \times LHV_d} \times 100 \ldots\ldots\ldots (Eq.\ 3.21)$$

Where, $\dot{m}_{dd}$ = mass flow rate of diesel fuel in a dual mode in kg/h

$\dot{m}_{bg}$ = mass flow rate of biogas in kg/h

LHV_{bg} = lower heating value of biogas in kJ/kg

LHV_d = lower heating value of diesel in kJ/kg

iv. Brake Specific Fuel Consumption (bsfc)

Specific fuel consumption is the measure of the engine's ability to produce power by utilizing the energy content of the fuel. The specific fuel consumption of an engine should be minimized for the desired power output to optimize engine performance (Sandalc et al., 2019). Brake specific fuel consumption is the ratio of the mass flow rate of fuel consumed by the engine to the brake power in which the engine is produced. In a dual-fuel engine, engine power is obtained from the combustion of two fuels. That is primary fuel biogas and secondary fuel diesel. Each fuel contributes a percentage to the total energy which is produced as output power. During the engine run with biogas-diesel dual fuel mode, fuel consumption measurement was accomplished by measuring the biogas fuel and diesel fuel consumption separately and their energy share to the total energy was also calculated separately.

When the testing engine run in diesel fuel mode, bsfc is calculated as (Ambarita H. , 2017):

$$\mathrm{bsfc} = \frac{\dot{\mathrm{m}}_{\mathrm{d}}}{\mathrm{P_b}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (Eq.\ 3.22)$$

Where; $\mathrm{bsfc} = \text{brake specific fuel consumption in kg/kWh}$

While in the case of diesel engine operated in dual fuel mode, bsfc is given by the following formula (Ambarita H. , 2017):

$$\mathrm{bsfc} = \frac{\dot{\mathrm{m}}_{\mathrm{dd}} + \dot{\mathrm{m}}_{\mathrm{bg}}}{\mathrm{P_b}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (Eq.\ 3.23)$$

3.8 Biogas Energy Share (BGES)

In dual fuel mode operation both the primary fuel (biogas) and pilot fuel (diesel) fuel contributes to the total energy in which a certain amount of output power is generated. During combustion of dual fuel, the consumption of the primary fuel remains unchanged, due to its flow rate is fixed, with the change in the load, while the pilot fuel consumption varied as the load varies.

The energy share of biogas in a dual fuel operation is the ratio of energy supplied by biogas (primary fuel) to the sum of the energy supplied by the primary fuel and the pilot fuel. Thus it is calculated as (Barik & Murugan, 2014 and (Bouguessa et al., 2020):

$$\mathrm{BGES} = \frac{\dot{\mathrm{m}}_{\mathrm{bg}} \times \mathrm{LHV_{bg}}}{\dot{\mathrm{m}}_{\mathrm{bg}} \times \mathrm{LHV_{bg}} + \dot{\mathrm{m}}_{\mathrm{dd}} \times \mathrm{LHV_d}} \times 100 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (Eq.\ 3.24)$$

Where; $\mathrm{BGES} = \text{Biogas energy share}$

As the engine started to operate in dual fuel mode diesel fuel consumption was reduced and replaced by biogas. The percentage of diesel fuel replaced by biogas which is known as **replacement ratio (r)** and given by (Verma et al., 2018 and Ambarita, 2017)):

$$\mathrm{r} = \frac{\dot{\mathrm{m}}_{\mathrm{d}} - \dot{\mathrm{m}}_{\mathrm{dd}}}{\dot{\mathrm{m}}_{\mathrm{d}}} \times 100 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (Eq.\ 3.25)$$

Biogas energy share (%) and diesel fuel replacement ratio (%) for each mode of operations with different engine load conditions are shown in Appendix B.

The **lower heating value** of biogas is calculated by the following formula (Von Mitzlaff, 1988):

$$LHV_{bg} = \frac{V_{CH_4}}{V_{tot}} \times \rho_{CH_4act} \times LHV_{CH_4} \ldots\ldots\ldots\ldots\ldots..\ldots\ldots\ldots.(Eq.\ 3.26)$$

Where; $LHV_{bg} = lower\ heating\ value\ of\ a\ given\ biogas\ (kJ/kg)$

$V_{CH_4} = percentage\ volume\ of\ methane\ in\ biogas\ (0.445)$

$V_{tot} = total\ percentage\ which\ is\ taken\ as\ 1$

$\rho_{CH_4act} = actual\ density\ of\ methane\ (kg/m^3)$

$LHV_{CH_4} = lower\ heating\ value\ of\ methane\ (50{,}000\ kJ/kg)$

Since the actual density of methane is calculated as follows (Von Mitzlaff, 1988):

$$\rho_{CH_4act} = \rho_{CH_4std} \times \frac{P_{act}}{P_{std}} \times \frac{T_{std}}{T_{act}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(Eq.\ 3.27)$$

Where; $\rho_{CH_4std} = density\ of\ methane\ at\ standard\ condition\ (0.72\ kg/m^3)$

$P_{act} = actual\ or\ ambient\ pressure$

$P_{std} = pressure\ at\ standard\ condition\ (101325\ Pa)$

$T_{std} = temperature\ at\ standard\ condition\ (273K)$

$T_{act} = actual\ or\ room\ temperature\ (298\ K)$

For calculation, the pressure ratio is taken as 1. Then the actual density of methane is:

$$\rho_{CH_4act} = 0.66 kg/m^3$$

Therefore, the lower heating value of the given biogas becomes:

$$LHV_{bg} = 14685\ kJ/m^3 = 20395.8\ kJ/kg$$

3.9 Emission Parameters and Measuring Procedure

The emission parameters of a dual fuel mode diesel engine such as CO, CO_2, HC, O_2, and NOx were measured by using an exhaust gas analyzer when the engine operates in diesel only and dual fuel mode at each load conditions. During the experiments, exhaust gases were measured using Automobile Exhaust Gas Analyzer SV-50 (which is found in Federal TVET Institute, Ethiopia) as shown below in Figure.3.9 and its measuring range is listed in Table 3.11 below. Its detailed product description is described in Appendix C-I.

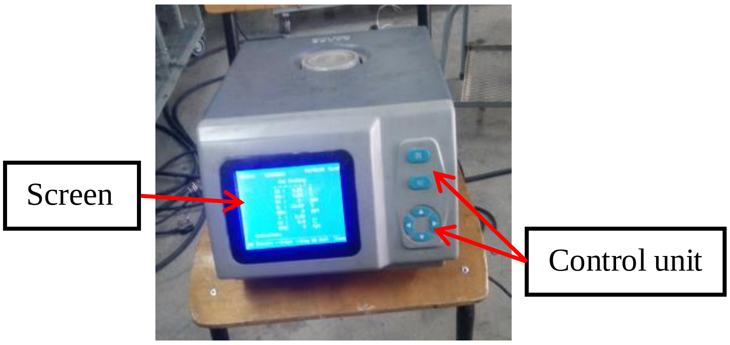


Figure 3.9: Automobile exhaust gas analyzer SV-50

Table 3.11: Measurement range of SV-50 automobile exhaust gas analyzer

species	Measurement range	Resolution	Allowed error	Relative error
HC	0 - 1000 ppm	1 ppm Vol.	±12ppm Vol.	±5%
CO	0 - 10.0 in % Vol.	0.01% Vol.	±0.06% Vol.	±5%
CO_2	0 - 20.0 in % Vol.	0.1% Vol.	±0.5%	±5%
O_2	0 - 25.0 % Vol.	0.01% Vol.	±0.1% Vol.	±5%
NOx	0 – 800 ppm	1 ppm	±10ppm	±5

Measuring Procedure

Initially connect (plugin) the exhaust gas analyzer from the power source and turn it ON. Since the probe and the sampling tube were connected to the gas analyzer. Then the probe tip was closed by plastic for at least 10 minutes before starting measurement. After 10 minutes remove the enclosed plastic and insert its probe tip into the engine exhaust pipe. Some portion of exhaust gases was taken by the gas analyzer through the sampling tube and sensing it and then measurement was started by pressing the record icon on the control unit of the analyzer. The content of the exhaust gas was started to display on the gas analyzer control unit display. The readings of HC (in ppm), NOx (in ppm), CO_2 (in % Vol.), O_2 (in % Vol.), and CO (in % Vol.) for each test were allowed to stabilize from 2 to 5 minutes and the results were recorded.

CHAPTER FOUR

RESULT AND DISCUSSION

4.1 Introduction

The experimental work analysis on the performance and emission characteristics of a dual fuel diesel engine are presented in this chapter. The simulation results of the design parameters of the mixer are also discussed in this chapter. This chapter also explains the effect of variation of load, and biogas flow rate/ percentage substitution of biogas in biogas-diesel dual fuel mode diesel engine performance and emission characteristics.

4.2 Simulation Result of Venturi Gas Mixer

Figure 4.1 shows the simulation result of methane mass fraction at 22 degrees with different beta ratios of gas mixing device, which is analyzed based on the pressure and velocity at the throat by varying convergent angle and beta ratios. From the venturi principle, the mixer must produce lower pressure to create a vacuum at the throat which is used to suck gas fuel easily and higher velocity at the mixer throat improves the mixing of air with biogas. The methane mass fraction produced at the outlet is the selection criteria. The pressure drop at the throat influences the throat velocity of a mixer. This consequently affects the turbulence and mixing of species in the downstream of the mixer. Moreover, as the pressure drop increases at the throat, more biogas is sucked. According to Bernoulli's principle, when a fluid passes through a venturi, its velocity raises and consequent increase in kinetic energy as a result its pressure decline at the throat. The high velocity at the throat portion tends to decrease in the divergent portion, which gives rise to turbulence in the downstream. This induced turbulence mixes the species. So, as the convergent angle increases from 21 degrees to 22 degrees the vacuum created at the throat increases due to an increase in velocity at the throat. In other words, lower pressure is observed and thereby change of pressure is more, which facilitates induction or suction of biogas to the engine through the mixer device at a higher convergent angle. As observed from the simulation results lowest pressure is observed at a 22-degree convergent angle. So, lower throat pressure and medium-high velocity result in similar methane distribution on the downstream portion of the venturi at a short distance from the throat as shown in Figure 4.1 at a beta ratio of 0.5.

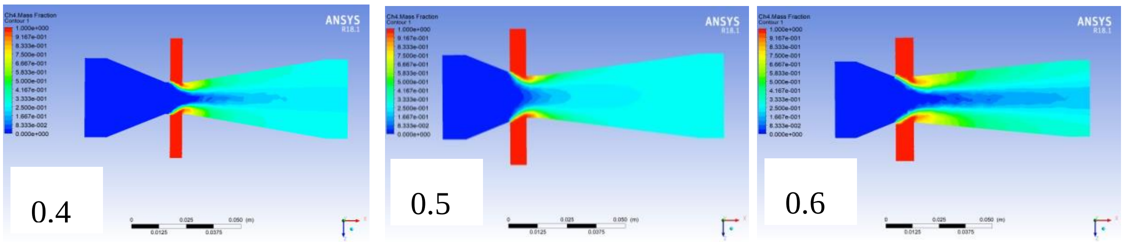


Figure 4.1: Methane mass fraction contour at 22 degree

4.3 Engine Performance Characteristics

The performance characteristics such as brake torque, brake power, brake thermal efficiency and brake specific fuel consumption of a diesel engine when it operated in diesel fuel and diesel with biogas dual fuel mode at different engine loads are discussed in the following sections below.

4.3.1 Brake Torque (T_b)

Figure 4.2 below shows that the variations of brake torque versus engine load that almost similar trends for all modes of operation and Table 4.1 show its numerical values. From Figure 4.2 below it is shown that at lower loads slightly torque is lower and increases as engine load increases for both modes of operation. The brake torque of diesel with biogas dual fuel was lower than the diesel fuel due to the lower calorific value of biogas. As the biogas flow rate increases the brake torque slightly decreases. This is because of an increase in BGES and diesel fuel replacement ratio. In other words, as the biogas flow rate increases the amount of pilot fuel injected to the combustion decreases. On the other hand, due to the presence of CO_2 as the biogas flow rate increases more carbon dioxide enters into the combustion chamber and acts as diluents, which causes lower in-cylinder temperature, as a result, incomplete combustion, occurs. This is a reason for the decrease in brake torque as the biogas flow rate increases with respect to neat diesel fuel.

Generally, the average engine torque reductions of D + BG@2L/min, D + BG@4L/min, and D + BG@6L/min flow rate with respect to diesel fuel mode were obtained 3.65%, 5.47% and 6.68% respectively.

Table 4.1: Brake torque for all tests

Load (%)	Brake Torque (N.m)			
	Diesel	D + BG@2L/min	D + BG@4L/min	D + BG@6L/min
10	3.2	3.1	3.1	3.1
20	3.6	3.5	3.45	3.4
30	3.8	3.7	3.6	3.6
40	4	3.9	3.8	3.7
50	4.2	4.1	4	3.9
60	4.5	4.3	4.2	4.2
70	4.7	4.5	4.5	4.4
80	4.9	4.6	4.55	4.5

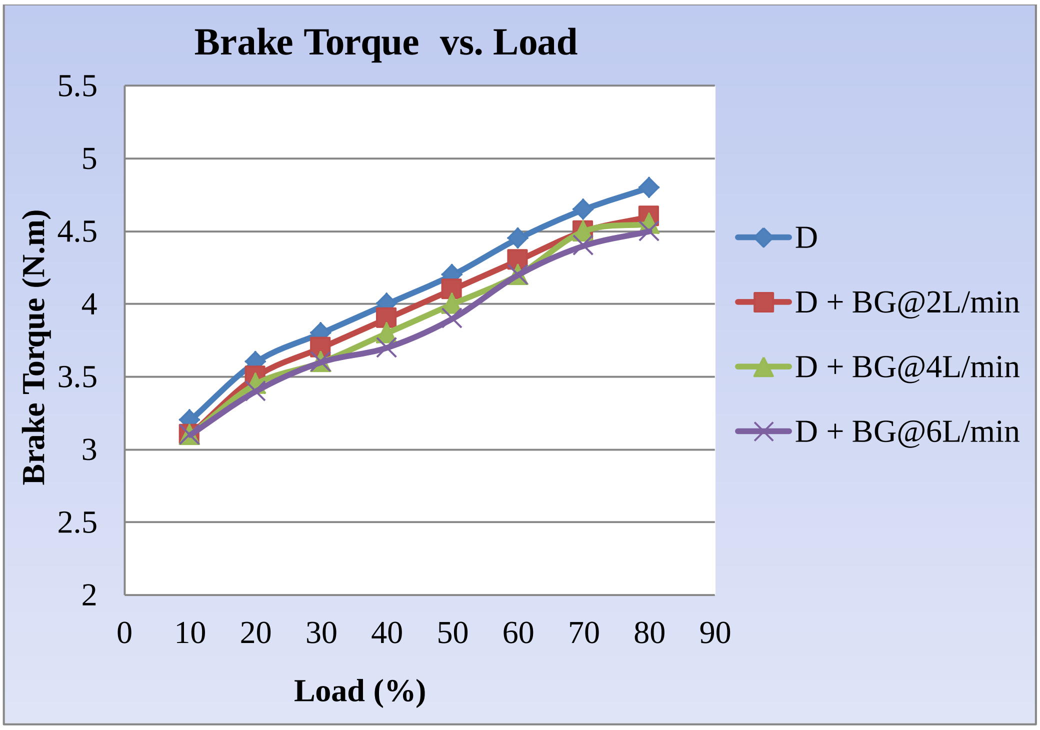


Figure 4.2: Variation of brake torque with respect to engine load

4.3.2 Brake power (P_b)

The numerical values of brake power for both diesel and DF modes at various loads are shown in Table 4.2 below.

Table 4.2: Brake power for all tests.

Load (%)	Brake Power (kW)			
	Diesel	D + BG@2L/min	D + BG@4L/min	D + BG@6L/min
10	0.26	0.25	0.25	0.25
20	0.4	0.39	0.38	0.38
30	0.52	0.5	0.49	0.49
40	0.65	0.63	0.61	0.6
50	0.79	0.78	0.76	0.74
60	0.97	0.93	0.9	0.9
70	1.14	1.09	1.09	1.06
80	1.33	1.23	1.22	1.21

As shown in Figure 4.3 below the variations of brake power versus engine load that almost similar trend for that of torque. As engine load increases, brake power also increasing up for both modes of operation. However, in dual fuel mode, it is slightly lower as compared to diesel mode. Since the gaseous fuel has lower energy density as compared to the pilot fuel. So, the performance of the engine depends on the amount of gaseous and pilot fuel used. As the engine operated in dual fuel mode the amount of air entered to the engine cylinder decreases due to substituted by biogas. This will create a chance for incomplete combustion due to the rich in the air-fuel ratio. Additionally, the higher auto-ignition temperature of biogas, due to the presence of CO_2 leads to a moderate decrease rise in pressure and thereby temperature during combustion, and also due to longer ignition delay of pilot fuel. These cause a slight decrease in power for dual fuel mode as compared with neat diesel fuel mode. On the other hand, as the biogas flow rate increases, there is a small reduction of power throughout the entire load range. This is due to the lower calorific value of biogas as well as the available oxygen in the combustion chamber decreases. Generally, the average brake power reduction of D + BG@2L/min, D + BG@4L/min, and D +

BG@6L/min flow rate with respect to diesel mode were obtained 2.81%, 4.45% and 5.61% respectively.

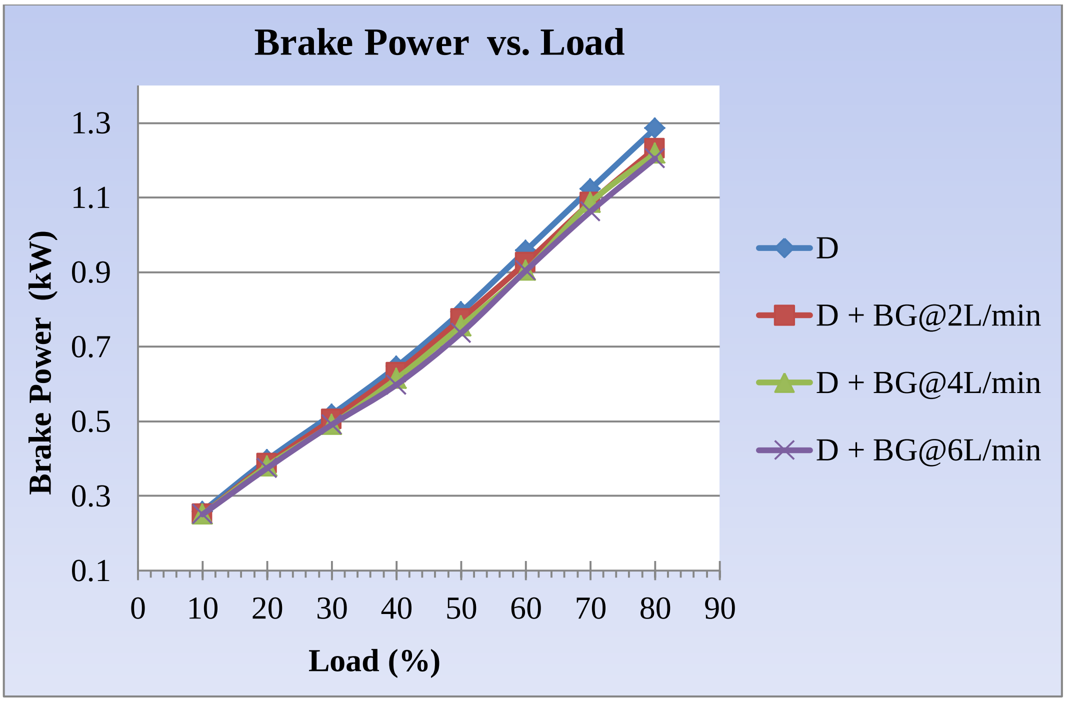


Figure 4.3: Variation of brake power with respect to engine load.

4.3.3 Brake thermal efficiency (BTE)

It is the measure of the engine's performance in terms of the conversion of heat energy into mechanical energy (Deheri et al., 2020). Figure 4.4 shows the variation of brake thermal efficiency for all biogas flow rates with diesel and diesel fuel at all loads and Table 4.3 shows its numerical value. Overall the brake thermal efficiency for all modes of the operation increases as the load increases but noticed lower in dual-fuel mode operation as compared to pure diesel mode. This is due to the high resistance to auto-ignition properties of biogas, due to its high octane number and high auto-ignition temperature acts as a heat sink in the combustion period of dual-fuel which causes to decrease in combustion temperature. This leads to lowers energy conversion efficiency of the dual fuel as compared to diesel fuel and thereby decreases BTE. Similarly, the presence of CO_2 in biogas, CO_2 behaves as an inert gas that subsequently increased specific heat of the intake charge and lowers the local gas temperature by absorbing heat energy from the combustion. This affects the burning speed of biogas-air charge and causes a reduction in flame propagation which results lower in brake thermal efficiency of dual-fuel mode. A similar trend was reported by (Rosha et al., 2018). On the other hand, in dual-fuel mode operations as biogas flow rate increases BTE decreases. This is due to an increase in the induction of biogas leads to further decreases the flame propagation speed and resulted in lower BTE. Generally, an average BTE reduction of D + BG@2L/min, D + BG@4L/min, and D + BG@6L/min flow rate with respect to diesel mode were obtained 15.94%, 20.04% and 23.58% respectively.

Table 4.3. Brake Thermal Efficiency for all tests.

Load (%)	Brake Thermal Efficiency (%)			
	Diesel	D + BG@2L/min	D + BG@4L/min	D + BG@6L/min
10	9.04	6.4	6.08	5.12
20	15.44	11.96	11.23	10.55
30	20.56	16.74	15.47	14.71
40	23.74	20.02	19.55	18.57
50	27.62	23.36	22.41	21.44
60	29.52	25.28	24.32	23.35
70	30.46	26.58	24.98	24.31
80	31.16	27.31	25.92	25.26

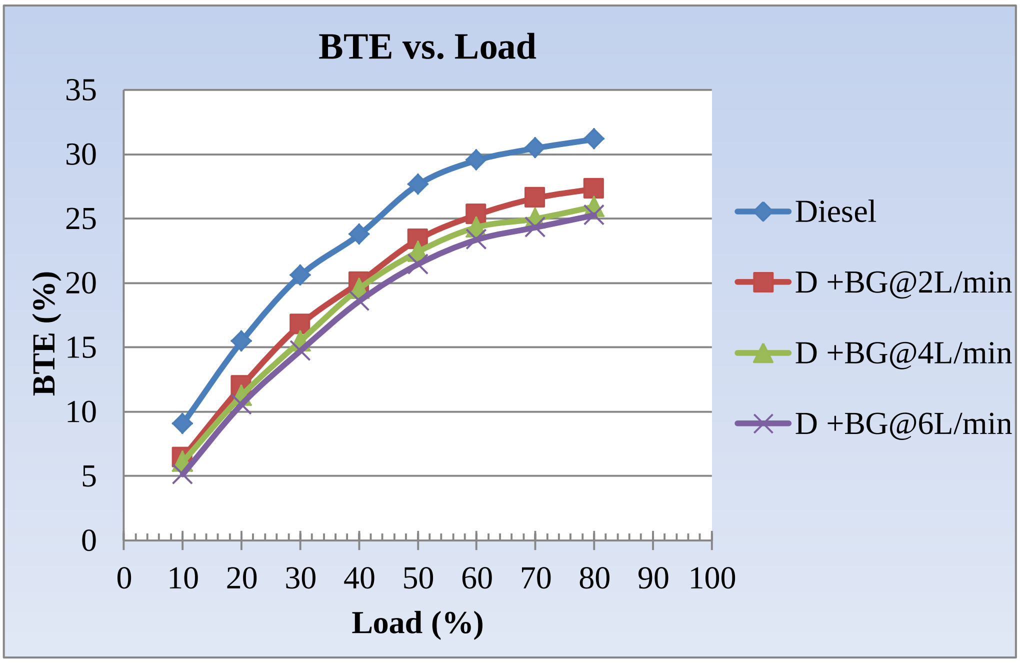


Figure 4.4: Variation of brake thermal efficiency versus engine load.

4.3.4 Brake specific fuel consumption (bsfc)

Brake specific fuel consumption depends on the heating value of the fuel (Sandalc et al., 2019). Figure 4.5 shows brake specific fuel consumption with respect to engine load for diesel fuel and all the biogas mixtures. Brake specific fuel consumption for both modes was found to be high at a low load of the engine. This is due to lower output power at a lower load. However, it was found to be lesser at high engine load for both modes of operations because of an increase in combustion rate due to high air-fuel ratio and high combustion temperature. Table 4.4 below shows the numerical value of brake specific fuel consumption for both neat diesel and dual-fuel operations.

Table 4.4: Brake specific fuel consumption for all tests.

load (%)	bsfc (g/kWh)			
	Diesel	D + BG@2L/min	D + BG@4L/min	D + BG@6L/min
10	488	571	604	634
20	430	496	532	567
30	380	421	442	455
40	351	389	406	416
50	310	352	352	367
60	288	315	327	333
70	275	295	302	313
80	271	284	287	291

From Figure 4.5 it is observed that, supplying biogas leads to higher fuel consumption as compared to diesel mode throughout the load range. This is due to the low energy density of and slow-burning of biogas causes higher BSFC in dual fuel mode operation. Moreover, due to raw biogas contains more percentage of non-combustible components which reduces its fuel quality. In other words, due to a lower percentage concentration of methane in raw biogas leads to having lower calorific value. Thus, lower calorific values of biogas lead to having higher brake specific fuel consumption in DF as compared to diesel fuel mode operation.

On the other hand, in dual-fuel mode, an increase in biogas flow rate leads to an increase in brake specific fuel consumption. This is because increasing the amount of biogas flow rate further decreases the amount of injected pilot fuel. Thus decreases the energy content of the mixture and causes higher BSFC as an increase in biogas flow rate.

Generally, the average brake specific fuel consumption increment of dual fuel mode using D + BG@2L/min, D + BG@4L/min, and D + BG@6L/min flow rate with respect to diesel mode was obtained 11.82%, 16.43% and 20.87% respectively.

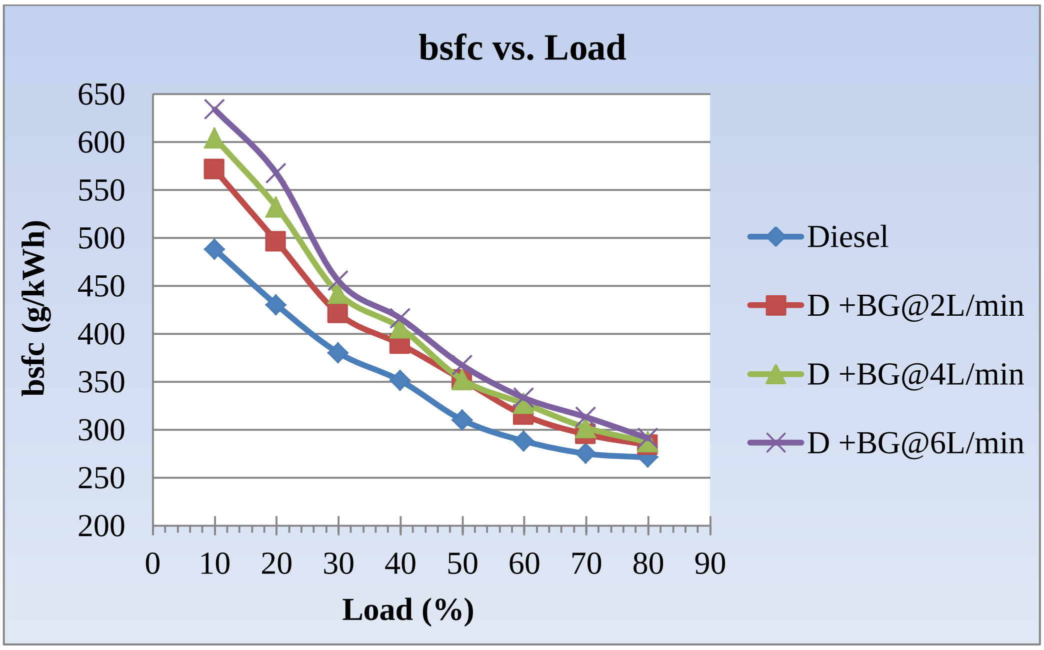


Figure 4.5: Variation of brake specific fuel consumption with respect to engine load.

4.4 Biogas Energy Share (BGES)

The variation in energy share of biogas with engine load is depicted in Figure 4.6 and its numerical values are listed in Table 4.5 blow. It can be observed from the Figure 4.6 that, the biogas energy share is high at low loads, while it decreases with increasing load, for all the flow rates of biogas in dual fuel operation. This is due to more diesel consumption at relatively high loads than that of low loads because of, higher thermal load imposed on the engine inhibits more diesel consumption and since biogas flow rate was set fixed for each DF operation, whereas pilot fuel varies. On the other hand, as biogas flow rate increases its energy share increases due to decrease in pilot fuel in compared with low biogas flow rate operations. But, as the load increases the diesel consumption is also increasing for all operations. At a biogas flow rate of 2L/min (0.134kg/h), 4L/min (0.268kg/h) and 6L/min (0.4kg/h), an average biogas energy share of 24.98%, 40.95%, 52.32% and average diesel replacement ratio of 13.41%, 18.16 and 23.29 was noticed respectively. Maximum biogas energy share of 65.82% with maximum diesel fuel replacement ratio of 30.53% was observed at 6L/min biogas flow rate on low engine load of 10%. Generally, increasing in biogas flow rate increases its energy share whereas diesel replacement ratio also increases.

Table 4.5: Biogas energy shares in DF mode

Load (%)	BGES (%)		
	D +BG@2L/min	D +BG@4L/min	D +BG@6L/min
10	35.6	54.78	65.82
20	30.88	48.78	60.25
30	28.46	45.54	57.25
40	25.07	41.36	52.86
50	24.05	39.84	52.39
60	20.63	35.7	47.69
70	18.57	32.33	43.24
80	16.59	29.36	39.02

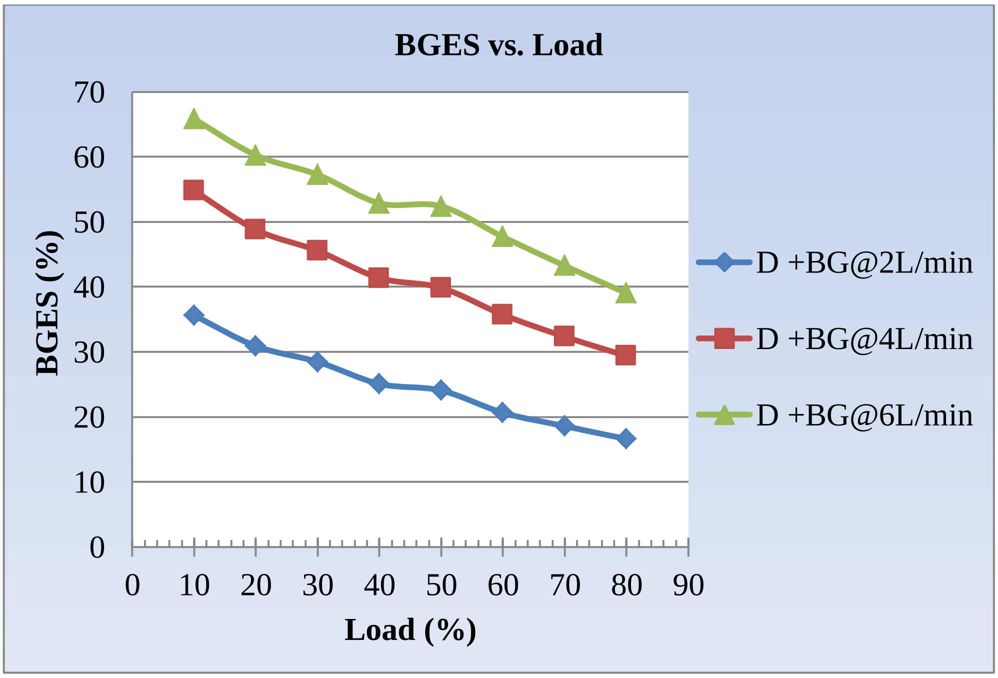


Figure 4.6: Variation of BGES with respect to engine load

4.5 Exhaust Gas Emissions

Knowing the exhaust emissions of a specified fuel is an important issue for future controlling its emissions through different options, either searching alternative fuels which results in less emission or optimizing the engine operating parameters. This portion addresses the emissions of pure diesel and biogas-diesel dual fuel mode diesel engine. Now emissions of CO, CO_2, HC, $O_{2,}$ and NO_x which were measured by using Automobile Exhaust Gas Analyzer SV-50 when the engine run in dual fuel mode at each load with different biogas flow rate are briefly discussed in this section with comparison to neat diesel fuel emissions.

4.5.1 Exhaust Emissions of Carbon Monoxide (% Vol.)

CO emission is produced due to the incomplete oxidation of carbon atoms caused by the fuel-rich mixture in which oxygen availability is less. It is also formed due to a lean mixture where the combustion temperature is below 1450 K (Wei & Geng, 2016). So, that CO emission depends on the oxygen concentration in the air, carbon content, and combustion efficiency of fuel. During combustion, the carbon present in the fuel oxidizes to form CO emission. However, if the availability of oxygen is lower, it causes incomplete combustion and hence comes with a higher CO value. Figure 4.7 shows the variation of CO emissions with respect to load for diesel fuel and diesel with biogas mixtures. In dual fuel mode operation, higher CO is observed than diesel. This is because of the lower flame speed of biogas due to the presence of CO_2, reduction of oxygen caused by the induction of biogas and higher specific heat of biogas, as compared to diesel, requires high combustion temperature to burn completely. The above reasons cause some fuel undergoes incomplete combustion leads to high CO emission. It also increases as an increase in biogas flow rate. This is due to the high biogas flow rate further increases CO_2 concentration and decreases the availability of O_2 in the combustion chamber. The average CO emission increment of D + BG@2L/min, D + BG@4L/min, and D + BG@6L/min flow rate from diesel mode was 10.18%, 19.91% and 31.86% respectively. Table 4.6 below shows the numerical value of carbon monoxide emissions.

Table 4.6: Carbon monoxide exhausts emission for all tests.

Load (%)	Carbon Monoxide Emissions (CO) in % Vol.			
	Diesel	D+BG@2L/min	D+BG@4L/min	D+BG@6L/min
10	0.2	0.22	0.24	0.25
20	0.19	0.21	0.23	0.25
30	0.2	0.23	0.24	0.27
40	0.24	0.25	0.28	0.32
50	0.3	0.32	0.35	0.38
60	0.33	0.37	0.39	0.45
70	0.37	0.41	0.45	0.49
80	0.43	0.48	0.52	0.56

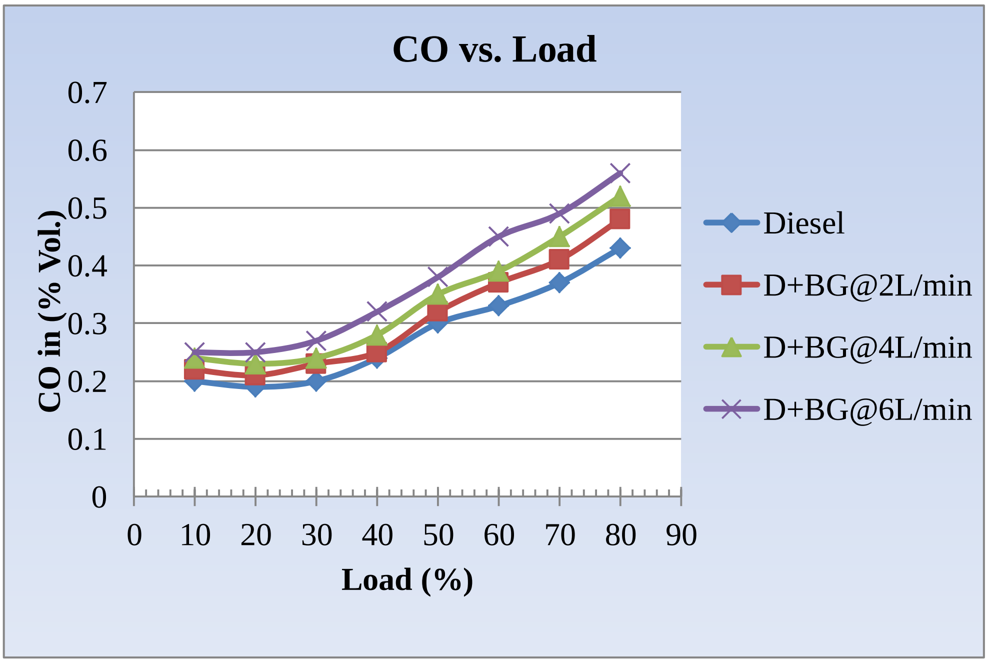


Figure 4.7: Variation of CO emissions with respect to load.

4.5.2 Exhaust Emissions of Carbon Dioxide (%Vol.)

Figure 4.8 shows CO_2 emission with respect to engine loads. The CO_2 emission increases with increases in load for both modes this is because of an increase in combustion rate due to high combustion temperature as compared to low loads. It may also due to high in-cylinder temperature at higher engine load some portion of carbon monoxide converts to

CO_2 and causes an increase in CO_2 emission. On the other hand, in dual fuel mode due to the existence of CO_2 in biogas, CO_2 emission is noticed higher as compared to diesel mode. Additionally, an increase in the flow rate of biogas leads to increase BGES which causes an increase in carbon dioxide. The average CO_2 emission increment of D + BG@2L/min, D + BG@4L/min, and D + BG@6L/min flow rate from diesel mode was 12.8%, 25.98% and 47.33% respectively. Table 4.7 below shows the numerical values of carbon dioxide emissions.

Table 4.7: Carbon dioxide exhausts emissions for all tests.

Load (%)	Carbon Dioxide Emissions (CO_2) in % Vol.			
	Diesel	D+BG@2L/min	D+BG@4L/min	D+BG@6L/min
10	2.67	3.2	3.75	4.75
20	3.2	3.8	4.4	5.5
30	3.7	4.3	4.92	5.9
40	4.2	4.9	5.51	6.4
50	4.7	5.3	5.98	7.04
60	5.3	5.8	6.4	7.6
70	5.9	6.4	7.01	7.9
80	6.5	7.1	7.6	8.2

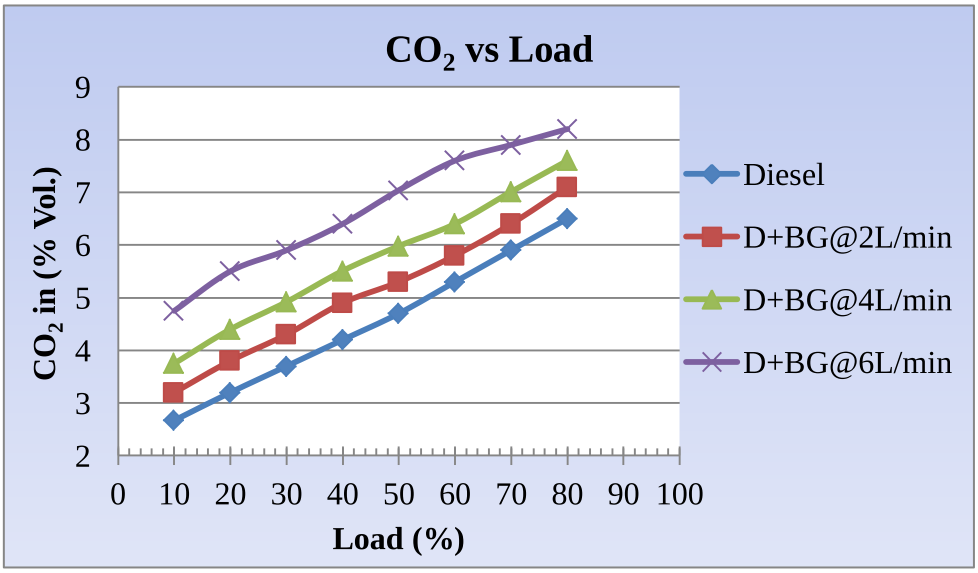


Figure 4.8: Variation of CO_2 emissions with respect to load.

4.5.3 Exhaust Oxygen (%Vol.)

The variation of exhaust oxygen for diesel and biogas-diesel dual fuel mode from 10% to 80% load variation is shown in Figure 4.9. Exhaust oxygen declines as load increases for both the diesel and dual fuels. In the case of pure diesel mode, volumetric efficiency is higher as compared to dual fuel mode for all loads. As a result, as the load increases volumetric efficiency decreases, and incomplete utilization of air increases. Whereas, in the case of dual fuel mode due to the induction of biogas along with the less amount of air causes a reduction in complete utilization of oxygen as compared to diesel fuel. This causes lesser exhaust O_2 in dual fuel mode than diesel and further lesser as biogas flow rate increases. The average exhaust O_2 reduction of D + BG@2L/min, D + BG@4L/min and D + BG@6L/min flow rate from diesel mode were 7.01%, 17.36% and 25.6% respectively. Table 4.8 below shows the numerical values of exhaust oxygen.

Table 4.8: Oxygen exhaust emissions for all tests.

Load (%)	Oxygen Emissions (O_2) in % Vol.			
	Diesel	D+BG@2L/min	D+BG@4L/min	D+BG@6L/min
10	19	18.25	16.87	15.57
20	18.96	18.05	16.26	14.8
30	18.66	17.51	15.63	14.1
40	18.33	16.87	15.08	13.38
50	17.96	16.46	14.58	12.98
60	17.61	16.11	14.12	12.63
70	17.26	15.78	13.69	12.28
80	16.86	15.47	13.28	11.87

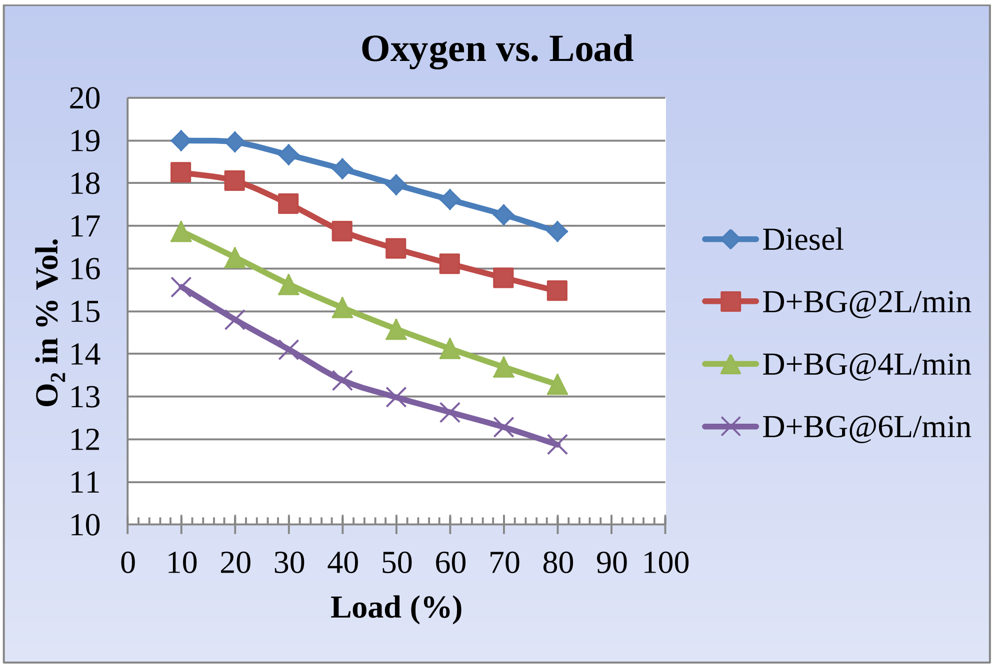


Figure 4.9: Variation of O_2 emissions with respect to load.

4.5.4 Exhaust Emissions of Hydrocarbons (ppm Vol.)

Unburnt hydrocarbon emission for diesel and biogas-diesel fuel with respect to load is shown in Figure 4.10. The unburnt hydrocarbon (UHC) emission in dual fuel operation is higher than diesel, under all test conditions. This is due to poor combustion of the dual-fuel caused by lower flame velocity of biogas due to the presence of CO_2 and reduction of oxygen concentration due to biogas substitution as compared to diesel. In addition to that, higher UHC in dual fuel mode is due to; longer ignition delay of biogas-diesel fuel as a result of the low self-ignition temperature of biogas, increase in BGES which results in lower in-cylinder temperature and carbon dioxide presence in biogas absorbed the heat and decrease the in-cylinder temperature which slows down the hydrocarbon oxidation process. Moreover, the overlapping between intake and exhaust valves for promoting scavenging is also the reason for unburnt HC emission to increase for dual fuel mode. Furthermore unburnt HC increases as the biogas flow rate increases, this is because as the biogas flow rate increases the amount of air enters the combustion chamber decreases which results in a rich mixture as compared to a low biogas flow rate and causes incomplete combustion inside the cylinder. A similar trend was reported by (Jagadish & Gumtapure, 2019). The average unburnt HC emission increment of D + BG@2L/min, D + BG@4L/min, and D + BG@6L/min flow rate from diesel mode were obtained 6.01%, 19.29% and 30.94% respectively. Table 4.9 below shows the numerical values of HC emissions.

Table 4.9: Hydrocarbon exhaust emissions for all tests.

Load (%)	Hydrocarbon Emissions (ppm).			
	Diesel	D+BG@2L/min	D+BG@4L/min	D+BG@6L/min
10	32.74	33.1	36.26	38.76
20	34.84	36.56	41.24	44.34
30	37.09	39.4	46.05	49.76
40	39.56	42.34	48.78	54.86
50	42.42	45.52	51.52	58.18
60	44.84	48.8	54.56	60.56
70	48.4	51.81	57.8	63.41
80	52.6	54.56	60	65

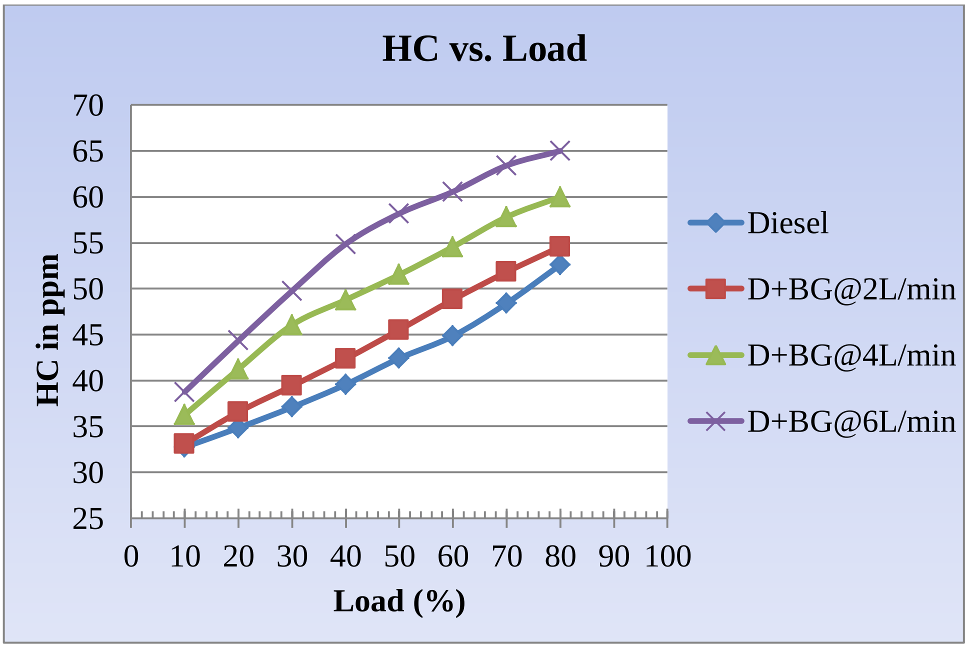


Figure 4.10: Variation of HC emissions with respect to load.

4.5.5 Nitrogen Oxide Exhaust Emissions (ppm Vol.)

The formation of NOx emission mainly depends on the availability of oxygen, higher temperature developed during the combustion, and the residence time for which oxygen-nitrogen reactions occurring to a significant completion level (Bouguessa et al., 2020).

Variations of NOx emission for both diesel and dual-fuel at all loads are shown in Figure 4.11 below. In general, the NOx emission increases as the load increases for both fuels. From the figure, it is observed that NOx emission is lower in the dual fuel mode operation compared to that of diesel for all loads. This is because of an increase in the equivalence ratio for the higher loads and also due to the induction of biogas through the intake manifold creates a reduction in oxygen concentrations of the biogas-air mixture such that the rate of formation of NOx reduces slowly. Besides, CO_2 presence in the biogas reduces the peak cylinder temperature since CO_2 possesses higher specific heat. Moreover, an increase in the biogas energy share leads to lower peak cylinder temperature in the dual fuel mode results in a lower NOx level. On the other hand, at the higher loads due to the enhanced and rapid combustion of biogas the NOx formation levels were found to be higher compared to lower loads.

The average NOx emission reduction of D + BG@2L/min, D + BG@4L/min, and D + BG@6L/min flow rate from diesel mode were obtained 19.91%, 27.33% and 39.16% respectively. Table 4.10 below shows the numerical values of NOx emissions.

Table 4.10: Nitrogen oxides exhaust emissions for all tests.

Load (%)	Nitrogen Oxides Emissions (NOx) in ppm.			
	Diesel	D+BG@2L/min	D+BG@4L/min	D+BG@6L/min
10	68.62	52.6	43.4	32.03
20	107.52	75.4	59.47	43.46
30	157.84	116.67	96.07	75.49
40	215.03	160.13	132.67	109.8
50	317.97	233.3	210.74	180.72
60	420.92	308.8	285.95	249.35
70	498.69	416.34	393.46	333.99
80	576.47	523.85	494.12	411.76

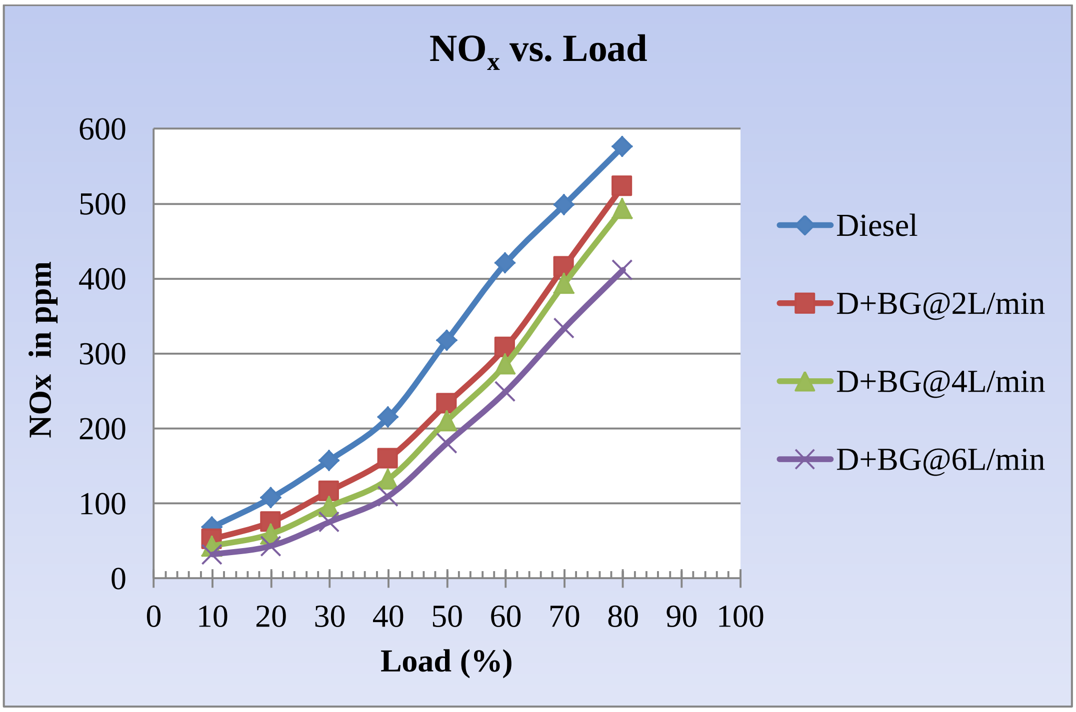


Figure 4.11: Variation of NOx emissions with respect to load.

CHAPTER FIVE

CONCLUSIONS AND RECOMMENDATIONS

5.1 Summary

As it has been introduced in the introduction part, the main objective of this work was to investigate the performance and emission characteristics of diesel engine running by biogas-diesel dual-fuel with varying the engine load and biogas flow rate at a constant engine speed of 1500RPM by using a venturi air-biogas mixer device. Using a venturi gas mixer device for the biogas-diesel dual-fuel engine has advantages for properly mixing air-biogas mixture and thereby due to complete combustion, improves performance and reduction of emission characteristics as compared to directly supplied biogas to the engine without a mixing device. In gas mixer design, the convergent angle and beta ratio were considered for improving mixture quality. The simulation results were selected based on the smallest distance on the downstream of the mixer as it has seen from the venturi throat at which methane distribution is similar.

Experiments were carried out on a diesel engine fueled with firstly diesel fuel for baseline data and secondly biogas-diesel dual fuel for examining the performance and emission parameters as compared to diesel fuel mode at constant engine speed for all operations. The effect of biogas flow rate variation on the performance and emission characteristics of a dual fuel mode at various engine loads was also observed.

Based on the experimental work, the following main findings are obtained:

- ✓ As engine load increases: brake torque, brake power and brake thermal efficiency of dual-fuel mode shown slight reduction and a medium increment of brake-specific fuel consumption was noticed as compared to baseline diesel fuel mode. Also, a decrease in biogas energy share for all flow rates was observed. Whereas emissions; CO, CO_2 and HC showed that little increment and reduction of O_2 and NOx emissions as compared to baseline data.
- ✓ As biogas flow rate increases for all load conditions: little reduction in brake torque and brake thermal efficiency, no significant change in brake power, medium increment of brake specific fuel consumption and higher in value of BGES was noticed. Whereas emissions; it was noticed little increment in CO, CO_2 and HC and little reduction in O_2 and NOx emissions.

5.2 Conclusion

Based on the experiments conducted in the biogas-diesel dual-fuel mode diesel engine under various load condition with different biogas flow rate at constant engine speed, and thereby from the results obtained, the following conclusions are drawn:

As biogas flow rate increases from 2L/min to 6L/min with an increasing of load from 10% to 80%:

- ✓ There is an increase in average biogas energy share of 24.98% to 52.32% and diesel fuel replacement ratio (r) of 13.41% to 23.29%.
- ✓ Relatively reduction in brake torque from 3.65% to 6.68%, brake power from 2.81% to 5.61%, BTE from 15.94% to 23.58% and an increment of BSFC from 11.82% to 20.87%.
- ✓ Increment in emissions like CO by 10.18% to 31.86%, CO_2 by 12.8% to 47.33% and HC by 6.01% to 30.94%.
- ✓ There is a reduction in O_2 and NOx emission from 7.01% to 25.5% and 19.91% to 39.16% respectively.

Generally, among different DF mode operations, due to small percentage of methane in a given biogas, with a biogas flow rate of 2L/min yields relatively same performance and emission characteristics and allows extremely low levels of NOx as compared to diesel fuel operation.

5.3 Recommendation

The following recommendations which are obtained based on the thesis outcome are:

- ✓ It is recommended to use a venturi mixer in the biogas-diesel dual-mode diesel engine for having better mixing of air and biogas.
- ✓ For more accurate results of performance and emission characteristics, the testing engine should be anchored or safely fixed on the ground basement.
- ✓ In the present study, since only 2L/min, 4L/min and 6L/min biogas flow rate was conducted. For comparison and further evaluation, it is recommended to increase the biogas flow rate at the possible level for observing the maximum biogas energy share and diesel fuel replacement ratio.
- ✓ Due to the absence of an exhaust gas analyzer that analyzes soot emission, particulate matter and sulfur dioxide, this work only focused on CO, CO_2, O_2, HC and NOx emissions.

- ✓ Since the biogas used in this work was raw biogas, for having better performance and lower emissions it is recommended to purify it or using purified biogas.
- ✓ Dual-fuel mode operation with 2L/min raw biogas flow rate has relatively similar performance and lower NOx emissions as compared with diesel fuel mode, so it is recommended to use raw biogas up to 2L/min flow rate.

Finally, I recommended the Department of Mechanical Systems and Vehicle Engineering of Adama Science and Technology University to have an alternative fuel performance and emission testing laboratory.

5.4 Future Work

From different aspects as has been mentioned in chapter one under the significance of the study, using renewable alternative fuel biogas for the IC engine has several advantages. From this point of view, several things can be made and improved in the future in which this study couldn't address. So the following areas need further improvement for future studies.

i. **Biogas purification**: for utilizing biogas as a fuel for internal combustion engines it should be purified. Hence from different components of biogas, only methane is combustible. Since raw biogas contains less amount of methane and a high amount of un-combustible gas like carbon dioxide which declines the in-cylinder temperature and thereby incomplete combustion to occur. Whereas purified biogas contains more than 90% methane content So that utilization of purified biogas in CI engine results in better performance and emission characteristics as compared to conventional diesel fuel.

ii. **Venturi mixer optimization**: even if purified biogas supplied to the CI engine, poorly mixed air-biogas mixture badly affects the performance and emission characteristics. So gas mixers should be employed for utilizing gas fuel in IC engines. This work mainly focused on numerical analysis for designing and then manufacturing a venturi mixer. But for having better mixing quality and thereby homogeneous air-biogas mixture, all the parameters of the venturi air-biogas mixer should optimize by using ANSYS fluent. So, the prediction of biogas flow analyses with the aid of using software such as Computational Fluid Dynamics (CFD) simulation is future research work.

REFERENCES

Abo-Serie, E., Ozgur, M., & Altinsik, K. (2015). Computational analysis of methane-air venturi mixer for optimum design. *unpublished*, 1-9. https://www.researchgate.net/publication/281684556.

Aklouche, Z., Loubar, K., Bentebbiche, A., Awad, S., & Tazerout, M. (2017). Experimental investigation of the equivalence ratio influence on combustion, performance and exhaust emissions of a dual fuel diesel engine operating on synthetic biogas fuel. *Energy Conversion and Management*, 291–299. https://dx.doi.org/10.1016/j.enconman.2017.09.050.

Ali, M., & Salih, M. (2013). Factors Affecting Performance of Dual Fuel Compression Ignition Engines. *Applied Mechanics and Materials*, 217-222. https://dx.doi.org/10.4028/www.scientific.net/AMM.388.217.

Ambarita, H. (2017). Performance and emission characteristics of a small diesel engine run in dual-fuel (diesel-biogas) mode. *Case Studies in Thermal Engineering*, 179–191. https://dx.doi.org/10.1016/j.csite.2017.06.003.

Ambarita, H. (2019). Experimental study on diesel engine coupled with a catalytic converter run on dual-fuel mode using biogas produced from agricultural waste. *IOP Publishing.* Medan 20155, Indonesia. https://dx.doi.org/10.1088/1757-899X/523/1/012064.

Arali, M., Anil, R., & Gawade, S. (2016). Design and Development of Intake Device for Biogas operated 4-Stroke SI Engine. *Energy*, 1-7. https://www.researchgate.net/publication/322634895.

Ayade, M., & Latey, A. (2016). performance and emission characteristics of biogas-petrol dual fuel in SI engine. *International Journal of Mechanical Engineering and Technology (IJMET)*, *7*(2), 45-54. http://www.iaeme.com/IJMET_07_02_006.pdf.

Barik, D., & Murugan, S. (2014). Investigation of combustion performance and emission characteristics of a DI diesel engine fueled with biogas-diesel in dual fuel mode. *Energy*, 760 – 771. https://dx.doi.org/10.1016/j.energy.2014.05.106.

Barik, D., & Murugan, S. (2016). Experimental investigation on the behavior of a DI diesel engine fueled with raw biogasediesel dual fuel at different injection timing. *Journal of the Energy Institute*, 373-388. https://dx.doi.org/10.1016/j.joei.2015.03.002.

Bharathiraja, B., Sudharsana, T., Jayamuthunagai, J., Praveenkumar, R., Chozhavendhan, S., & Iyyappan, J. (2018). Biogas production – A review on composition, fuel properties, feed stock and principles of anaerobic digestion. *Renewable and Sustainable Energy Reviews*, 570-582. https://ideas.repec.org/a/eee/rensus/v90y2018icp570-582.html.

Bhuimbar, S., & Kumarappa, S. (2015). Engine Performance and Emission Characteristics of a Single Cylinder Four Stroke CI Engine on Dual Fuel Mode by Using Compressed Biogas as a Fuel. *International Journal for Research in Applied Science & Engineering Technology*, *3*(9), 376-382. https://www.ijraset.com/fileserve.php?FID=3277.

Bora, B., Debnath, B., Gupta, N., Saha, U., & Sahoo, N. (2013). Investigation on the flow behaviour of a venturi type gas mixer designed for dual fuel diesel engines. *International Journal of Emerging Technology and Advanced Engineering*, 202-209. http://citeseerx.ist.psu.edu/viewdoc/summary?doi=10.1.1.414.2833.

Bora, J., & Saha, K. (2016). Optimisation of injection timing and compression ratio of a raw biogas powered dual fuel diesel engine. *Applied Thermal Engineering*, 111–121. https://dx.doi.org/10.1016/j.applthermaleng.2015.08.111.

Bora, J., & Saha, K. (June 25-27, 2014). On the attainment of optimum injection timing of pilot fuel in a dual fuel diesel engine run on biogas. *Proceedings of the ASME 2014 12th Biennial Conference on Engineering Systems Design and Analysis* (pp. 1-10). Copenhagen, Denmark: American Society of Mechanical Engineers (ASME). https://proceedings.asmedigitalcollection.asme.org/.

Bora, J., Saha, K., Chatterjee, S., & Veer, V. (2014). Effect of compression ratio on performance, combustion and emission characteristics of a dual fuel diesel engine run on raw biogas. *Energy Conversion and Management*, 1000–1009. https://dx.doi.org/10.1016/j.enconman.2014.07.080.

Bouguessa, R., Tarabet, L., Loubar, K., Belmrabet, T., & Tazerout, M. (2020). Experimental investigation on biogas enrichment with hydrogen for improving the combustion in diesel engine operating under dual fuel mode. *International Journal of Hydrogen Energy*, 1-12. https://doi.org/10.1016/j.ijhydene.2020.01.003.

Carpintero Durango, J., Fabregas Villegas, J., Santamaria De La Cruz, H., Perez, S., & Valencia Ochoa, G. (2019). Computational validation of the design of an air-biogas mixer for a turboalimented diesel engine. *International Journal of Engineering &Technology*, *8*(3), 337-342. https://doi.org/10.14419/ijet.v8i3.29538.

Chandekar, C., & Debnath, K. (2018). Computational investigation of air-biogas mixing device for different biogas substitutions and engine load variations. *Renewable Energy*, 811- 824. https://doi.org/10.1016/j.renene.2018.05.003.

Chintala, V., & Subramanian, K. (2013). A computational fluid dynamics study for optimization of gas injector orientation for performance improvement of a dual-fuel diesel engine. *Energy*, 709–721. https://doi.org/10.1016/j.energy.2013.06.009.

Dahake, M., Patil, S., & Patil, S. (2016). Performance and Emission Improvement through Optimization of Venturi Type Gas Mixer for CNG Engines. . *International research and Journal of Engineering Technology*, 994–999.

Danardono, D., Kim, K., Lee, S., & Lee, J. (2011). Optimization the design of venturi gas mixer for syngas engine using three-dimensional CFD modeling. *Journal of Mechanical Science and Technology 25 (9)*, 2285-2296. https://doi.org/10.1007/s12206-011-0612-8.

Das, S., Kashyap, D., Kalita, P., Kulkarni, V., & Itaya, Y. (2020). Clean gaseous fuel application in diesel engine: A sustainable option for rural electrification in India. *Renewable and Sustainable Energy Reviews*, 1-28. https://doi.org/10.1016/j.rser.2019.109485.

Defaria, N., Bueno, M., Ayad, E., & Belchior, R. (2017). Thermodynamic simulation model for predicting the performance of spark ignition engines using biogas as fuel. *Energy Conversion and Management*, 1096–1108.

Deheri, C., Acharya, K., Thatoi, N., & Mohanty, P. (2020). A review on performance of biogas and hydrogen on diesel engine in dual fuel mode. *Fuel*, 1-17. https://doi.org/10.1016/j.fuel.2019.116337.

Feroskhan, M., Ismail, S., Kumar, A., Kumar, V., & Aftab, K. (2016). Investigation of the effects of biogas flow rate and cerium oxide addition on the performance of a dual fuel CI engine. *Biofuels*, 1-10. http://dx.doi.org/10.1080/17597269.2016.1215072.

Ganesan, V. (2016). *Internal combustion engines. fourth ed. ; 2016.* New Delhi: McGraw Hill Education (India) Private Limited.

Gnanamoorthi, V., & Mohandoss, M. (2018). Combustion, performance and emission analysis of dual fuel engine using tsrb biogas. *Energy Sources, Part A: Recovery, Utilization, and Environmental Effects*, 1-13. https://doi.org/10.1080/15567036.2018.1550538.

Grabel, W. (2001). Engineering fluid mechanics. New York, NY 10001, U nited States of America: Taylor & Francis.

Gurung, D., Rajvanshi, A., & Lalhriatpuia, S. (2018). Performance Analysis of Combined Biogas-Diesel Run Dual-Fuel Engine. *Advances in Smart Grid and Renewable Energy*, 559-566. https://doi.org/10.1007/978-981-10-4286-7_55.

Heywood, J. B. (1988). *Internal Combustion Engine Fundamentals.* New York : mcgraw-Hill.

Hogstrom, R., Vesala, H., & Heinonen, M. (2017). Particulate content of biogas. *18th International Congress of Metrology*, 1-3. https://doi.org/10.1051/metrology/201708002.

Idriss Husen, & Mekonnen Maschal. (2018). Methane Gas Emission and its Management Practices from Solid Waste Stream, Case Study: Addis Ababa and its Surrounding Oromia Special Zone Towns. *Environment Pollution and Climate Change*, *2*(3), 1-6. https://doi.org/10.4172/2573-458X.1000160.

Jagadish, C., & Gumtapure, V. (2019). Experimental investigation of methane-enriched biogas in a single cylinder diesel engine by the dual fuel mode. *Energy Sources, Part A: Recovery, Utilization and Environmental Effects*, 1-14. https://doi.org/10.1080/15567036.2019.1647314.

John K. Vennard. (1982). *Elementary Fluid Mechanics.* Robert L.Street: John Wiley & Sons.

Kadirgama, K., Noor, M., Rahim, A., Devarajan, R., Rejab, M., & Zuki, N. (March 8-10, 2008). Design and Simulate Mixing of Compressed Natural Gas with Air in a mixing device. *Malaysian Technical Universities Conference on Engineering and Technology.* Putra Palace, Perlis, Malaysia.

Kamp, M., & Forn, B. (2016). Ethiopia's emerging domestic biogas sector: Current status, bottlenecks and drivers. *Renewable and Sustainable Energy Reviews*, 475-488. https://doi.org/10.1016/j.rser.2016.01.068.

Karim, A. (2015). *Dual-fuel diesel engine. first ed.* London: CRC press. https://www.routledge.com/Dual-Fuel-Diesel-Engines/Karim/p/book/9781498703086.

Karim, Z. A., & Sulaiman, S. A. (2018). *Alternative Fuels for Compression Ignition Engines.* Singapore: Springer Nature. https://www.springer.com/gp/book/9789811077531.

Khoshnevisan, B., Tsapekos, P., Alfaro, N., Díaz, I., Fdz-Polanco, M., Rafiee, S., et al. (2017). A review on prospects and challenges of biological H2S removal from biogas with focus on biotrickling filtration and microaerobic desulfurization. *Biofuel Research Journal*, 741–750. https://doi.org/10.18331/BRJ2017.4.4.6.

Kumar, D. (2012). *Studies on improvement of intake manifold for compressed natural gas engine.* Patiala-147001 India: Unpublished Master's Thesis Submitted to Thapar University, 21 pp.

Mahmood, A., Adam, M., Sahari, B., & Masuri, U. (2016). Investigation on the airgascharacteristics of air-gas mixer designed for Bi-Engines. *International Journals of Applied Engineering Research*, 7786-7794. https://doi.org/10.3390/en10091373.

Makareviciene, V., Sendzikiene, E., Pukalskas, S., Rimkus, A., & Vegneris, R. (2013). Performance and emission characteristics of biogas used in diesel engine operation. *Energy Conversion and Management*, 224-233. http://dx.doi.org/10.1016/j.enconman.2013.06.012.

Mao, C., Feng, Y., Wang, X., & Ren, G. (2015). Review on research achievements of biogas from anaerobic digestion. *Renewable and Sustainable Energy*, 540–555. https://doi.org/10.1016/j.rser.2015.02.032.

Muhssen, S., Masuri, U., Sahari, B., & Hairuddin, A. (2019). Computational Fluid Dynamics Investigation of Air-Gas Pre Mixing Controller Mixer Designed for CNG-Diesel Dual-Fuel Engines. *CFD Letters*, 47-62. www.akademiabaru.com/cfdl.html.

Nasir, M., Ghazi, M., & Omar, R. (2012). Anaerobic digestion technology in livestock manure treatment for biogas production: a review. *Engineering in Life Sciences*, 258-269. https://doi.org/10.1002/elsc.201100150.

Neeranjan Singh, H., & Layek, A. (December, 2018). Experimental Study of the Effect of Compression Ratio on the Characteristics of a Biogas Fueled Dual Fuel Compression Ignition Engine. *International Journal of Renewable Energy Research, 8*(4), 2075-2084.

Palaniswamy, D., Ramesh, G., Sivasankaran, S., & Sooryaprakash, K. (2016). CFD Analysis for homogenous effect of biogas and air in the intake manifold of dual fuel CI engine. *International Journal of Advanced Engineering Technology*, 265-268. https://www.technicaljournalsonline.com/ijeat/202016/20167248.pdf.

Pictak, A., Imioaek, M., Gmieja, M., & Wierzbicki, S. (2011). Choice of a pilot dose in dual fuel selfignition engine of a generator depending on its load. *Journal of KONES Powertrain and Transport*, 129-136. http://yadda.icm.edu.pl/yadda/element.pdf.

Prajapati, K., Randa, R., & Parmar, N. (2015). Experimental Study on Utilization of Biogas in IC Engine. *International Journal of Engineering Sciences & Research Technology*, 827-835. http://www.ijesrt.com/issues/August-2015/121.pdf.

Pulkrabek, W. W. (1996). *Engineering Fundamentals of the Internal Combustion Engine.* Upper saddle river, New Jersey 07458: Prentice Hall.

Rahman, A., & Ramesh, A. (2019). Experimental studies on the effects of methane fraction and injection strategies in a biogas diesel common rail dual fuel engine. *Fuel*, 147-165. https://doi.org/10.1016/j.fuel.2018.08.091.

Ramasamy, D., Mahendran, S., Kadirgama, K., & Noor, M. (6-27 MAY 2010). Design of compressed natural gas mixer using computational fluid dynamics. *National Conference in Mechanical Engineering Research and Postgraduate Students*, (pp. 614-620). FKM Conference Hall, UMP, Kuantan, Pahang, Malaysia.

Ray, S., Mohanty, K., & Mohanty, C. (2013). A Study on Application of Biogas as fuel in Compression Ignition Engines. *International Journal of Innovations in Engineering and Technology*, 239-245. http://ijiet.com/wp-content/uploads/2013/10/37.pdf.

Romanczyk, M. (2017). Influence of gas inlet angle on the mixing process in a Venturi mixer. *ITM Web of Conferences* (pp. 1-7). Czestochowa, Poland: EDP Sciences. https://doi.org/10.1051/itmconf/20171507005.

Rosha, P., Dhir, A., & Mohapatra, S. (2018). Influence of gaseous fuel induction on the various engine characteristics of a dual fuel compression ignition engine: a review. *Renewable and Sustainable Energy Reviews*, 3333-3349. https://dx.doi.org/10.1016/j.rser.2017.10.055.

Ryckebosch, E., Drouillon, M., & Vervaeren, H. (2011). Techniques for transformation of biogas to biomethane. *Biomass Bioenergy*, 1633–1645. https://doi.org/10.1016/j.biombioe.2011.02.033.

Sahoo, B. (2011). *Clean development mechanism potential of compression ignition diesel engines using gaseous fuels in dual fuel mode.* IIT Guwahati, Guwahati, India. http://gyan.iitg.ernet.in/handle/123456789/118.

Sahoo, B., Sahoo, N., & Saha, U. (2009). Effect of engine parameters and type of gaseous fuel on the performance of dual-fuel gas diesel engines: A critical review. *Renewable and Sustainable Energy Reviews*, 1151–1184. http://dx.doi.org/10.1016/j.rser.2008.08.003.

Sai Kanth, S. (2015). Theoretical Investigation: Biogas as an Alternate Fuel for IC Engines. *International Journal of Engineering Research and Technology*, *8*, 113-124. http://www.irphouse.com/ijert/ijertv8n2_05.pdf.

Salve, D., Kolekar, H., & Jadhav, M. (2016). Experimental Evaluation of Dual Fuel CI Engine Using Synthesized Biogas. *International Engineering Research Journal (IERJ)*, 1223-1229. http://www.ierjournal.org/pupload/mit/EE1-1.pdf.

Sandalc, T., Isın, O., Galata, S., Karagoz, Y., & Guler, I. (2019). Effect of hythane enrichment on performance, emission and combustion characteristics of an ci engine. *Hydrogen Energy* , 3208–20. https://doi.org/10.1016/j.ijhydene.2018.12.069.

Sarkar, A., & K.Saha, U. (2018). Role of global fuel-air equivalence ratio and preheating on the behavior of a biogas driven dual fuel diesel engine. *Fuel,* 743–754. https://doi.org/10.1016/j.fuel.2018.06.016.

Satapathy, K., Murugan, S., & Barik, D. (2015). Combustion analysis of the diesel–biogas dual fuel direct injection diesel engine. *International Journal of Ambient Energy*, 1-8. http://dx.doi.org/10.1080/01430750.2015.1086681.

Shaikh, M., & Shukla, A. H. (2016). Optimization of Natural Gas Mixture Design Using Computational Method for HCCI Engine by Improving Swirl Effect. *International Journal for Scientific Research & Development, Vol. 4, Issue 03*, 1085-1087. http://inpressco.com/wp-content/uploads/2016/08/Paper31509-1514.pdf.

Singh, A., Avinash Agarwal, R., Kumar Agarwal, A., Dhar, A., & Kumar Shukla, M. (2018). *Prospects of Alternative Transportation Fuels.* Uttar Pradesh, India: Springer Nature. https://www.amazon.com/Prospects-Alternative-Transportation-Environment-Sustainability-ebook/dp/B077GB4G5P.

Sivalingam, M., & Barik, D. (2013). Performance and Emission Characteristics of a Biogas Fueled DI Diesel Engine. *SAE International*, 1-11. https://dx.doi.org/10.4271/2013-01-2507.

Solomon, S., Sindibu, T., & Ermias, D. (2018). Biogas and Bio-fertilizer Production Potential of Abattoir Waste as Means of Sustainable Waste Management Option in Hawassa City, Southern Ethiopia. *Journal of Applied Sciences and Environmental Management, 22*(4), 553-559. https://doi.org/10.4314/jasem.v22i4.21.

Sorathia, S., Zala, H., Suthar, L., & Pipaliya, K. (2014). performance of biogas premixed charge constant speed diesel engines used in an agricultural field. *International Journal of Advanced Engineering Technology*, 33-40. https://www.technicaljournalsonline.com/ijeat/208.pdf.

Turco, M., Ausiello, A., & Micoli, L. (2016). *Treatment of Biogas for Feeding High Temperature Fuel Cells: Removal of Harmful Compounds by Adsorption Processes.* Naples, Italy: Springer In. https://link.springer.com/content/pdf/10.1007%2F978-3-319-03215-3.pdf.

Vanga, B., & Minhtu, B. (2019). Soot Emission Analysis in Combustion of Bio-gas Diesel Dual Fuel Engine. *Environmental Science and Sustainable Development*, 67-79. https://creativecommons.org/licenses/by/4.0/.

Verma, S., Das, L., Kaushik, S., & Tyagi, S. (2018). An experimental investigation of exergetic performance and emission characteristics of hydrogen supplemented biogas-diesel dual fuel engine. *International Journal of Hydrogen Energy*, 2452-2468. https://doi.org/10.1016/j.ijhydene.2017.12.032 .

Verma, S., M. Das, L., C. Kaushik, S., & S. Bhatti, S. (2019). The effects of compression ratio and EGR on the performance and emission characteristics of diesel-biogas dual fuel engine. *Applied Thermal Engineering*, 1090–1103. https://doi.org/10.1016/j.applthermaleng.2019.01.080.

Vitazek, Klucik, J., Uhrinova, D., Mikulova, Z., & Mojzis, M. (2016). Thermodynamics of combustion gases from biogas. *Research in Agricultural Engineering*, 8-13. https://doi.org/10.17221/34/2016-RAE.

Von Mitzlaff, K. (1988). *Engines for Biogas.* D-6236 Eschborn, Germany: Deutsches Zentrum fwicklungstechnologien - GATE.

Wei, L., & Geng, P. (2016). A review on natural gas/diesel dual fuel combustion, emissions and performance. *Fuel Process Technology*, 264–78. https://doi.org/10.1016/j.fuproc.2015.09.018.

Wieczorek, N., Kucuker, M., & Kuchta, K. (2015). Microalgae-bacteria flocs (MaB-Flocs) as a substrate for fermentative biogas production. *Bioresource Technology*, 130–136. https://doi.org/10.1016/j.biortech.2015.06.104.

Wierzbicki, S. (2012). Biogas as a fuel for diesel engines. *Journal of KONES Powertrain and Transport*, 477-482. http://yadda.icm.edu.pl/201920wierzbicki.pdf.

Wilkie, C. (2019, December 16). (University of Florida; Bioenergy School: Soil and Water Sciences Department) Retrieved from https://biogas.ifas.ufl.edu/feedstocks.asp

Yonas, B., Olav, L., Ajebu, N., Ayana, A., & Aune, B. (2019). Methane Emissions from Ruminant Livestock in Ethiopia: Promising Forage Species to Reduce CH4 Emissions. *agriculture*, 1-16. https://doi.org/10.3390/agriculture9060130.

Yoon, H., & Lee, C. (2011). Experimental investigation on the combustion and exhaust emission characteristics of biogas–biodiesel dual-fuel combustion in a CI engine. *Fuel Processing Technology*, 992-1000. https://doi.org/10.1016/j.fuproc.2010.12.021.

APPENDICES

Appendix A: Drawings

Appendix B: Biogas energy share and diesel fuel replacement ratio

Table B.1: Biogas energy share and diesel replacement ratio (r)

Mode of operation	Load (%)	$\dot{m}_{dd}$ (kg/h)	$\dot{m}_{bg}$ (kg/h)	r (%)	BGES (%)
Dual-mode Diesel + Biogas at 2L/min flow rate	0	0.0995	0.1344	19.56	39.6
	20	0.146	0.1344	19.16	30.88
	40	0.195	0.1344	14.53	25.07
	60	0.251	0.1344	8.26	20.63
	80	0.328	0.1344	7.61	16.59
Dual-mode Diesel + Biogas at 4L/min flow rate	0	0.0897	0.2688	27.48	59.26
	20	0.137	0.2688	24.14	48.78
	40	0.185	0.2688	18.91	41.36
	60	0.235	0.2688	14.11	35.70
	80	0.314	0.2688	11.55	29.35
Dual-mode Diesel + Biogas at 6L/min flow rate	0	0.084	0.4032	32.42	70.08
	20	0.129	0.4032	28.46	60.25
	40	0.175	0.4032	23.47	52.86
	60	0.215	0.4032	21.49	47.69
	80	0.306	0.4032	13.81	39.02

Appendix C: Material Specification

i. Automobile Exhaust Gas Analyzer SV-50

- ✓ Environment conditions: Temperature range from -5~40°C, Humidity ≤95% and atmosphere pressure from 670~10^6kpa
- ✓ Power supply: AC220V ±10%; 50Hz ±1Hz
- ✓ Sampling mode: tail gas directly sampling with a sampling tube length of 5m and length of sampling head is 900mm.
- ✓ Oil temperature: inserting the head of a thermo scope into the staff gauge of engine lubricant, the length of which is the same as that of oil staff gauge, and the length of lead is 5m.
- ✓ Preheating time: 10 minutes and Oil temperature measuring range: 0~120°C.

Table C.1: Automobile Exhaust Gas Analyzer SV-50 measuring range

species	Measurement range	Resolution	Allowed error	Relative error
HC	0 - 1000 ppm	1 ppm	±12ppm	±5%
CO	0 - 10.0 in % Vol.	0.01%	±0.06%	±5%
CO_2	0 - 20.0 in % Vol.	0.1%	±0.5%	±5%
O_2	0 - 25.0 % Vol.	0.01%	±0.1%	±5%
NOx	0 – 800 ppm	1 ppm	±10ppm	±5

Figure C.1. Rear view of automobile exhaust gas analyzer SV-50

ii. Rotameter

It is a device that measures the volumetric flow rates of a fluid (gases) in a closed tube. It belongs to a class of meters called variable area meters, which measure flow rate by allowing the cross-sectional area the fluid (gas) travels through to vary, causing a measurable effect. The measuring range of the employed Rotameter is from 0 to 50 L/min and found in Defense Engineering College, Debrezeit, Ethiopia.

Figure C.2. Rotameter

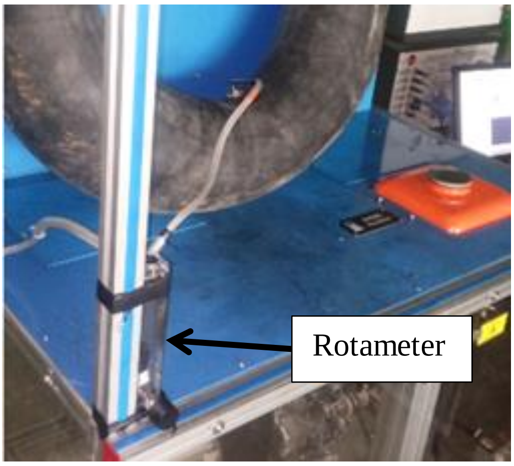
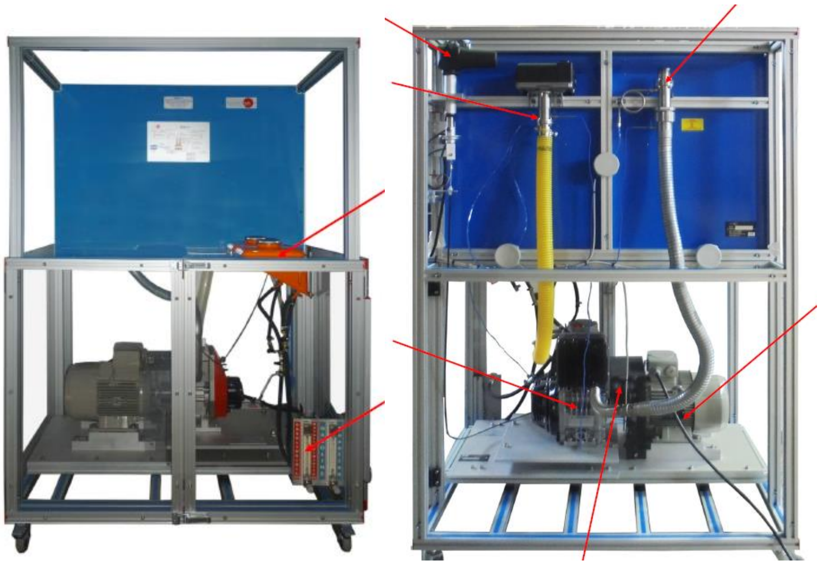


Figure C.3. Position of Rotameter

iii. TBMC3-02 Test Bench Diesel Engine



a. Front view b. Rear view

Figure C.4. TBMC3-02 Test Bench Diesel Engine

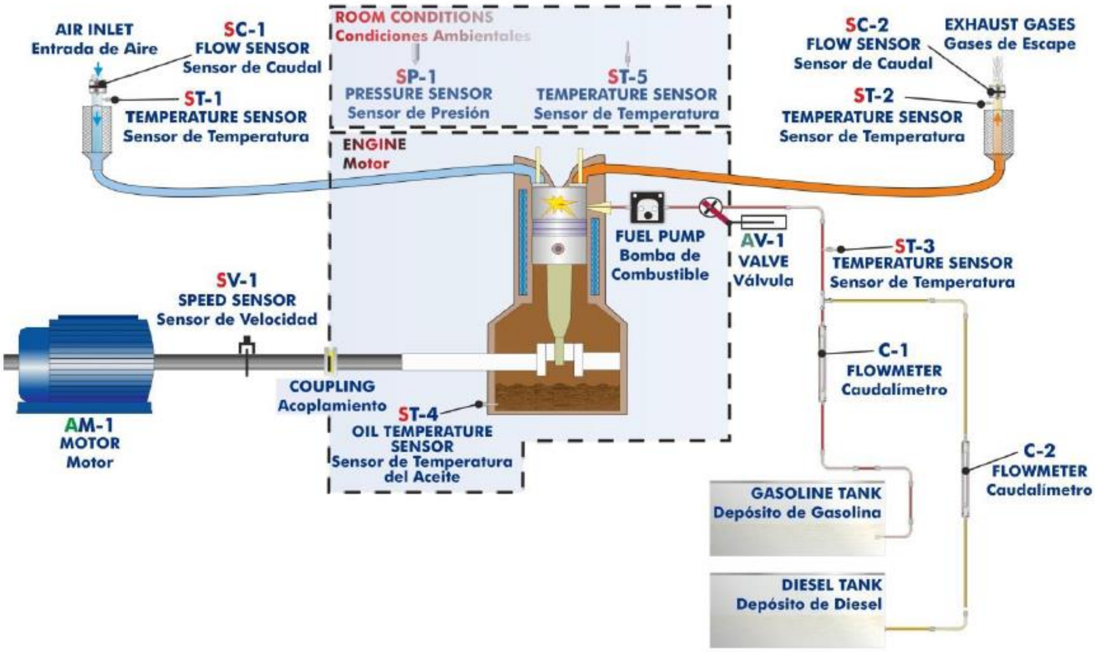


Figure C.5. Description of the process

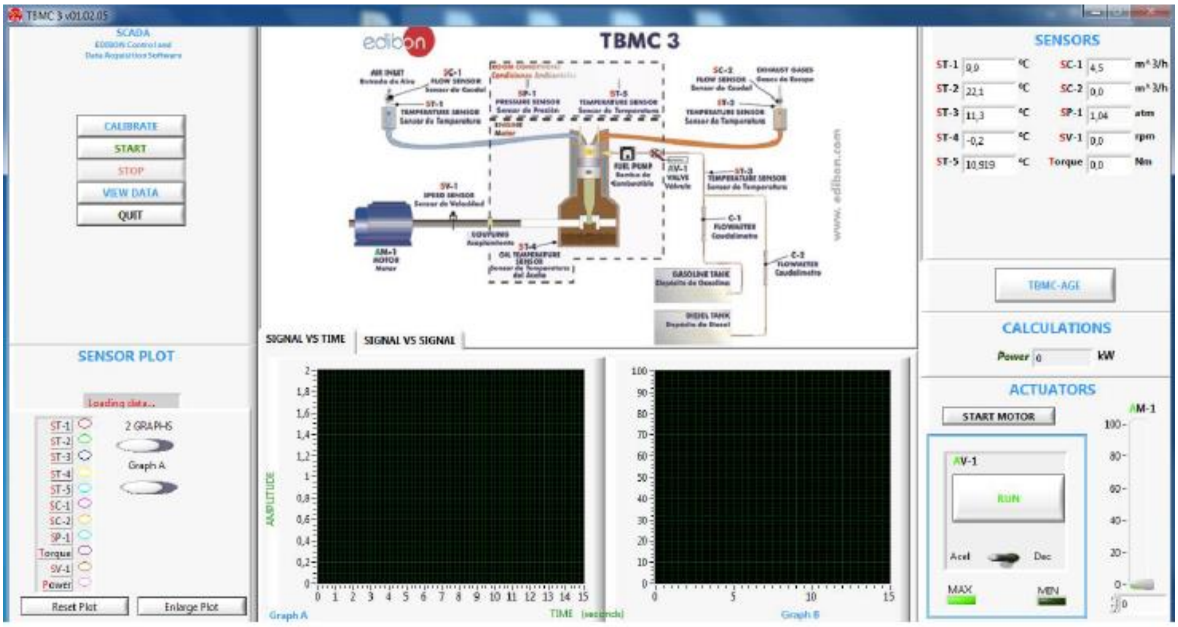


Figure C.6. TBMC-3 Software unit

Appendix D: Experimental Uncertainties

An uncertainty or error analysis is necessary to establish the bounds on the accuracy of the estimated parameters. Evaluations of some unknown uncertainties from known physical quantities were obtained using the following general equation:

$$\frac{U_Y}{Y} = \left[\sum_{i=1}^{n}\left(\frac{1}{y}\frac{\partial Y}{\partial xi}U_{xi}\right)^2\right]^{\frac{1}{2}}$$

In the equation cited, Y is the physical parameter that is dependent on the parameters, xi. The symbol U_Y denotes uncertainty in Y.

Uncertainties in Rotameter

Uncertainty in Rotameter is obtained by computing its Resolution/Range.

$$Rotameter\ uncertainity = (\pm 0.1\ L/min)/(50\mathrm{L/min}) \times 100 = 0.2$$

Uncertainty in Emission Constituents

For each emission parameters the uncertainties measurements error is established in Appendix C Table C.1. Uncertainty in emission constituents are obtained by computing its Resolution/Range.

$$\Delta CO/CO = (\mathrm{Resolution/range})_{CO} \times 100$$
$$\Delta CO_2/CO_2 = (\mathrm{Resolution/range})_{CO_2} \times 100$$
$$\Delta HC/HC = (\mathrm{Resolution/range})_{HC} \times 100$$
$$\Delta NO_x/NO_x = (\mathrm{Resolution/range})_{NO_x} \times 100$$

Table D.1: Rotameter measurement range.

Device	Measuring ranges	Resolution	accuracy	Relative errors (%)
Rotameter	0 – 50L/min	±0.1L/min	±5L/min	±1

Table D.2: Overall measurement uncertainty for emission parameters.

Emission parameters	Diesel mode uncertainty (%)	Biogas-diesel mode uncertainty (%)
CO emission	±0.1	±0.1
CO_2 emission	±0.5	±0.5
HC emission	±0.1	±0.1
O_2	±0.04	±0.04
NOx emission	±0.125	±0.125

Appendix E: Pictures

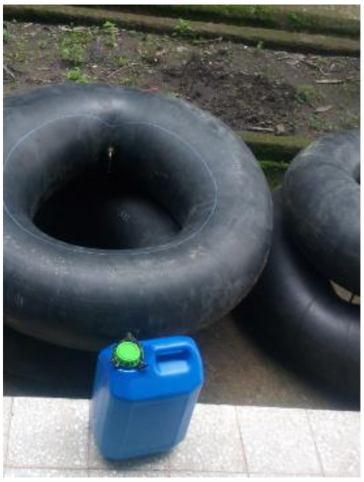



Figure E.1. During compressed biogas in a tire tube by using ordinary compressor

a. Adjusting Rotameter valve

b. Biogas goes empty

Figure E.2. During the engine running in dual fuel mode


Figure E.3. During recording emissions

Appendix F: Letters

Letter of recommendation as shown below, which explains experimental work performed in the automotive laboratory, mechanical engineering department from Addis Ababa Science and Technology University.

አዲስ አበባ ሳይንስና ቴክኖሎጂ ዩኒቨርሲቲ
ኤሌክትሪካል እና ሜካኒካል ምህድስና ኮሌጅ
መካኒካል ምህንድስና ትምህርት ክፍል

Addis Ababa Science and Technology University
College of Electrical and Mechanical Engineering
Department of Mechanical Engineering

ቁጥር: [illegible]/2012
ቀን 11/12/2012

ለአዳማ ሳይንስና ቴክኖሎጂ ዩኒቨርሲቲ
<u>አዳማ</u>

<u>ጉዳዩ፡ የላብራቶሪ አገልግሎት ያገኙ መሆኑን ስለማሳወቅ</u>

ተማሪ **መልካሙ ገነት** ለጥናት እና ምርምር የሚሆን የናፍጣ እና ባዮጋዝ ቅልቅል ናሙና የቤተሙከራ አገልግሎት እን ዲያገኙ ወደ አዲስ አበባ ሳይንስና ቴክኖሎጂ ዩኒቨርሲቲ **በቀን 21/ሀምሌ/2012**፤ ቁጥር፡ **7/2-1-3/7m80/0374/12** በተጻፈ በደብዳቤ ትብብር እንዲደረግላቸው በጠየቃችሁን መሰረት ተማሪው በመካኒካል ምህንድስና ትምህርት ክፍል ስር በሚገኝ ላብራቶሪ Engine dynamometer performance test እና exhaust gas analysis test አገልግሎት ያገኙ መሆኑን እናሳውቃለን።

ከሰላምታ ጋር

ኤሊያስ ገብረሚካኤል ሀብቴ
የሜካኒካል ምህንድስና
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