Experimental Investigation on the Performance and Exhaust Emission of Si Engine Using E15% Gasoline Dual Fuel Mode with LPG

Assefa Ambachew Tilahun

A Thesis Submitted to Department of Mechanical Engineering

School of Mechanical, Chemical & Materials Engineering.

Presented in Partial Fulfillment of the Requirement for the Degree of Master’s in Automotive Engineering

Office of Graduate Studies

Adama Science and Technology University

January, 2024

Adama, Ethiopia

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DECLARATION

I hereby declare that this Master Thesis entitled “Experimental investigation on the performance and exhaust emissions of SI engine using E15% gasoline dual fuel mode with LPG” is my original work. That is, it has not been submitted for the award of any academic degree, diploma or certificate in any other university. All sources of materials that are used for this thesis have been duly acknowledged through citation.

Assefa Ambachew Tilahun ____________ ____________

Name of student

RECOMMENDATION

I the advisor(s) of this thesis, hereby certify that I have read the revised version of the thesis entitled "Experimental investigation on the performance and exhaust emissions of SI engine using E15% gasoline dual fuel mode with LPG" prepared under my guidance by Assefa

Ambachew Tilahun submitted in partial fulfillment of the requirements for the degree of Mater's of Science in automotive engineering. Therefore, I recommend the submission of revised version of the thesis to the department following the applicable procedures.

Alemayehu Wakjira (Ph.D.)	________________	______________
Major Advisor	Signature	Date

APPROVAL PAGE

I the advisors of the thesis entitled "Experimental Investigation on The Performance and Exhaust Emission of Si Engine Using E15% Gasoline Dual Fuel Mode with LPG" and developed by Assefa Ambachew Tilahun, hereby certify that the recommendation and suggestions made by the board of examiners are appropriately incorporated into the final version of the thesis.

Alemayehu Wakjira (Ph.D.)	__________________	_______________
Major Advisor	Signature	Date

APPROVAL OF BOARD OF EXAMINERS

We, the undersigned, members of the Board of examiners of the thesis by Assefa Ambachew Tilahun have read and evaluated the thesis entitled "Experimental Investigation on The Performance and Exhaust Emission of Si Engine Using E15% Gasoline Dual Fuel Mode With LPG" and examined the candidate during the open defense. This is, therefore, to certify that this thesis is accepted for partial fulfillment of the requirement of the degree of Master of Science in Mechanical, Chemical & Materials Engineering (Specialization in Automotive Engineering).

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Table of Contents

LIST OF FIGURE

LIST OF TABLES

NOMENCLATURE

BSFC	Brake specific fuel consumption
CFD	Computational Fluid Dynamics
CO	Carbon monoxide
CO2	Carbon dioxide
HC	Hydrocarbon
IC	Internal Combustion
GDI	Gasoline Direct Injection
LPG	Liquefied Petroleum Gas
LPI	LPG Port Injection
LPM	Litter per minute
NG	Natural gas
NO2	Nitrogen Oxide
SI	Spark Ignition

ABSTRACT

Significant changes in the global ecosystem, combined with the possibility of an oil supply shortage, have sparked a flurry of curiosity about the creation of new energy sources, particularly petroleum-based fuels, which are generally thought to be more harmful to human health than alternative fuels. Fossil fuel supplies are decreasing, and their prices are rising on a daily basis, badly impacting the economies of emerging countries that rely on petroleum imports. This work is aiming to investigate the performance and exhaust emissions of SI engines running by E15% gasoline dual fuel with LPG. The investigations have been concentrated on finding the optimized mixture of gasoline, ethanol, and LPG. ANSYS Workbench 16 software was adopted to model and analysis of venture type gas mixer. The mixer produces lower pressure to create a vacuum at the throat, which is utilized to readily suck gas fuel, and higher velocity at the mixer throat improves the mixing of air and LPG gas fuel. For the experimental investigation, the SI engine with compression ratio 8.2:1 is tested using neat Gasoline, E15% gasoline and E15% dual-fueled with LPG at (2, 3, 4) LPM by varying engine load 0- 100%. It was observed that ethanol addition to gasoline makes the engine operation leaner and improves engine combustion, performance, and exhaust emission. The ethanol content in fuel blends decreases BTHE by 19.23% and increases BSFC by 2.27% compared with pure gasoline. Carbon monoxide and Hydrocarbon emissions were also decreased by 4.16% and 1.44% respectively. CO2 and NOx were increased by 8.69% and 4.65. In a dual fuel mode, the addition of LPG fuel through a venturi gas mixer at the engine's air inlet also ensures sufficient oxygen molecule for complete carbon combustion for efficient thermal conversion, which raises the engine's thermal efficiency. LPG at 3LPM show lower of BTHE by 11.05% and shows higher of BSFC by 6.81%. Because of LPG higher cylinder gas temperature, CO and HC also reduced by 20.83% and 4.78% respectively.

Keywords— Ethanol, LPG, Mixer, Performance and emission, SI engine.

CHAPTER ONE

1. INTRODUCTION

1.1 Background of the study

Internal combustion engines (ICEs) are now widely used in a variety of industries, including transportation and commerce, which has made the energy problem and environmental pollution worse. As a result, saving fossil fuel and reducing harmful emissions are attracting attention from both the general public and the scientific community. The emphasis will turn to improving the engine and fuel system with the introduction of these and other new automotive technologies. To achieve their maximum social and environmental benefits, many emerging innovations would require higher octane fuel. Higher octane fuel makes for more optimal combustion phasing in engines with high specific outputs or compression ratios. Both fuel consumption and CO_2 emissions will be improved as a result of this (Thakur et al. 2017). By eliminating the need for high-load fuel enrichment, higher octane gasoline will improve the after-treatment system's conversion efficiency even further. Because of these considerations, higher octane fuel has been advocated.

In many nations, harmful exhaust emissions and the fuel efficiency of internal combustion engines are becoming major problems, and regulations governing these factors are becoming stricter. As a result, research has concentrated on lessening pollutants and enhancing internal combustion engines' thermal efficiency. Air pollution is fast becoming a serious global problem with the increasing population and its subsequent demands. Searching an alternative conventional fuel would help to reduced pollution. Vehicles running on cleaner fuels produce fewer harmful emissions and can offer some savings of fuel costs, compared with petrol or diesel fuels (Kim et al. 2017)

Liquefied petroleum gas is a mixture of hydrocarbons, mostly propane (C3H8) and butane (C_4H_{10}) isomers. The composition of LPG depends on its end use and varies greatly according to season, country, properties of the crude oil/gas supply used and refining process. Now a days, European LPG is a by-product derived almost equally from natural gas (NG) extraction (55%) and petroleum refineries (45%)(**Gumus,2011)**. Commercially available LPG for the automobile market must adhere to a standard that only regulates fuel attributes and does not set compositions. As a rule of thumb, countries having relatively cold climates tend to use a higher propane percentage, while warmer countries mostly use butane (Xing et al. 2017). Some studies have been dedicated to the investigation of the feasibility of using gaseous fuels as alternative engine fuels from performance and emissions perspectives. Many of the results reported so far have concentrated on the use of natural gas as the alternate fuel for internal combustion engines. The effect of injection timing of pilot fuel and pilot fuel quantity on the performance and emissions of an indirect diesel engine fueled with methane or propane also investigated (Saleh 2008).

1.2 Statement of the Problem

Harmful exhaust pollution and internal combustion engine fuel efficiency have become major concerns, and regulations on exhaust emissions and fuel economy are getting stricter. As a result, researchers have concentrated their efforts on minimizing exhaust emissions and advancing the thermal performance of internal combustion engines (ICE). These problems led to a search and using alternative fuels, which are attractive given their environmental friend nature. LPG is one of the alternative fuels and is used as a fuel for internal combustion engines, especially for SI engines. To overcome the challenges, by considering the properties of ethanol and LPG, it can obtain an optimal fuel that can minimize fuel economy and exhaust emissions problems. So, this thesis is aiming to examine engine performance and exhaust emission of SI engine running by using E15% gasoline dual-fueled mode with LPG as a fuel.

1.3 Objectives

1.3.1 General Objective

The general objective of this thesis work is to examine the spark ignition engine performance and exhaust emission characteristics using E15% gasoline dual fuel mode with LPG as running fuel.

1.3.2 Specific Objectives

The following set of activities has been performed to arrive at the intended main objectives:

- Geometry modeling and CFD analysis of venture gas mixer to examine the effect of velocity and turbulence kinetic energy of LPG and air inside the mixer.

- Conducting baseline experiment using pure gasoline.
- Conducting experiment using E15% gasoline fuel.
- Investigation of the engine characteristic parameter on E15% gasoline dual fuel with LPG at (2LPM, 3LPM, and 4LPM) under varying engine load (0-100%) with an increment of 20%
- Perform numerical study on engine parameter and venturi mixer.

1.4 Scope of the Study

This thesis focuses on numerical and experimental studies to explain the effect of E15% gasoline dual-fuel mode with LPG on engine performances and exhaust emissions of SI engines. To investigate this, a 3D Computational Fluid Dynamics (CFD) code and experimental analysis were adopted.

1.5 Significance of the study

The final result of this work will be expected to identify the merits and demerits of LPG-fueled engines against gasoline-fuel engines for the same operating condition. It will be expected to identify the optimum blending ratio between the two fuels to maximize its performance level. As a result, this investigation promotes to utilization of LPG as an alternative fuel in Ethiopia as well as to trigger further studies to investigate other alternative fuels and enhance engine performance.

1.6 Organizations of the thesis

This thesis' main body is broken up into five chapters. The second chapter examines the pertinent journal articles, proceedings, and publications that were needed for the investigation. Also, what is done in this thesis will be stated with reference to and by comparison with earlier studies. at the third chapter, the experiment setup and methodology are discussed for modification and engine performance parameters. The fourth chapter contains results from the performance analysis of the SI engine as well as discussions based on these findings. The fifth and final chapter analyzes the conclusions reached in light of the evaluation's findings as well as suggestions for further research.

CHAPTER TWO

2. LITERATURE REVIEW

This portion examines an alternative fuel utilized in SI engines and describes its composition. Alternative fuels are the most common type of energy that can be used with the current combustion technology. Alternative fuels are well known for having complementary qualities that improve combustion efficiency, which improves performance and reduces exhaust emissions of an engine.

2.1 The Common Limitation of gasoline engine

Automotive engines contribute significantly to environmental pollution. SI engine exhaust emissions contain nitrogen oxides (NOx), including nitric oxide (NO) and small amounts of nitrogen dioxide (NO_2), carbon monoxide (CO), unburned hydrocarbons (UHC), carbon dioxide (CO_2), and sulfur oxides. Carbon monoxide is existed during incomplete burning of fuels. When combustion is quenched close to the cold side of the cylinder or when there isn't enough oxygen close to the fuel (hydrocarbon) for complete combustion, this occurs. CO is a poisonous gas that, in high amounts, results in death as well as nausea, headaches, and exhaustion. In addition, it produces CO_2 when it combines with O_3 in the upper atmosphere, which depletes the ozone layer. The levels of pollution produced by SI engines depend on their operating conditions. Sparkignition engine emissions are influenced by four key operational factors, but they also have an impact on the stability and dependability of the system. These are Speed, Load, Air/Fuel Ratio, and Spark Timing. These variables can be categorized as engine control parameters since they can be changed to produce a particular engine performance.

New combustion models and alternative fuels were the subject of numerous investigations. Yet, a variety of alternative fuels that are still having operational issues are added to gasoline. Furthermore, the implementation of tougher fuel regulations is hampered by a limited understanding of the effects of gasoline characteristics on fuel economy and contemporary car emissions. In 2020, 70% of light-duty and commercial automobiles will be propelled by gasoline engines, according to the IEA. 58% of passenger automobiles will still be using ICE in 2050. The impact of gasoline components on fuel efficiency, combustion, and emissions, particularly fuel economy, must thus be thoroughly studied. And in this way, the final investigated results can be used to guide the fuel refining process to increase fuel qualit**y**.

2.2 The Advantage of Alternative fuels for SI engine

In spark ignition engines, the impact of ethanol as a fuel addition has been researched over the years. Based to research by (**Abdel-Rahman and Osman 1997**). 10% ethanol added to gasoline as a fuel additive improves engine power by 5%. Alcohol has some advantages beyond gasoline when used as fuel for spark-ignition engines. Compared to gasoline, ethanol offers better antiknock properties. Increasing the compression ratio can increase the engine's thermal efficiency.

When the peak temperature inside the cylinder is not as high, alcohol burns with lower flame temperatures and their brightness. in order to reduce NOx emissions and heat losses. The latent heat of vaporization of ethanol is high. Hence, the greater charge density improves volumetric efficiency since the latent heat cools the intake air. However, ethanol's heating value is lower than that of gasoline due to its higher oxygen concentration **(Koç et al. 2009)**. In its purest form, ethanol comprises 35% oxygen atoms. So, it can be handled like a hydrocarbon that has undergone partial oxidation. The so-called leaning effect can occur when ethanol is added to the blended fuel because it may improve the amount of oxygen available for combustion. Due to the leaning effect, CO emissions will greatly drop; under some operating conditions, HC and NOx emissions will substantially drop (Iodice et al. 2016) .Liquefied Petroleum Gas (LPG) is often incorrectly identified as propane; about the mixture's chief ingredient. LPG is a mixture of petroleum and natural gases that exist in a liquid state at ambient temperature when under moderate pressure (less than 13.78 bar). Liquefied petroleum gas, known as LPG, is used as an alternative fuel for vehicles with a combustion engine. LPG is a mixture of propane, butane, and other substances of a small amount and it is obtained as a by-product of being manufactured during the refining of petroleum **(Synák et al. 2019)**.

The gaseous nature of the fuel/air mixture in an LPG vehicle's combustion chambers eliminates the cold-start problems associated with liquid fuels. LPG defuses in air-fuel mixing at lower inlet temperature than is possible with gasoline. This leads to easier starting, more reliable idling, smoother acceleration, and more complete and efficient burning with less unburned hydrocarbons present in the exhaust. In contrast to gasoline engines, which produce high emission levels while running cold, LPG engine emissions remain similar whether the engine is cold or hot. Its highoctane rating means the engine's power output and fuel efficiency can be increased beyond what would be possible with a gasoline engine without causing destructive knocking. Such fine-tuning can help compensate for the fuel's lower energy density. The higher ignition temperature of gas compared with petroleum-based fuel leads to reduced auto ignition delay, is less hazardous than any other petroleum-based fuel, is expected to produce less CO, NOx emissions, and may cause less ozone formation than gasoline **(Mustaffa et al. 2019)**.

2.3 Properties of Liquefied petroleum gas

The liquid known as liquefied petroleum gas (LPG) is completely odorless, colorless, and highly combustible. Propane (C3H8), n-butane (C4H10), isobutene (methyl-propane), and varying ratios of other butanes (C4H8) are the main hydrocarbons that make up LPG **(Warade and Lawankar 2013).** In the presence of an ignition source, its leakage or abrupt release from a pressurized tank would result in the development of a vapor cloud, which would then ignite and cause an explosion. A major catastrophe brought on by an LPG explosion occurred in China's Shaguodawang restaurant in October 2015. Because the LPG cylinder valve was not securely linked, LPG leaked and combined with the air to generate an explosive mixture, which exploded in an open flame. 17 persons perished as a result of this explosion. Numerous researchers have looked at the characteristics of LPG explosions due to the frequency of these events and the seriousness of the effects **(Chen et al. 2019).**

LPG at atmospheric pressure and temperature is a gas that is 1.5 to 2.0 times heavier than air, but it can be liquefied when moderate pressure is applied or when the temperature is sufficiently reduced. It can be easily condensed, packaged, stored, and utilized, which makes it an ideal energy source for a wide range of applications. Normally, the gas is stored in liquid form under pressure in a steel container, cylinder, or tank. The pressure inside the container will depend on the type of LPG (commercial butane or commercial propane) and the outside temperature **(Lanje and Deshmukh 2012).**

2.4 Literature on the effect of using LPG with gasoline for SI engine

(Gumus, 2011) studied the, effects of variation in volumetric efficiency on the engine emissions characteristics with different LPG usage levels (25%, 50%, 75%, and 100%), on an engine operated with a new generation closed loop, multi-point, and sequential gas injection system were investigated. For this purpose, experiments were carried out under constant engine speed (3800 rpm) and different load (5%, 30%, 60%, 90%) conditions. The variations in volumetric efficiency, air–fuel ratio, brake thermal efficiency, brake-specific fuel consumption, brake-specific energy consumption, and exhaust gasses were examined. The volumetric efficiency decreased considerably at the use of 25% LPG level. As for the 50%, 75% and 100% LPG usage, volumetric efficiency decreased in proportion to LPG usage level. Air–fuel ratio decreases with the increase in LPG usage level and the minimum air–fuel ratio value was obtained at 100% LPG usage. At the use of a mixture containing 25% LPG, brake-specific fuel and energy consumption decreased while the brake thermal efficiency was maintained. Positive results were obtained at all LPG usage levels in terms of exhaust emissions. The best results were achieved at using 100% LPG for exhaust emissions. (Geo et al. 2019) this study was conducted to compare the gasoline direct injection and LPG port injection systems to examine how hazardous pollutants from commercial vehicles contribute to air pollution utilizing gasoline and LPG fuels. From the results of exhaust emissions such as THC, CO, NOx, and

CO2, the vehicle with the GDI system has a higher emission level than that with the LPI system

2.5 Energy content of fuels

The calorific value of a fuel is the amount of heat produced per kilogram (solid or liquid) or per cubic meter (gas) when it is burned in a calorimeter with an excess of oxygen. The amount of heat generated by a fuel's combustion at constant pressure and at normal temperature and pressure is called calorific value or heat of combustion. Water vapor is produced when a fuel product is burned. The 'higher calorific value' (HCV) is obtained when H_2O is existed in the combustion products as a liquid. By condensing this water vapor, certain techniques are used to recover the amount of heat it contains. The higher heating value (HHV, called as gross calorific value or gross energy) of a fuel is stated as the amount of heat generated by a particular quantity once it has been combusted and the products have returned to 25°C. When condensation of reaction products is practical, a higher heating value can be used to estimate the heating values for fuels since it takes into account the latent heat of vaporization of water in combustion products. The lower heating value (also known as net calorific value or LHV) of a fuel is the amount of heat generated when burning a specific amount (at 25°C or another reference state) and raising the temperature of the combustion products to 150°C. The lower heating value makes the assumption that the reaction products are not recovered and that water in the fuel will vaporize at its latent heat. If condensation of combustion products is difficult or heat below 150 degrees Celsius cannot be employed, it is advantageous for comparing fuels. (Feng et al. 2020). The calorific value of

LPG that contains more butane is higher than one that contains more propane. The engine's power output depends on the energy content per unit volume of fuel injected into its combustion chamber. The reduction in energy content per unit volume fuel is caused by adding different percentages of

Ethanol to a certain blend of Gasoline **(Mustafa 2009)**

2.6 Literatures on Blending of Ethanol and Gasoline

Alcohol has been used as a fuel for engines since the 19th century. Among the numerous alcohols, ethanol is known as the most suitable renewable, bio-based and eco-friendly fuel for spark-ignition engines. The effect of ethanol on gasoline as a fuel alternative has been studied through the years in spark-ignition engines. Investigated the engine performance and pollutant emission of a commercial SI engine using ethanol-gasoline blended fuels with various blended rates (0%, 5%, 10%, 20%, 30%). with increasing the ethanol content, the heating value of the blended fuels is decreased, while the octane number of the blended fuels increases. It was also found that with increasing the ethanol content, the Reid vapor pressure of the blended fuels initially increases to a maximum at 10% ethanol addition, and then decreases. The engine's output torque and fuel consumption slightly rise; CO and HC emissions sharply decline due to the leaning effect of the ethanol addition, while CO2 emissions rise as a result of improved combustion and NOx emissions are more dependent on engine operating conditions than the ethanol content. **(Hsieh et al. 2002).**

(Yüksel and Yüksel 2004) This investigation aims to increase the percentage of ethanol in bleeding fuel ration with gasoline. Carburetor modification is all that is required. Using an ethanolgasoline blend fuel, the engine output and pollutant emissions of a commercial SI engine were studied. The results of the experiments showed that using ethanol-gasoline blend fuel caused the engine's torque output consumption to slightly increase, the CO and HC emissions to significantly decrease due to the learning effect of the ethanol addition, and the CO2 emissions to significantly increase due to improved combustion. According to this study, employing ethanol-gasoline blend fuel would lower CO and HC emissions by around 80% and 50%, respectively, while increasing CO2 emissions by about 20%, depending on engine settings.

(Celik 2008) ethanol was used as fuel at a high compression ratio to improve performance and to minimize emissions in a small gasoline engine with low efficiency. Initially, the engine whose compression ratio was 6:1 was tested with gasoline, E25 (75% gasoline + 25% ethanol), E50, E75, and E100 fuels at a constant load and speed. It was determined from the experimental results that the most suitable fuel in terms of performance and emissions was E50. Then, the compression ratio was increased to 10:1. The engine was tested with E0 fuel at a compression ratio of 6:1 and with E50 fuel at a compression ratio of 10:1 at full load and various speeds without any knock. The cylinder pressures were recorded for each compression ratio and fuel. The results showed that engine power increased by about 29% when running with E50 fuel compared to running with E0 fuel. Moreover, the specific fuel consumption, and CO, CO2, HC, and NOx emissions were reduced by about 3%, 53%, 10%, 12%, and 19%, respectively.

(Zhang and Sarathy 2022) analyses the ideal ethanol-gasoline fuel blend ratio to increase the braking thermal efficiency of commercial SI engines. By varying the amount of wide-open throttle (% WOT), engine speed, and blend rate of ethanol-gasoline fuels, it was possible to compare numerous blend rates of ethanol-gasoline fuels with the fuel containing 10% of ethanol by volume (E10). According to the experimental findings, a proper ethanol-gasoline mixture ratio could improve engine torque output, particularly at low engine speeds. E40 and E50 fuels provide the maximum brake thermal efficiency at 58-73% WOT and 2000-2500 rpm. E20 to E40 fuels provide the highest MBT at 70 – 100% WOT and 1000 – 4000 rpm. (Thakur et al. 2017) This study used a gasoline-ethanol combination to enhance the performance parameters of SI engines. As seen during the trial, ethanol blends at lower proportions improved engine torque and stopping power. There was an increase in engine brake power and torque using E5, E10, and E20 of 2.31%, 2.77%, and 4.16%, respectively. For the higher volume of ethanol content that was really measured, BSFC rose. Using E20, E25, E30, E75, and E100, the BSFC increased by 5.17%, 10%, 20%, 37%, and 56%, respectively.

(Chansauria and Mandloi 2018) examines the impact of ethanol on the performance of sparkignition engines. In adding ethanol addition to gasoline leads to smoother operation; improves combustion and decreases the duration of combustion. It is reported that an overall decrease of 71% in the pollutants was cited with the ethanol content of 15% in the blend and also it increases the power output and overall efficiency of the engine and better volumetric efficiency could be achieved. The water content of ethanol fuel has a significant effect on the reduction of NOX emissions. CO2 emissions are an increasing trend with the adding of Ethanol in gasoline. **(**Mourad and Mahmoud 2019) studied the impact of low and middle gasoline-ethanol blends on fuel use and emission rates in the real world. The main objective of this study was to assess the effect of a mid-level blend on non-flexible fuel vehicles. Ignition timing advance (ITA) varied concerning engine load and fuel octane for each vehicle. More ITA was observed under high engine load for E10P and E27. Fuel injection type, engine CR, and aspiration type contributed to inter-vehicle variability of ITA. Large ITA was typically associated with octaneinduced engine efficiency gain. Adaptation of each vehicle to each fuel was also indicated by the lack of large increases in emission rates under a range of ambient temperature and relative humidity conditions.

2.7 Effect of using LPG with gasoline for SI engine

Logically, gas and liquid injections would give different values of volumetric efficiency. The latter strongly affects the engine performance more than the other parameters do. In particular, gas injection produces minimum volumetric efficiency as compared to the liquid injection because of either the lack of intake charge cooling due to vaporization of liquid fuel, or the displacement of fresh air in the intake system due to the volume occupied by the gaseous fuel. Due to the good nature of LPG features, such as a greater hydrogen-to-carbon ratio and the absence of harmful aromatics in the emissions, LPG has emerged as one of the significant alternative fuels and is a very promising energy source. Liquefied petroleum gas, is one of the popular alternative fuels for internal combustion engines. Currently, it is quite easy to adapt a gasoline Multipoint Injection (MPI) engine's fuel system to LPG supply. There are several types of systems that adapt the engine, and many manufacturers, which provide different structural solutions **(Mitukiewicz, Dychto, and Leyko 2015).**

(Alexander and Porpatham 2019) studded an effective experimental approach that can be applied to stationary single cylinder LPG fueled lean burn SI engine. This approach has been applied to operate the engine under different equivalence ratio at wide open throttle condition. Experiments were conducted at a constant speed of 1500 rpm and at compression ratio of 10.5:1.as shown in figure 2.1 in this study the effects of adding small amount of ethanol (5%, 10% and 20%) along with LPG were investigated. Ethanol on volume basis was injected along with carbureted LPG. A maximum

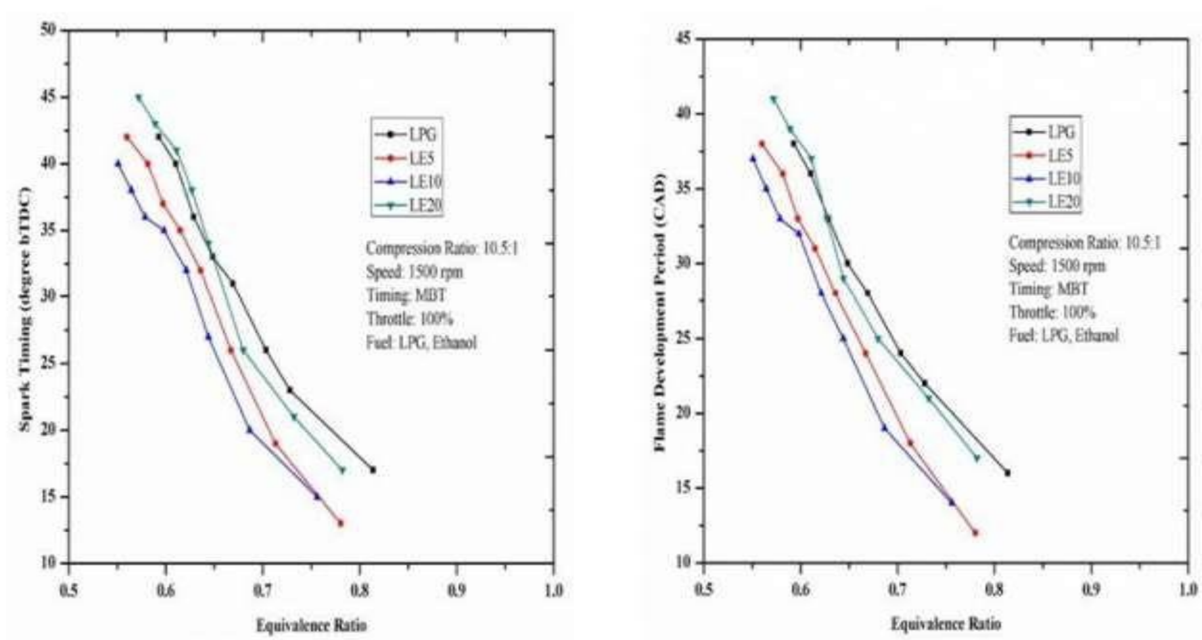

Figure 2.1:Variation of spark timing and flame development period with equivalent ratio.

(**Setiyo et al. 2016**) studied the impact of LPG usage rate on engine performance, emissions, and fuel economy in a spark-ignition engine with gasoline and LPG injection system. The experiment was carried out by modifying a four-cylinder 4-stroke engine with a dual fuel system. For this purpose, vehicle dynamometer, different LPG usage rates (as heat value 0%, 25%, 50%, 75%, 100%), fixed speed (3800 rpm) and various engine loads (5%, 30%, 60%, 90%) is carried out. The results of the experiment showed that exhaust emissions and fuel economy decreased in all

LPG usage rates compared to gasoline, but in terms of vehicle performance, only a 25% LPG usage rate achieved a positive result. Conducting to compare the gasoline direct injection and LPG
port injection systems to evaluate how hazardous pollutants from passenger cars contribute to air pollution using gasoline and LPG fuels. From the results of exhaust emissions such as THC, CO, NOx, and CO_2, the vehicle with the GDI system has a higher emission level than that with the LPI system. In addition, the results of CO_2 emissions show a similar result **(Lim and Im 2013).**

(Morganti et al. 2017) This paper investigated the auto ignition of C_3/C_4 hydrocarbon mixtures in a Cooperative Fuel Research octane rating engine. The in-cylinder heat transfer and residual gas composition were affected by auto ignition. In particular, nitric oxide (NO) was found to be a strong promoter of both fuel oxidation and auto ignition for propane, n-butane, and isobutene. For these fuels, the inclusion of NO in the residual gas at physically reasonable concentrations also enabled good agreement between the measured and modelled auto ignition timing. In all cases, physically reasonable concentrations of both carbon monoxide (CO) and unburned hydrocarbons (UHC) had an insignificant effect on auto ignition in comparison with NO. Exhaust emissions for CO, CO_2, NO_x, HC, O_2 using LPG-ethanol blends with different percentages of fuel blends at variable engine speeds ranging between 1000 and 5000 rpm were analysed. The optimum percentage of vaporized ethanol is 10% and this percentage results reduce 15-20% CO, 30% HC, and 30% NOx but increases 10% CO_2 emissions. The water content of ethanol reduces the exhaust emission temperature, so NOx reduces significantly **(Chen et al. 2019**Volumetric efficiency study was done by Cinar et al., in their study the camshaft of the engine was re-designed using the classical spline method for different valve lifts. at 1700-3200 rpm engine speeds range and wide-open throttle, the variations of engine brake torque, power, brake specific fuel consumption, and exhaust emissions like as HC, CO, $CO_{2,}$ and $NO_{x,}$ and exhaust gas temperature were investigated with unleaded gasoline and LPG fuels for different valve lifts. An improvement was observed on engine torque, BSFC, HC, and CO emissions with the increase of the valve lift to 8 mm and LPG fuel. At the low engine speeds, engine performance and exhaust emissions deteriorated with the increase of valve lift for gasoline fuel, while improvements were found at middle and higher engine speeds.

(Ravi et al. 2018) This work aims at investigating the effect of piston crown geometry on in-cylinder fluid motion theoretically using CFD simulation with 25%, 30%, 35%, and 40% squish area pistons. Also, experiments were conducted on LPG fuelled lean burn SI engine at the above-
mentioned squish area pistons at 100% throttle condition. The engine was operated at a constant speed of 1500 rpm with a fixed compression ratio of 10:1 by maintaining a constant squish velocity of 4 m/s. The performance, combustion, and emission characteristics were compared and discussed. For a squish velocity of 7.5 m/s occurring at 10°CA TDC, the turbulence kinetic energy was 61.5% higher for 30% compared to the 25% squish area piston. The turbulence intensity was 10% higher for 30% compared to 25% squish area piston and the length scale was 13% higher for 30% compared to 25% squish area piston. Overall, 30% squish area piston provides better charge motion and it Favors short flame travel and enhanced combustion of the fuel.

(**Mustaffa et al. 2019**) compared the LPG and ULP in the same SI engine was performed and analysed thoroughly at specified testing conditions. The comparison of performance parameters showed that liquid LPG injection has increased the engine torque and BMEP as compared with ULP. For BSFC, a significant improvement was observed at 50% and 75% throttle positions. Because of emissions, average NOx, HC, and CO emissions were higher for liquid LPG as compared with the ULP. The article includes the classification of LPG from the economic and safety point of view. It also assesses liquefied petroleum gas as an alternative fuel in terms of emission production. When comparison of LPG prices with diesel and petrol fuel, the price of LPG is on average almost 50 % lower. LPG has slightly more positive results on the production of CO_2 in comparison to petrol engines. Therefore it has good eco-friendliness **(Synák et al. 2019).**

2.8 Summary of the Literature

The above literature review shows the improved combustion, performance, and emissions characteristics with LPG and ethanol blends with gasoline as an alternative fuel at various conditions such as different compression ratios, air-fuel ratios, different percent of ethanol, and LPG. Based on the emissions and performance, it’s concluded that the LPG represents a better fuel alternative for gasoline and therefore must be taken into consideration in the future for transport purposes.

According to the literatures, power and torque produced from engine with using LPG is lower than that with using gasoline and fuel consumption increased with the increasing of engine speed and engine load. Lacking of air for combustion resulted in thick mixture and introduced HC instead of CO_2. For these reasons, engine should be modified to achieve complete combustion.

And different properties of ethanol and LPG which are crucial for the assessment of performance and emission characteristics of an engine have been discussed and compared to those of conventional gasoline. Also, octane number and volumetric efficiency increase with the increase in the percentage of ethanol in the blend. In the short term, LPG and ethanol as alternative fuel reviewed could displace 15percent of current usage of oil, or bring significant reductions in CO, HC emissions and help to reduce harmful greenhouse gas emissions.

2.9 Research Gap

Systematically studying the effect of gasoline components on fuel economy, combustion, and emissions is necessary. And in this way, the research results can be used to guide the fuel refining process to improve fuel quality. There are few complete comparative studies about the effect of different components of gasoline on fuel economy. Many studies aimed at new combustion models and alternative fuels. Literatures work on adding different percent of ethanol to improve the fuel quality and reduce fuel economy. When blending percent of ethanol increases, it creates engine instability problems and increases the water content in the fuel. LPG has the potential to reduce this problem. LPG has a higher-octane rating (105-110) than petrol (91-97), allowing it to be used at a higher temperature. This enables it to use at a higher compression ratio, as a result, prevents the occurrence of knock phenomena while running the engine at medium-high load conditions. It is delivered to the engine in the form of gas instead of liquid which can illuminate the fuel returning system to the cylinder.

CHAPTER THREE

3. MATERIALS AND METHODOLOGY

3.1 Introduction

In this portion, the simulation and experimental work should be carried out to achieve the intended main objectives of the thesis. It is suggested to gather quantitative information experimentally and using theoretical designing formulas. As much as possible assumptions were minimized to reduce errors. The CFD software ANSYS Fluent was used for the computational methods is described with the necessary illustrations. Designing of venture-type gas mixer device and its analysis is described here.

3.2 Materials

Under this portion the equipment's being utilized in this research along with detail of its characteristics and measurement capabilities described. It also contains the experimental setup and procedure, device calibration, equations and measurements that were used to obtain the performance and emission data. 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 equipment used for the experimental investigation are described with the necessary illustrations. In this thesis work, the main component of the process is fabricated (air /LPG mixing device), and will be employed in the modified experimental setup for experimentation. In this regard, the equipment used for manufacturing the mixer device is shown in Table 3.1. The material and equipment used for the experimental test are listed in Table 3.2.

Table 3.1:Materials used for mixer manufacturing

No.	Items	Quantity
1	Lathe machine	1
2	Drilling machine	1
3	Drilling tool	20mm, 3.5mm diameter
4	cutting tool	2
5	steel shaft	45mm diameter with 210mm length
6	Glass paper	3 pieces

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

No.	Items	Quantity
1	SI engine	1
2	Dynamometer	1
4	Rotameter (flow meter)	1
5	Pressure gauge	2
6	Gasoline fuel	7 L
7	LPG	12.5Kg
8	Flow control valve	2
9	Exhaust gas analyzer	1
10	Water pump	1

3.3 Methods

Under methodology there were combinations of procedures that were used to complete this study. Those were designing, fluent analysis and then manufacturing of the mixer device, and also organizing fuels, preparing the experimental setup of the test engine for experiment and analysis of experimental results. This thesis was developed based on neat gasoline, E15% gasoline and E15% dual fuel system with LPG fuels used to run a SI engine on test bed. Parameters such as specific fuel consumption, brake thermal efficiency and pollution were analyzed at varying engine load with constant speed were compared with that run on pure gasoline. This thesis work was conducted based on the following schedules summarized as shown in Figure 3.1

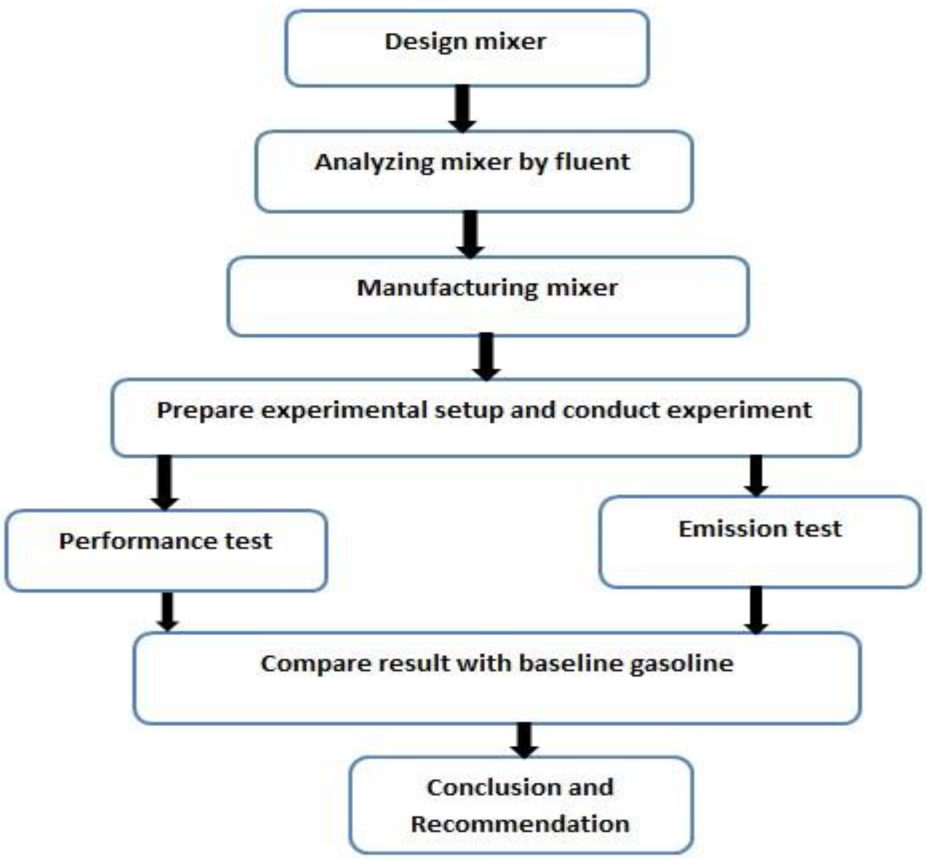

Figure3.1:Block diagram of the procedure of this thesis work

3.4 Venture-type gas mixer

The mixture is a this device is a flow passage of decreasing cross-section which reaches a minimum at the throat. It is used to mix the LPG fuel with air before it enters the engine. The air passing through the mixer during suction stroke increases velocity and the pressure in the mixer throat decreases. The decrease in pressure at the throat of the mixer creates a depression in the LPG regulator allowing proportional gas to pass through and mix with the incoming airstream. Figure 3.2 gas mixer is a device that uses the venturi impact based on Bernoulli's principle of a converging-diverging nozzle converts the pressure of the fluid (air) to the velocity at the throat to create a low-pressure zone. As the fluid passes through the throat tube, a pressure drop occurs. This pressure reduction is called the venturi effect. Gaseous fuel is sucked by the low pressure, which results in a perfect mix of air and fuel. If there is no additional pressure source at the air or fuel gas entrance, then the air-fuel ratio of the mixer will only depend on the mixer design **(Danardono et al.** flow of decreasing cross-section which reaches a minimum at the throat.it is used to mix the LPG fue with air before the mixture during suction stroke increases in velocity and pressure in the mixture throat decreases. The decrease in pressure at the throat of the mixture creates a depression in the LPG regulator allowing proportional gas to pass through and mix with the incoming airstream.

A gas mixture is a device that uses the venturi effect and applies Bernoulli's principle of a converging-diverging nozzle and converts the pressure of the fluid(air)to the velocity at the throat to create a low-pressure zone. as the fluid passes through the throat tube, a pressure drop occurs. This pressure reduction is called the venturi effect. the low pressure sucks gaseous fuel and creates a proper air-fuel ratio mixture. If there is no other pressure source at the air-fuel gas inlet. then the air-fuel ratio of the mixture will only depend on the mixture design.

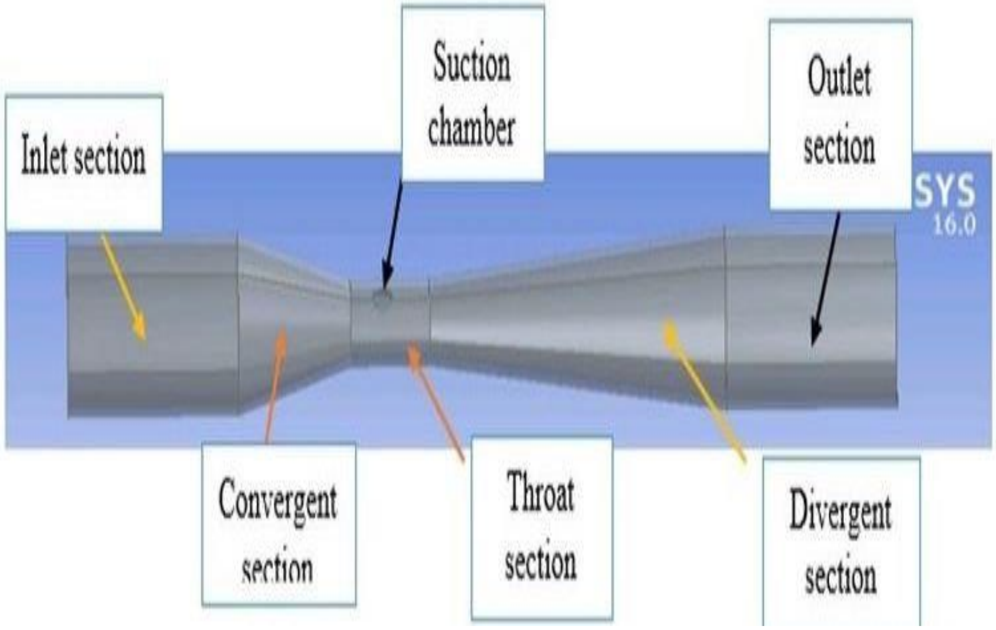

Figure 3.2:Venture-gas mixture

$$\dot{m}_a = Q_a \times \rho_a \tag{3-1}$$

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

Q_a = required airflow rate (m^3/s)

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

The finding airflow rate to the engine cylinder is calculated by the following formula (Dahake, Patil, and Patil 2016b).

$$Q_a = \frac{\eta_v \times V_d \times N}{2 \times 60} \qquad (3\text{-}2)$$

Where; η_v = volumetric efficiency

V_d = Displacement volume (m^3)

N = Engine speed (rpm)

The displacement volume **(V_d)** for the testing engine is obtained by the formula

$$V_d = \frac{\pi \times b^2}{4} \times s \times N_c \qquad (3\text{-}3)$$

Where; b = cylinder bore diameter

s = cylinder stroke

N_c = number of cylinders

$$V_d = \frac{\pi * 0.077^2}{4} * 0.058 * 1 = 2.7 * 10^{-4} m^3$$

The required airflow rate enters into the testing engine cylinder is

$$\dot{Q}_a = \frac{0.8 * 2.7 * 10^{-4} m^2 * 3600}{2 * 60} = 0.00815\, {m^3}/{s}$$

And the mass airflow rate to the engine

$$\dot{m}_a = 0.00815\, {m3}/{s} * 1.225\, {kg}/{m3} = 0.0099\, {kg}/{s}$$

Because the highest engine speed, at which maximum power production is produced, is used to determine the inlet air velocity to the mixer, the throttle must be wide open. As a result, the product of inlet air velocity and venture entrance area equals the volumetric airflow rate to the mixer inlet.

As a result, it comes from

$$Q_a = V_i \times A_i \tag{3-4}$$

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

$A_i = inlet\ area\ of\ the\ mixer$

To find the inlet velocity, first, the inlet area (A_i) of the mixer should be calculated

Where; D_i= inlet diameter of the mixer which is equal to the inlet manifold hose diameter of the indicated testing engine (i.e., 43 mm). Then the inlet area of the mixer is;

$$A_i = \frac{\pi * (D_i)\text{^}2}{4} \tag{3.5}$$

$$\frac{\pi * (0.0044)^2}{4} = 0.001519 m^2$$

Hence, air velocity at the inlet of the mixer in gasoline mode is:

$$V_i = \frac{Q_a}{A_i} = \frac{0.00815}{0.001519} = 5.365 m/s$$

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

Therefore, the throat diameter is equal to the product of beta and air inlet diameter of the mixer.

That is:

$$D_t = \beta \times D_i \tag{3-6}$$

Where; $D_t = throat\ diameter\ (m)$

□= throat to pipe diameter ratio (beta ratio).

$$D_t = 0.5 * 0.044m = 0.022m$$

Then the throat area (A_t) becomes;

$$\mathrm{A_t} = \frac{\pi * (\mathrm{D_i})\text{^}2}{4}$$

$$A_t = \frac{\pi * (0.022)^2}{4} = 0.000380m^2$$

During the continuity equation volume flow rate at the inlet is same to the volume flow rate at the throat and also that of at the outlet of the venturi.

$$Q_i = Q_t = Q_o = constant \tag{3-7}$$

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

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

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

Thus;

$$Q_t = V_t \times A_t \tag{3-8}$$

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

$$V_t = \frac{Q_t}{A_t} = \frac{0.00815 {m^3}/{s}}{0.000380m^2} = 21.44\ {m}/{s}$$

The air velocity is increased by the neck of the venturi as a linear function of the change in crosssectional area, and the Mach number (M) at the highest flow velocity (at the throat) must be less than 0.3 to be considered an incompressible flow. The Mach number at the venturi's throat is computed.

$$V = MC \tag{3-9}$$

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

M = Mach number,

C = sound speed (m/s)

$$M = \frac{V}{C} = \frac{21.44\ {}^{m}/_{s}}{343\,{}^{m}/_{s}} = 0.0623$$

The density of a fluid can be considered as constant in low Mach number gas flows, independent of pressure fluctuations in the flow. As a result, the fluid is deemed incompressible, and these flows are mentioned to as incompressible flows. Bernoulli's equation, in its most frequent version, is true at any arbitrary location along a streamline (Dahake, Patil, and Patil 2016b).

$$\frac{P_1}{\rho} + gZ_1 + {V_1}^2/2 = \frac{P_2}{\rho} + gZ_2 + {V_2}^2/2 \tag{3-10}$$

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:

$$\Delta P = \frac{{V_t}^2 - {V_i}^2}{2} \times \rho_a = 255.09 \text{ Pa}$$

3.5 Design calculation for Dual-Fuel

The implementation of a dual-fuel combustion strategy has recently been explored as a means to improve the thermal efficiencies of IC engines while simultaneously reducing their pollutions (Elnajjar, Selim, and Hamdan 2013). Dual-fuel combustion strategies have been demonstrated to be suitable on both spark-ignited and compression-ignited engines. On SI engines, dual-fuel technologies can be leveraged to combat knock. Knock typically occurs in high temperature and high-pressure in-cylinder conditions at which the fuel-air mixture will auto-ignite creating pressure shock waves in the cylinder. Knock can significantly damage the engine and is most prevalent at high loads where the efficiency reaches its peak. As such, high-efficiency engine performance with gasoline fuel is often limited by knocking (Saleh 2008).

The total mass flow rate on dual fuel mode is the sum of the mass flow rate of air and LPG.

$$\dot{m}_{T\,at\,intake\,manifold} = (\dot{m}_a + \dot{m}_{LPG})_{dual\,fuel} \tag{3-11}$$

Were. $(\dot{m}_{LPG})_{dual\,fuel} =$ The mass flow rate of LPG in dual fuel mode (kg/s)

Therefore; according to equation (3.1),

$$\dot{m}_a + \dot{m}_{LPG} = \dot{Q}_a \rho_a + \dot{Q}_{LPG} \rho_{LPG}$$

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

$$\rho_{LPG} = \frac{P}{\gamma \times T} \tag{3-12}$$

Where; P = ambient pressure (Pa)

T = ambient temperature (K)

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

The universal LPG constant is calculated as:

$$\gamma = \frac{R}{M_{LPG}} \tag{3-13}$$

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

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

$$\dot{Q}_a = 0.00810 - 1.54\dot{Q}_{LPG} \tag{3-14}$$

Liquefied petroleum gas has more components mainly propane, and butane are combusted. So, for stoichiometric combustion of propane, one mole of propane is completely burnt by five moles of air as shown below.

$$C_3H_8 + 5(O_2 + 3.76\ N_2) \Rightarrow 3CO_2 + 4H_2O + 18.8N_2$$

From this reaction, the stoichiometric air-fuel ratio is

$$\frac{mass\ of\ air}{mass\ of\ fuel} = \frac{\dot{m}_a}{\dot{m}_f} = 15.7 \qquad (3\text{-}15)$$

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

$\dot{Q}_{C_3H_8} =$ volume flow rate $of\ propane$ (m^3/s

$\rho_{C_3H_8} = density\ of\ propane$

Where the density of propane and universal gas constant is;

$$\rho_{C_3H_8} = \frac{P}{\gamma \times T} \quad and\ \gamma = \frac{R}{M_{C3H_8}}$$

Where; $M_{C_3H_8} = molecular\ mass\ of\ propane\ (g/mol)$

Then the volume flow rate of propane is:

$$\dot{Q}_{C_3H_8} = 0.7\dot{Q}_{LPG} \qquad (3\text{-}16)$$

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

$$\dot{Q}_a = 16.24\dot{Q}_{LPG} \qquad (3\text{-}17)$$

Combine (Eq. 3.17) and (Eq. 3.14);

$$\dot{Q}_a = 0.0074\,\mathrm{m}^3/\mathrm{s}\ and\ \dot{Q}_{LPG} = 0.000455\,\mathrm{m}^3/\mathrm{s}$$

$$\dot{m}_a = \dot{Q}_a * \rho_a = 0.0074\,m^3/s * 1.225\,kg/m^3 = 32.634\ kg/hr$$

$$\dot{m}_{LPG} = \dot{Q}_{LPG} * \rho_{LPG} = 0.000455\,m^3/s * 1.89\ kg/m^3 = 3.09\,kg/hr$$

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

$$V_i = \frac{\dot{Q}_a}{A_i} = \frac{0.0074m^3/s}{0.0009616m^2} = 7.69\ m/s$$

The velocity of LPG is obtained from Bernoulli's equation

$$V_{LPG} = \sqrt{\frac{2*\Delta P}{\rho_{LPG}}} \qquad (3\text{-}18)$$

Where, V_{LPG} is the flow velocity of LPG by (m/s) and the exit velocity of the fuel is 26.4m$/$s.

The fuel inlet area can be obtained from

.

$$A_{LPG} = \frac{Q_{L\dot{P}G}}{\dot{V}_{LPG}} = \frac{0.000455m^3/s}{26.4m/s.} = 1.73 * 10^{-5}m^2 = 0.173cm^2$$

The LPG fuel inlet diameter of the mixer depends on the number of fuel inlet holes.

$$A_{LPG} = \pi \frac{{D_{LPG}}^2}{4} * n \qquad (3\text{-}19)$$

Where D_{LPG} diameter of the LPG inlet and 'n' is the number of holes which is 2.

From the relation of area and diameter, the diameter of the LPG inlet can be calculated below.

$$D_{LPG} = \sqrt{\frac{4A_{LPG}}{2\pi}} = \sqrt{\frac{4*1.73*10^{-5}m^2}{2\pi}} = 3.39 * 10^{-3}m = 3.39mm$$

3.6 Designing and CFD analysis of venture type gas mixer

In this work, the geometric model was created in SOLIDWORK 2016 and then loaded into ANSYS 16 (CFD/fluent) for post-processing, meshing, and simulation. The new mixer models were meshes using the tetrahedron's approach, patch conforming, inflation, and curvature. The mixer was separated into tetrahedral pieces using the tetrahedron approach. Solid work was used to draw the mixer models without the movable mechanical mechanism. Computational fluid dynamics (CFD) is based on the numerical solutions of the primary governing equations of fluid dynamics, particularly the continuity, momentum, and energy equations. The computations are done using the commercial CFD code Fluent. The simple algorithm and a first-order upwind approach are the foundations of the numerical simulation. In comparison to standard initialization, hybrid initialization improves computation robustness and speeds up convergence. In the exist study, a tetrahedral structured mesh is used in all the simulations. Figure 3.3 shows the mesh generation tetrahedron mesh was chosen for the discretization of the mixer for the reason that the gas flow inside is the most important region in the venture mixer. Considerations during the design of the gas mixer are listed in the table 3.3

Table 3.3:Table Design Dimensions of Venture Mixer.

Inlet diameter and length	D = 44mm L =46 mm
Conical convergent section	θ_1=20° L, horizontal=28.85mm
Middle throat section	d=19.4mm L=20mm
Number and diameter of holes	n=2 dh=3.39mm
Conical divergent section	θ_2=8° L, horizontal=77.7mm
Outlet section	D=44mm L=46mm

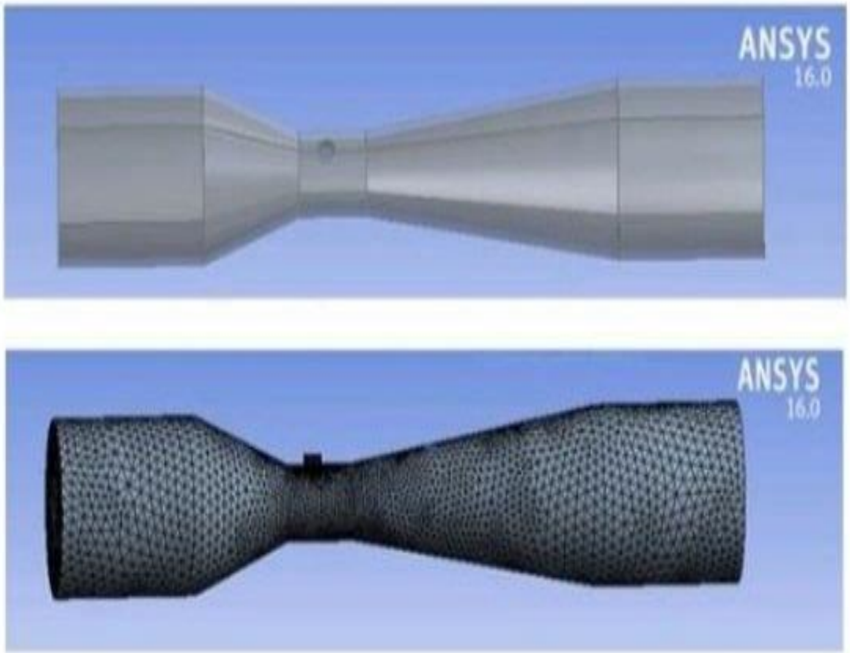

Figure 3.3: Geometry and Mesh of Venturi Type Mixer.

3.6.1 Grid Independence Test

A grid independence test was carried out to examine the stability of the mesh resolution used in the simulation and recognize the optimum mesh condition for precise computational results. grid independence study is performed to reduce the influence of the number of the grid size on the computational results

Table 3.4:Grid independence test of mixture geometer

Mesh type	node	Element	Velocity	Error
Course	3391	15356	54.2	-
Medium	3580	16341	65.4	20.6
Fine	99820	528606	72.9	11.46
Extra fine	113817	761423	73.6	0.96

3.7 Boundary conditions

- ✓ **Turbulence flow:** This involves the use of a turbulence model, which generally requires the solution of additional transport equations. In this case, the standard k-ε model is used.
- ✓ **Air inlet boundary:** In inlet boundary conditions, the distribution of all flow variables must be specified at inlet boundaries mainly flow velocity. This type of boundary condition is common and specified mostly where the entrance flow velocity is known.
- ✓ **Fuel inlet**: Fixed static pressure inlet boundary condition was used at the fuel inlets. Fuel will be inducted into the venturi because of the low pressure created at the throat. By using this boundary condition type, the mass of the fuel inducted into the venturi will be part of the solution.
- ✓ **Outlet boundary condition**: In outlet boundary conditions, the distribution of all flow variables must be specified, mainly flow velocity.

3.8 Turbulence model

The main key in internal combustion engines is turbulence modeling. To improve engine performance and minimize emissions, turbulence modeling is important for more understanding of the phenomena of spray, mixing, and combustion in an engine. The model of turbulence is used the standard *k–e* turbulence model It is based on the computation of the turbulence dissipation function (e) and the turbulence kinetic energy (k). K–e model is the most common model in the engine gaseous in-cylinder flow study. This model is used to solve the in-cylinder flow problems for the intake manifold. To predict the turbulent flows, we use the Favre-averaged Naiver-Stokes equations. The following equations describe the mass conservation equation, momentum conservation, and energy conservation equation (Kacem et al. 2016).

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = 0 \qquad (3\text{-}20)$$

$$\frac{\partial \rho u_i}{\partial t_i} + \frac{\partial}{\partial x_i}\left(\rho u_i u_j\right) + \frac{\partial p}{\partial x_i} = \frac{\partial}{\partial x_j}\left(\tau_{ij} + \tau_{ij}{}^{R}\right) + S_i \qquad (3\text{-}21)$$

$$\frac{\partial \rho E}{\partial t} + \frac{\partial \rho u_i}{\partial x_i}\left(E + \frac{p}{\rho}\right) = \frac{\partial}{\partial x_i}\left(u_i\left(\tau_{ij} + \tau_{ij}{}^{R}\right) + q_i\right) - \tau_{ij}{}^{R}\frac{\partial u_i}{\partial x_j} + \rho\varepsilon + S_i u_i + Q_H \qquad (3\text{-}22)$$

Where ‘‘u” is the fluid velocity, ‘‘ρ” is the fluid density, ‘‘sij” is the viscous shear stress tensor, ‘‘Si” is a mass-distributed external force per unit mass, ‘‘h” is the thermal enthalpy, ‘‘QH” is a heat source or sink per unit volume, ‘‘qi” is the diffusive heat flux. The subscripts are used to denote summation over the three coordinate directions.

The following equation are used to explanation the turbulent kinetic energy and its dissipation rate:

$$\frac{\partial \rho k}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i k) = \frac{\partial}{\partial x_i}((\mu + \frac{\mu_t}{\sigma_k})\frac{\partial k}{\partial x_i}) + S_k \qquad (3\text{-}23)$$

$$\frac{\partial \rho \varepsilon}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i \varepsilon) = \frac{\partial}{\partial x_i}\left(\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_i}\right) + S_\varepsilon \tag{3-24}$$

Where: S_K and $S\varepsilon$ are turbulence kinetic energy and dissipation rate.

3.9 Manufacturing Process of Venturi Gas Mixer

The designed venturi mixer device is manufactured from aluminum material by using a lathe machine and drilling machine at Ethiopian Defence University College of Engineering manufacturing workshop. Operations that were implemented during manufacturing the mixer are:

Turning: High speed rotation of the work piece allows material to be removed by a stationary cutting tool. Turning operation is executed for minimizing the external diameter of the aluminum shaft up to 40 mm external diameter and also for surface finishing by using a cutting tool on the lathe machine.

Tapering: tapering was performed on the convergent and divergent sections of the mixer. Firstly, the cutting tool is set at $21°$ on the compound head and then tapering the air inlet section or a convergent section. Secondly, the cutting tool is again set at 8^0 on the compound head for tapering the divergent or mixture outlet section.

Drilling: the aluminum shaft was drilled longitudinally at the center entirely by a drilling tool from minimum size to maximum size internal diameter of the mixer and also LPG inlet holes are drilled transversally by a diameter of 5mm drilling tool at the throat after dividing the circumference of the throat into two equal places and scratched by scratcher. Finally, the surface finishing of the tapered and turned surface was performed by using glass paper.

Figure 3.4: Prototype of Gas Mixer

3.10 Gasoline blend with ethanol

Blending is the process of having a product of desired ratios by mixing certain fuels in some suitable proportion. In this thesis work, an experiment is conducted on pure gasoline blending with 15% ethanol on a gasoline engine. Blending these fuels alters the physical and chemical characteristics of the original fuels. Ethanol increases the heat output of the unleaded gasoline, which produces more complete combustion resulting in slightly lower emissions from unburned hydrocarbons.

Figure 3.5:Fuels

Table 3.5:Thermodynamic Properties of Gasoline, Ethanol, and Lpg (Celik 2008)

Fuel property	gasoline	ethanol	LPG	
			propane	Butane
Formula	C_8H_{18}	C_2H_5OH	C_3H_8	C_4H_{10}
Molar C/H ratio	0.444	0.333	0.375	0.4
Molecular weight(kg/kmol)	114.18	46.07	44.1	58
Latent heating value (MJ/kg)	44	26.9	46.1	45.5
Stoichiometric Air/fuel ratio	14.6	9	15.6	15.4
Auto ignition temperature (K)	257	425	723	560
The heat of vaporization (kJ/kg)	305	840		
Research octane number	88–100	108.6	105+	103
Motor octane number	80–90	89.7	95.4	
Freezing point (K)	233	159	93	135
Boiling point (K)	300–498	351	229	272.5
Density (kg/m^3)	0.72 – 0.78	0.785	0.51	0.58
Specific gravity	0.739	0.788	0.585	

In general, adding ethanol to petrol makes fuel run more smoothly, enhances combustion, and cuts down on combustion time. Higher octane fuel ratings have positive effects on how efficiently petrol engines operate; the higher the octane rating of the fuel, the better(**Chansauria and Mandloi 2018**).

3.11 Dual fuel mode of E15% Gasoline fuel with LPG

Once the testing results of the petrol and ethanol mixture had been recorded in terms of braking torque, brake thermal efficiency, brake specific fuel consumption, and emission phenomena, dual-fueling with LPG was once more attempted. Under conditions of atmospheric pressure and temperature, LPG is 1.5 to 2.0 times heavier than air; however, when moderate pressure is applied or the temperature is appropriately decreased, it may liquefy. It's simple to compress, package, store, and use, making it an excellent energy source for a variety of applications. Normally The gas is stored under pressure as a liquid in a steel tank, cylinder, or container. The pressure with in the container is determined by the kind of LPG used (commercial butane or commercial propane) and the temperature outside (**Kansuwan and Kwankaomeng 2010**). LPG which contains more butane has a greater calorific value than LPG containing more propane. Even though ethanol has a higher octane rating than gasoline, its addition may reduce the engine's power since ethanol has a considerably lower energy value than gasoline. The energy content per unit volume of fuel fed into the combustion chamber determines the engine's power output. The decrease in energy content per unit volume of fuel produced by varying concentrations **(Elnajjar, Hamdan, and Selim 2013). Table 3.6 explains about lifan 177F** engine model, type, ignition, starting system, its compression ratio , displacement, bore and stroke diameter and engine oil capacity.

Table3.6: Testing Engine Specification.

Engine model	4 strokes, air-cooled
Engine type	9HP Lifan Engine (177F-B)
Ignition system	Non-contact transistorized ignition
Bore x stroke (mm x mm)	77mm x 58 mm
Fuel	Unleaded Petrol
Compression ratio	8.2:1
Displacement	270 cc
Net weight (kg)	25 kg
Max torque (N.M)	15.5 Nm at 2500 RPM
Starting system	Manual
Engine oil capacity	1.1L

3.12 Experimental setup and procedure

The SI enhene, hydraulic dynamometer, LPG kit, fuel supply, metering systems, measuring instruments, etc. make up the experimental setup. The primary goal of the experiment is to look into how a SI engine performs when fuels like LPG and petrol are burned in the engine under various load and speed conditions. The single-cylinder, four-stroke Lifan (Model-177F) engine was employed in this study; it has a 270cc displacement and an 8.2:1 compression ratio. Lifan 177F model was utilized as the test engine in the analysis and some modifications of the engine were done to use LPG with E15% gasoline as the burning fuel and to ease the experiment process. For instance, a mixer is introduced in the air intake manifold of an engine. The mixer enables air and LPG to mix to obtain the proper ratio before entering the combustion chamber.

It is a venture-type mixer that is connected to the LPG steel tank via a flexible hose. Figure 3.6 and 3.7 shows experimental stet up of spark ignition engine. in figure 3.6 the lpg gas and fresh air enters into the ventur gas mixer to make perfect mixture befor enters into carburetor. inside the carborator E15gasoline with lpg gas mix to make combustion in the cylinder to rotates th camshaft.The dynamoneter and emission gas analyzer connected with si engine to read the load and exhaust emission of si engine.

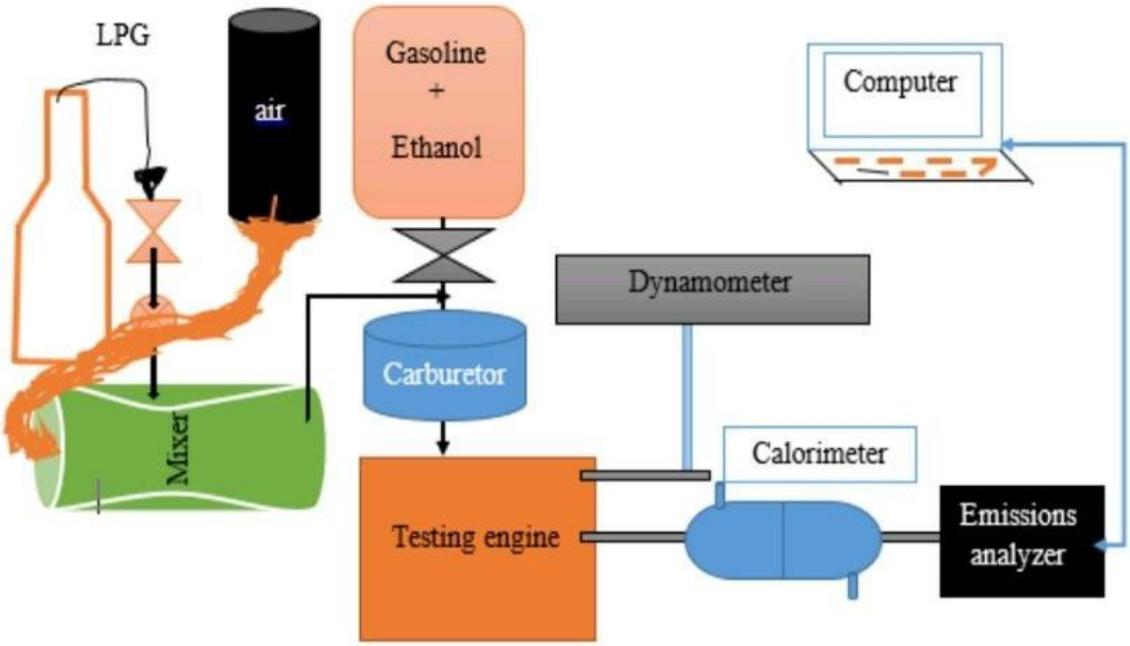

Figure 3.6:The Schematic View of Si Engine on Gasoline/Ethanol as A Fuel Dual-Fueled With LPG

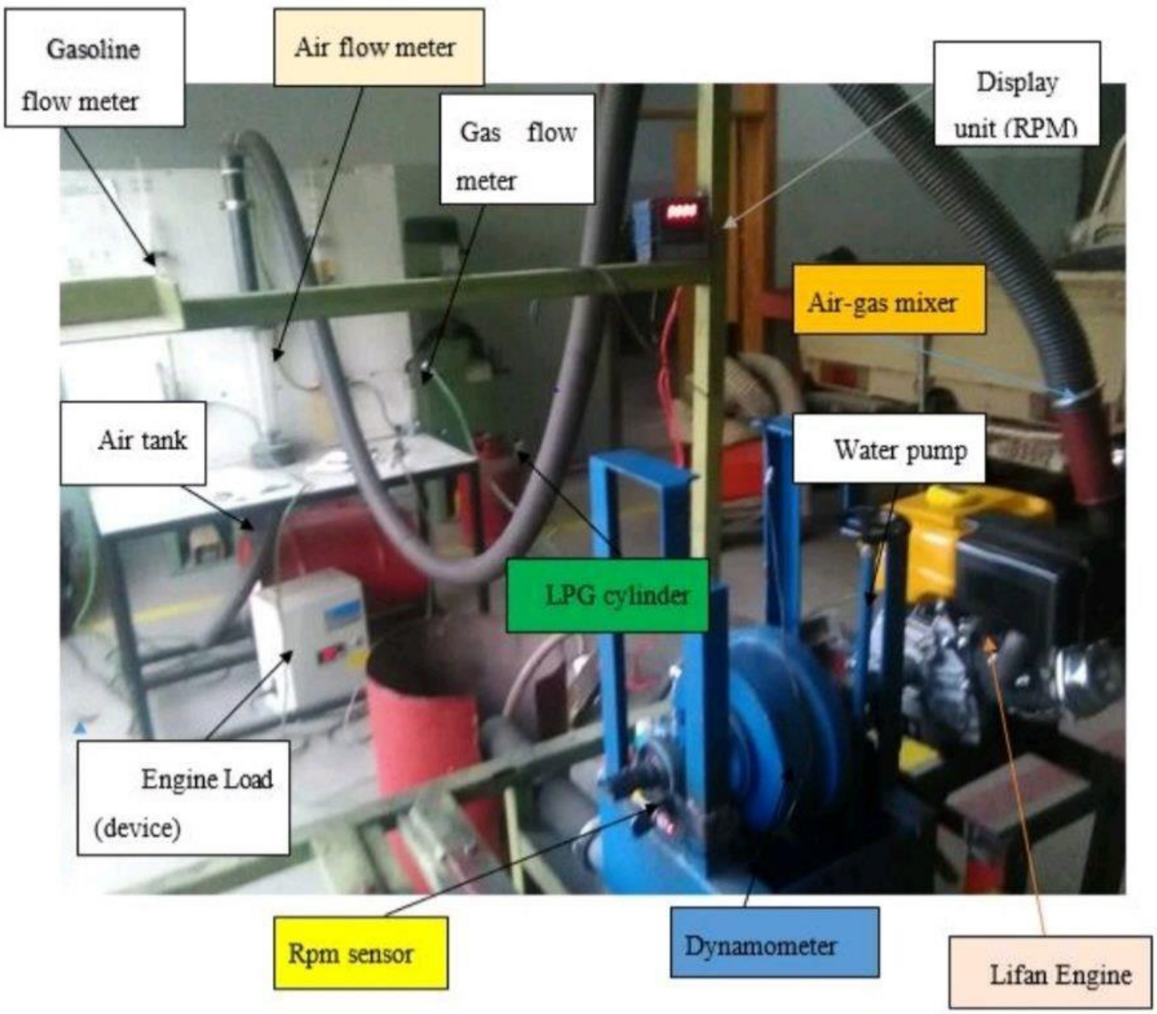

Figure 3.7:Modified Photographic View of Test Engine (new experimental setup in defence university)

3.13 Measuring Instrumentation and Principles

Instrumentation and principles are used for the measurement of different engine parameters when conducting any experiments. Measuring instrumentation is done to evaluate numerical and experimental results of engine performance, exhaust emission, and thermodynamics analysis.

3.13.1 Speed Measurement Method

The speed of the engine is measured by using a speed sensor fixed to a panel board or display board. The speed is measured with the help of an inductive type sensor fitted in the dynamometer. The signals produced are supplied to the show unit that is graduated to point the speed directly in the range of revolutions per minute (rpm).

3.13.2 Fuel flow rate and airflow rate measurement method

Figure 3.8 is airflow meter and fuel flow meter. Airflow meter an integrating type is used to measure the flow rate of combustion air. The air flow is measured by timing a known volume through the meter. The inlet of the air flow meter is fitted with an air filter Fuel consumption is calculated as a mass flow per unit of time and is represented by the symbol mf in engine testing. The more useful parameter of fuel flow measurement is specific fuel consumption. The gasoline level drop in a measurement container over a certain amount of time was used to calculate the engine's fuel consumption. Gasoline and E15% gasoline fuel is supplied to the engine from the fuel tank through a burette and a flow bulb.

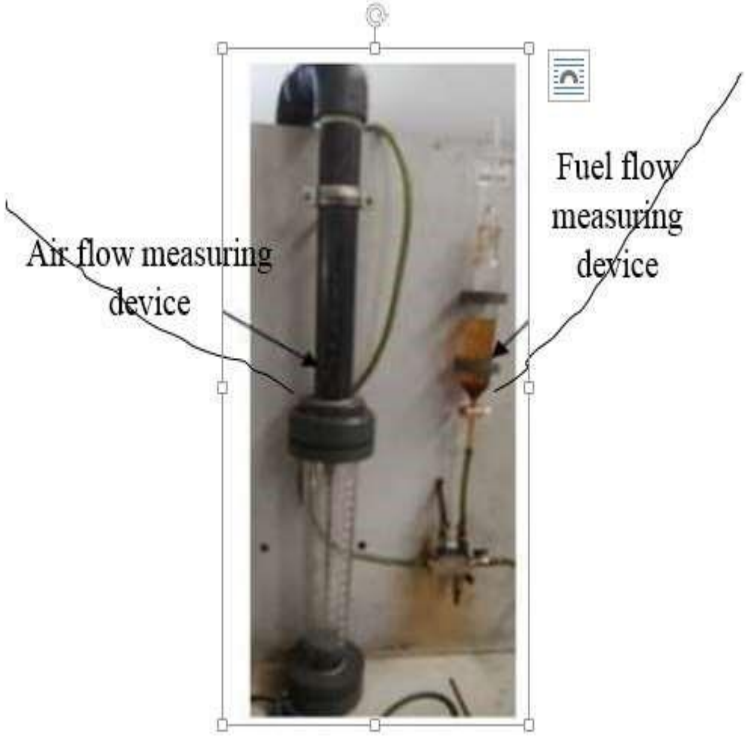

Figure 3.8:Air Flow Rate and Fuel Consumption Measurement Device

3.14 LPG flow meter/Rota meter

The basic Rota meter is the glass tube indicating type. The tube is precision for mad of borosilicate glass, and the float is accurately machined from metal, glass or plastic. The metal float is usually mad of stainless steel to offered corrosion resistance. The float has a precise measuring edge, and a scale positioned next to the tube allows you to see the reading. The range of flow rate is 0-10 LPM and operating temperature is 0-50 degree Celsius. For this thesis, LPG consumption was 2, 3 and 4 LPM. Figure 3.9 is basic rotameter that connect with lpg cylinder to support the flow of lpg from the cylinder to si engine thorough gas hose.

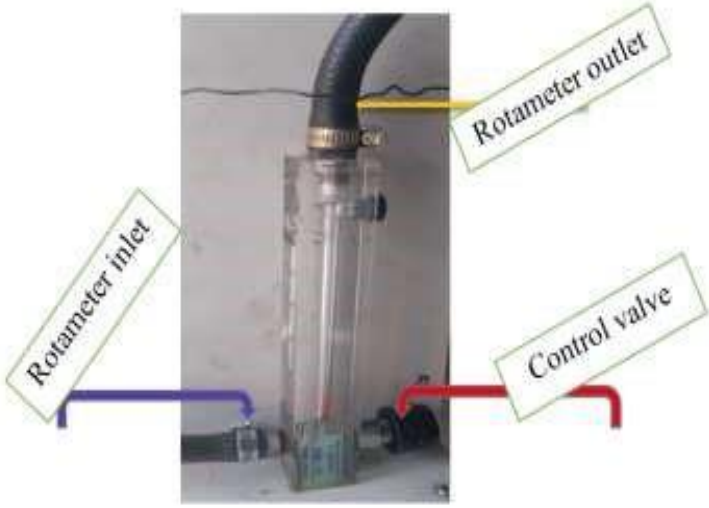

Figure 3.9:LPG Flow Meter/ Rotameter

3.15 Engine performance parameters

3.15.1 Brake Power (BP):

This power varies in amount depending on how much torque is being supplied to the crankshaft when the engine is running. This power is related to the torque applied to the crankshaft. Similarly, the brake power is also displayed on the computer when the engine runs, and its value varying as the engine loads changes (Mourad and Mahmoud 2019).

$$bp\ (kW) = \frac{2\pi NT}{60*1000} \qquad (3\text{-}25)$$

3.15.2 Break specific fuel consumption (BSFC):

It is an engine performance parameter that mainly determines the quantity of fuel consumed by the engine gram per hour (g/h) to develop a unit break power in KW while running at constant engine speed" N" in rpm. 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 (Mourad and Mahmoud 2019).

$$\text{BSFC}(\text{kg/kWhr}) = \frac{\dot{m}_{\text{fuel}}}{bp} \qquad (3\text{-}26)$$

3.15.3 Brake thermal efficiency (BTHE):

The ratio of output braking power to fuel-supplied energy is known as brake thermal efficiency.(Mourad and Mahmoud 2019).

$$BTE(\%) = \frac{bp}{\dot{m}_{fuel} * LHV} * 100 \tag{3-27}$$

3.16 Emissions Parameters and Measuring Procedure

In reality, the exhaust gas of an internal combustion engine contains both partial and complete combustion products, including CO2, H2O, soot, and unburned hydrocarbons. Under lean operating conditions the amounts of incomplete combustion products are small. Under fuel-rich operating conditions these amounts become more substantial since there is insufficient oxygen to complete combustion. Combustion may be defined as a relatively rapid chemical combination of hydrogen and carbon in the fuel with the oxygen in the air resulting in the liberation of energy in the form of heat. The combustion process is a fast exothermic gas-phase reaction. Complete combustion means that all carbon is converted to CO_2; all hydrogen is converted to H_2O.

Complete combustion is nearly impossible to achieve. (Masi 2012)**.**

Initially connect (plugin) the exhaust gas analyzer from the power supply and turn it ON. Figure 3.10 a gas analyzer was used to measure the engine exhaust emissions without catalytic treatment, and on the

exhaust side, a long tailpipe with heat insulation was fitted. Exhaust gas sampling points are offered for emission measurements. This was utilized to gauge the engine exhaust's emissions of nitric oxide (NO), carbon dioxide (CO2), carbon monoxide (CO), and unburnt hydrocarbon (HC).

After the exhaust gas temperature was stabilized, the tests were all run under steady-state conditions. The existing test engine has undergone some changes. To improve the reliability of the recordings, the temperature of the environment was kept constant during the studies. During each test, a constant load was applied, followed by a stabilization period. To allow for stability, the engine was always started before starting recording at each load situation. To verify repeatability, each test was repeated three times.

Figure 3.10:Automotive Exhaust Emission Analyzer

- **Carbon monoxide (CO) Emission:** CO is produced in an engine when it is operated with a fuel-rich equivalence ratio. When there is not enough oxygen to convert all carbon to CO_2, some fuel does not get burned and some carbon ends up as CO.
- **Carbon dioxide (CO_2):** The CO_2 emission is a sign of the complete combustion of fuel in the combustion chamber in the exist of excess oxygen.

- **Hydrocarbons: -** is a product of incomplete combustion of fuel. If a rich fuel mixture is used there is not enough oxygen to react with all the carbon, resulting in high levels of HC in the exhaust products. Often as an engine ages clearance between the pistons and cylinder walls increases due to wear. This increases oil consumption and contributes to an increase in HC Emissions.
- **NOx emissions: -** One of the most harmful pollutants emitted by spark-ignition engines is nitrogen oxide, or NOx. NOx emissions are influenced by three things. The duration of the reaction, the amount of oxygen present in the chamber, and the temperature of the gas inside the cylinder are some of these variables.

CHAPTER FOUR

4. RESULT AND DISCUSSION

4.1 Introduction

This portion presents numerical simulations of venturi mixer and analysis of experimental work on the performance and emission of E15% gasoline dual-fueled with LPG used as a running fuel for air cooled single cylinder gasoline engine. For each of the experiments, the impact of blended fuels on engine performance and exhaust emission parameters were recorded and evaluated.

4.2 Venturi gas mixer simulation

Gas mixer is a device used to determine the amount of gas and air before entering the engine. It injects the gas into the intake air stream of an internal combustion chamber using a combination of radial holes and radial tubes located around the perimeter of an air flow passage. Figure 4.1 vshows that the mixer produces lower pressure to create a vacuum at the throat, which is utilized to readily suck gas fuel the designing of gas mixture venturi is significant to keep the mixture in laminar flow in venturi for proper mixing of gas into incoming air**.** A venturi gas mixer is a device used to supply fuel and air to a carburetor in dual fuel mode operation. Bernoulli's principle is that, when a fluid passes through a venturi, its velocity raises and consequent increase in kinetic energy as a result of its pressure decrease at the throat. The high velocity at the throat portion tends to decrease in the divergent portion, which gives rise to turbulence downstream. According to the venturi principle highest velocity at the mixer throat improves the mixing of air with LPG gas fuel as seen in Figure 4.2.

The flow via the venturi mixer is turbulent, so it was important to use a turbulence model in numerical calculations. For this purpose, a two-equation turbulence model k-ε was used to analyze the Turbulence Kinetic Energy (TKE), along with the flow through the venturi mixer.

Too small throat causes air flow limitation at the throat whereas too big throat causes small pressure drop at the throat to begin the introduction of gas. This is undesirable for better mixing fuel and air. Similarly, more hole diameter and number of holes gives rich mixture whereas less hole diameter and number of holes provide lean mixture. Increased throat diameter has given good test results because of less pressure drop created at throat. Figure 4.3 shows that, highest TKE and turbulence energy dissipation rate lead to better mixing quality. Better mixing quality is generally accompanied by an increase in pressure drop.

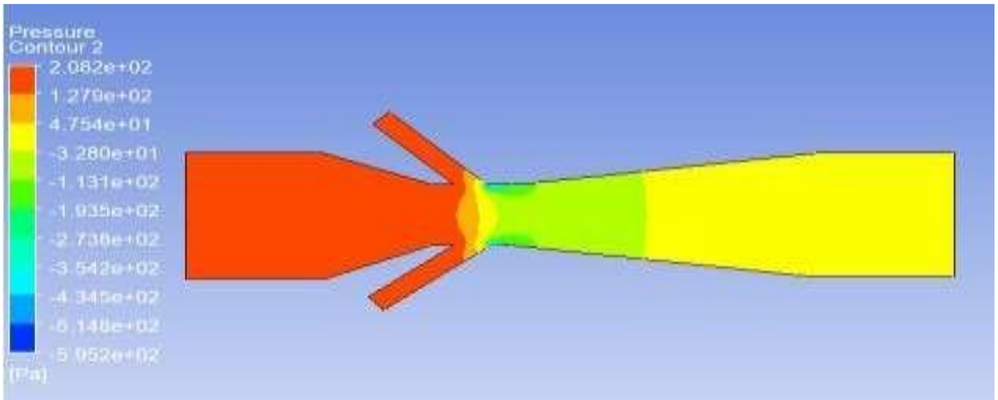

Figure 4.1:Pressure Drop

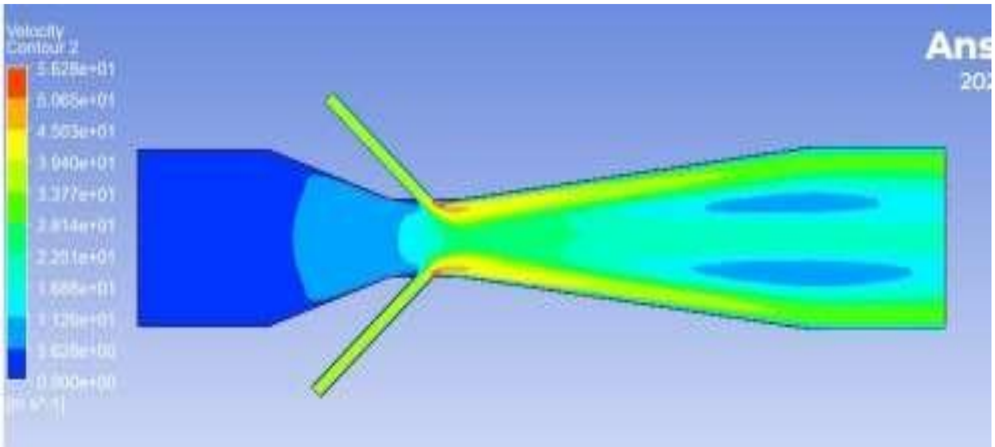

Figure 4.2:Velocity of the Fuel inside the Mixer

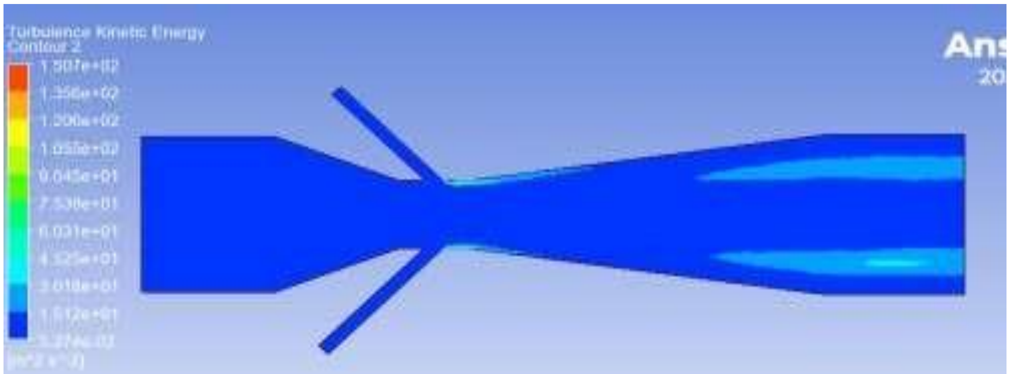

Figure 4.3:Turbulence Kinetic Energy

4.3 Experimental Investigations of Engine performance

To investigate the SI engine's performance and emission behavior, an experimental investigation was conducted. The performance characteristics such as brake thermal efficiency and brake specific fuel consumption is described on this portion.

4.3.1 Brake thermal efficiency (BTE)

It is the measure of the engine’s performance in terms of the conversion of heat energy into mechanical energy or the transformation of energy available in fuel into useful power production is referred to as brake thermal efficiency. Figure 4.4 below shows the variation of brake thermal efficiency for pure gasoline, E15% gasoline and dual fuel mode with different amount of LPG fuel at all loads. 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 gasoline mode.

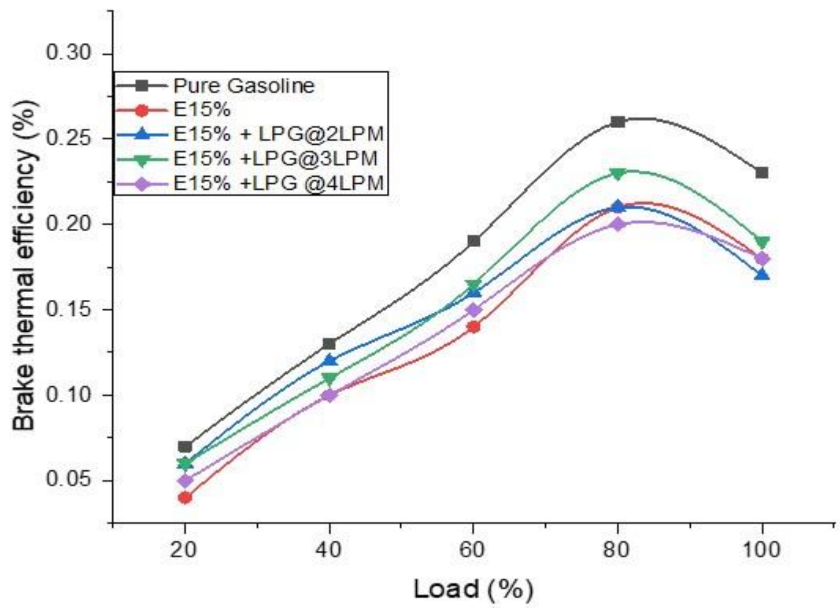

Figure 4.4:Variation of Brake Thermal Efficiency versus Engine Load.

4.3.2 Brake-specific fuel consumption (BSFC)

Calorific value and the density of the blend are the major characteristics of the BSFC of a fuel. Figure 4.5 shows brake specific fuel consumption with respect to engine load for pure gasoline fuel, E15% gasoline and LPG mixtures. Brake specific fuel consumption for all 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 all modes of operations because of an increase in combustion rate due to high air-fuel ratio and high combustion temperature. contrarily, in dualfuel mode, an increase in LPG flow rate leads to an increase in brake specific fuel consumption. This is because increasing the amount of LPG flow rate further decreases the amount of injected gasoline fuel. Moreover, the lower thermal value of LPG is higher than gasoline in terms of mass but, in terms of volume, lower thermal value of gasoline is greater than LPG. This means that more volumes of fuel must be used to achieve the same amount of power what gasoline have. This is the reason, BSFC values of LPG was higher than gasoline.

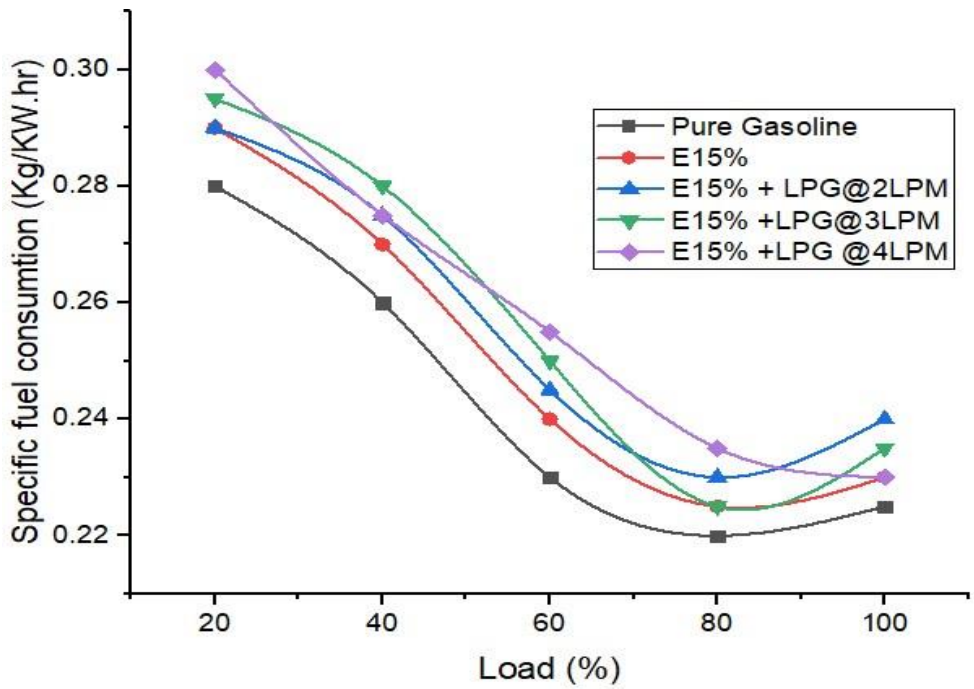

Figure 4.5:Variation of Brake Specific Fuel Consumption with Respect to Engine Load.

4.4 Experimental Investigations of exhaust emission

Knowing the exhaust emissions of a specified fuel is an important issue for future controlling its emissions via different options, either searching alternative fuels which results in less emission or optimizing the engine operating parameters. This section addresses the emissions of neat gasoline, E15% gasoline and dual fuel mode with LPG on SI engine.

4.4.1 Exhaust Emissions of Carbon Monoxide (% Vol.)

CO is a product of incomplete combustion due to an insufficient amount of oxygen availability in the air-fuel mixture or there is not enough time in the cycle for completion of combustion. Another reason for higher CO emissions may also the inadequate mixture formation of gaseous and liquid fuel. As a result, CO emissions are impacted by oxygen levels in the air, carbon content, and fuel combustion efficiency. The carbon in the fuel oxidizes during combustion, resulting in CO emissions. When oxygen availability is inadequate, however, incomplete combustion occurs, resulting in a higher CO value.

From experimental result the CO emission minimize as compared with pure gasoline fuel for E15% blend. Better combustion results in lower emissions. The volumetric specific emission of carbon monoxide at maximum load varies between 1.5 and 2.3% for all fuels. This effect is attributed because the compared with pure gasoline fuel for various blends. Stoichiometric air-fuel ratio of the ethanol blends. Due to better combustion these emissions are decreases and increase of actual air-fuel ratio of the decreased. This effect is attributed because the ethanol blends as a result of the oxygen content in stoichiometric air-fuel ratio of the ethanol blends the ethanol. The effect of different amounts of LPG used by dual fuel mode with E15% on the CO emissions under the different loading comparable to the pure petrol and E15% shown in dual fuel operation in Figure 4.6.

CO emissions minimize with the increase in LPG usage level. This is because, the carbonhydrogen ratio of LPG fuel is low and LPG in a gas state burns efficiently with a more homogenous mixture. The CO concentration from using dual fuel with LPG was significantly lowest compared to E15% and pure gasoline.

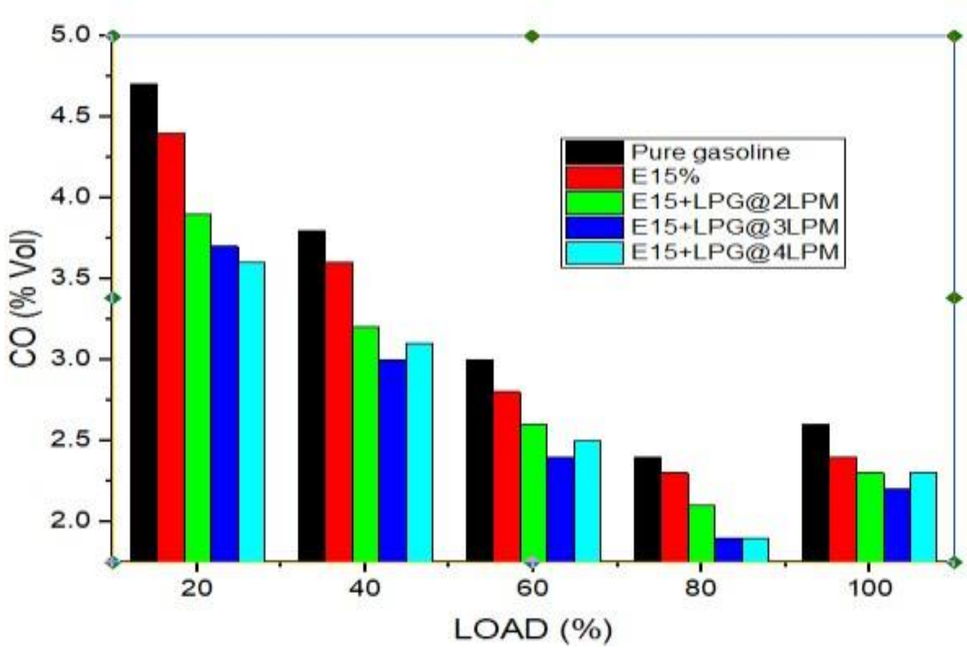

Figure 4.6:Co Emission Of E15% Dual-Fueled With LPG

4.4.2 Exhaust Emissions of Carbon Dioxide (%Vol.)

Regarding to CO_2 emissions, due to LPG fuel has lower carbon content than petrol or diesel and it emit less CO_2 which plays a significant influence in global warming during combustion. Lower CO_2 emission was measured with LPG according to gasoline. The CO2 release is a sign that the fuel has burned completely in the combustion chamber while there is an excess of oxygen present. When the engine condition goes leaner, the combustion process is more complete and the concentration of CO_2 emission gets higher. CO_2 emission increases as the percentage content of ethanol increases in the blend. The CO_2 concentrations have an opposite behavior in comparation to CO concentrations. This is due to the improved combustion process as a result of the oxygen content in the ethanol fuel. The CO_2 emission indicated the potential of complete combustion. The CO_2 concentration increased as the engine load increased.

Figure 4.7 shows the increasing effect of the dual fuel mode of an engine by LPG. Because of having a low C/H ratio, it creates complete combustion. But the amount of CO_2 produced decreases at 4LPM of LPG. This happened because of less oxygen at the air intake manifold.

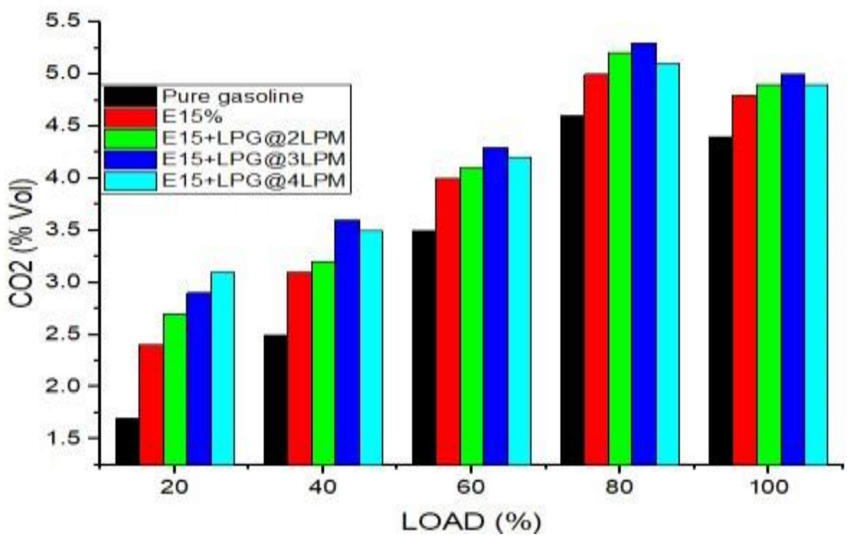

Figure 4.7:Co2 Emission Of E15% Dual-Fueled With Lpg

4.4.3 Exhaust Emissions of Hydrocarbons (ppm Vol.)

The unburned hydrocarbon emission is a very crucial parameter that could be used to present the losses in indicator power. The presence of HC in exhaust gasses points out that fuel didn't burn completely. Main reasons for failure to complete combustion and formation of HC are lower temperature, lack of oxygen and non- homogeneity of mixture. The figure below shows, it can be found that the HC emission decreases when using ethanol, while it increases when using LPG at all compression ratios compared with gasoline. Again, this behavior could be explained in the same way as previously mentioned above. Several studies have also reported similar HC trends when using LPG and ethanol as fuel for SI engines. HC emissions with various engine speed for LPG and gasoline were illustrated in Figure 4.8, HC emissions reduce with the increase in LPG usage level. Because the mixture becomes more homogeneous with the increase in LPG usage level, combustion is improved and HC emissions decrease. HC and CO emissions are strongly related to cylinder gas temperature at post-flame. LPG has higher cylinder gas temperature than that of gasoline, and this will enhance HC and CO oxidation in the post flame. This factor also leads to the low value of HC and CO emissions.

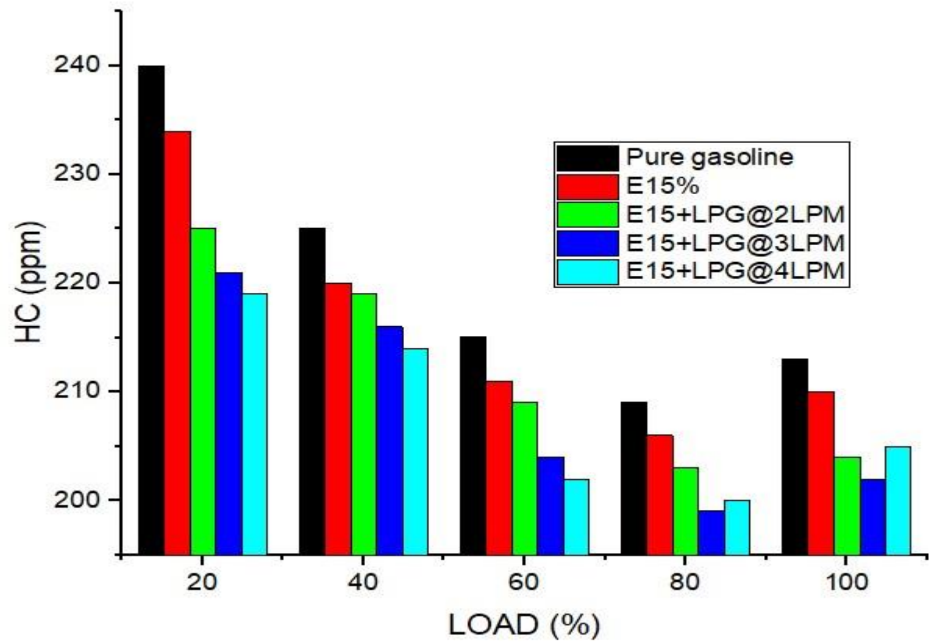

Figure 4.8:Hc Emission Of E15% Dual-Fueled With Lpg

4.4.4 Exhaust Emissions Nitrogen Oxide (ppm Vol.)

The formation of NOx emission mainly depends on the availability of oxygen, higher temperature developed during the combustion, and the residence period for which oxygen-nitrogen reactions occur to a substantial completion level. It is seen that the greater NOx emissions generally observed with an increasing the engine load. Especially at the higher engine load, combustion temperatures increased depending on the increments of the fuel mass taken into the cylinder and this results in higher NOx emissions. Moreover, heating value and burning speed of the LPG fuel are higher than the gasoline fuel with lack of charge cooling effect which obtained with gasoline fuel by its evaporation and also air-fuel mixture taken into the cylinder increased with the increase of the valve lift.

It can be also stated that lower exhaust temperature allows for less oxidation of the hydrocarbons released from the crevices during expansion stroke. The temperature on cylinder wall is lower owing to coolant effect. This phenomenon prevents to oxidize hydrocarbon molecules and causes to obtain lower exhaust gas temperature. Consequently, NOx emissions tend to decrease at lower exhaust gas temperatures.

As shown on Fig 4.9, NOx emissions increased with LPG according to gasoline due to higher cylinder gas temperature and due to LPG's increased calorific value compared to gasoline, which would greatly boost engine performance and hence raise exhaust gas temperature. High gas temperatures resulted in greater NOx emissions, with extra LPG having a substantial impact on NOx emissions.

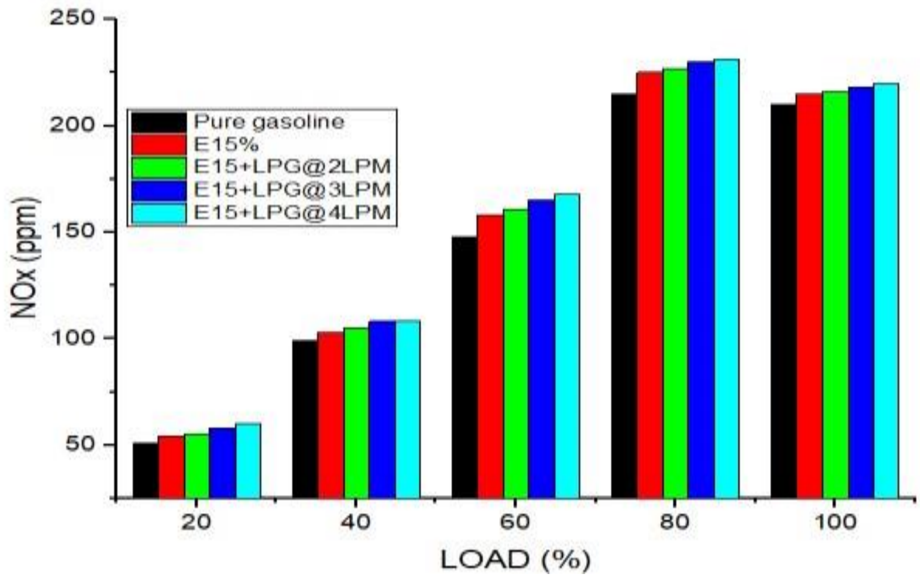

Figure 4.9: NOx Emission of E15% Dual-Fueled with LPG

CHAPTER FIVE

5. CONCLUSION AND RECOMONDATION

5.1 Conclusion

In this study, the performance characteristics of a single cylinder SI engine was determined under three fuel modes of operation (pure gasoline, E15% gasoline and dual fuel mode with LPG at 2,3 and 4 LPM). From the present work it has been concluded that LPG fuel is one of a promising alternative fuel for spark ignition engine as compare with gasoline in terms of engine performance characteristics and exhaust emission of the engine. The study was focused on the performance and emission testing of the SI engine using gasoline and E15% gasoline blend as base line fuel. The same engine was evaluated for its performance and emission with LPG once the necessary elements for the LPG fuel system were secured and fitted. The conclusion is drawn from the comparison of thus obtained test results.

From the experimental investigation, at a constant speed and different engine loads (no-load, 20%, 40%, 60%, 80%, and 100%) the optimal engine load is carried out (80%) according to the performance of the engine. A higher-octane rating of the fuel has a better implication on the efficiency of the gasoline engines, the more the octane rating of the fuel the better in gasoline engines. It was observed that the decrement of break thermal efficiency for E15% gasoline by 19.23% and dual fuel mode by LPG@3L/M 11.05%. This is due to an increasing calorific value in the case of LPG. It was found that with increasing of the percent of engine load (0%- 100%) brake specific fuel consumption of the engine would be increased, up to the optimal engine load (80%) and slightly decreases at 100% for all tested fuels. Based on analytical and experimental observation, it has been found that the average fuel consumption for E15% and E10+LPG@3LPM was increased by 2.27% and 6.81% respectively compared with pure gasoline. Based on the emission characteristic, it was seen that at 15% addition of ethanol to gasoline significantly decreases the emission of CO and HC by nearly 4.16% and 1.44% respectively. When E15% dual fuel with LPG, also reduce the emission of CO and HC by 20.83% and 4.78% respectively at 3LPM. Ethanol has an adverse effect on CO_2 emissions, with the addition of ethanol in gasoline proportion of CO_2 and NOx increases in the exhaust emissions by 15.21% and 6.97% respectively.

5.2 Recommendation

- Using more numbers of holes for venture gas mixer with inclined angle has better mixing quality. So, it needs more hole rather than 2 holes.
- Studying all factor affecting engine performance parameter is very important to identify crucial one and more optimizing the result.
- The availability of LPG in the local market is only 25% propane and 75% butane. It would be preferable to look into the effects of raising the propane proportion. When compared to butane, propane has lower carbon to hydrogen ratio and better vaporizing properties.
- The NOx emission was slightly increased. Then it needs to reduce by using some techniques like Exhaust Gas Recycle
- the effect of lpg and ethanol fuels on gasoline engine performance brake thermal efficiency and, brake specific fuel consumption and eexhaust emission of si engine were investigat in this thesis however the research of engine performance parameter and exhaust emission of the engine is not limited future study as extension and elaboration of this research is suggested as follows .
- This thesis limited to singile cylinder air cooled si engine it needes further studing on multi cylinder is engine.
- For having better mixing quality and homogeneous air gas mixture , the venture gas mixture needs additional fuel inlet hole.

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