ADAMA SCIENCE AND TECHNOLOGY UNIVERSITY

School of Mechanical, Chemical and Materials Engineering

Thermodynamic Analysis, Performance and Emission Evaluation of Ethanol- Gasoline Blend Fuel (E15)

A thesis submitted to Adama Science and Technology University in partial fulfillment of the requirements for the award of the degree of Master of Science in Automotive Engineering

by

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APPROVED BY BOARD OF EXAMINERS

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CANDIDATE'S DECLARATION

I hereby declare that the work which is being presented in the thesis entitled "**Thermodynamic Analysis and Performance Evaluation of Ethanol- Gasoline Blend Fuel (E15)"** in partial fulfillment of the requirements for the award of the degree of Master of Science in Automotive Engineering is an authentic record of my own work carried out from November 2017 to (Ass.profosser) September 2018 under the supervision of Melesse HaileDepartment Mechanical systems and Vehicle Engineering department, Adama Science and Technology University, Ethiopia.

The matter embodied in this thesis has not been submitted by me for the award of any other degree or diploma. All relevant resources of information used in this thesis have been duly acknowledged.

Name of Student

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Signature Date

This is to certify that the above statement made by the candidate is correct to the best of my knowledge and belief. This thesis has been submitted for examination with my approval.

Name of Advisor

Melesse Haile (Asst. Professor) ------------------------- -----------------

Signature Date

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The security of strength and success in all my life is God. So on the first place I would like to thanks my great God for all support in my life. The role of my family has significant meaning in the very beginning of my life so I would like to express my darling to them. Next, I would like to express my special gratitude for my adviser Melese Haile (Ass.professor) who is not only adviser on this thesis but also thought me c++ program starting from the beginning. I also thanks Idiris Ilmi the program head of Mechanical System and Vehicle Engineering for helping me by facilitating contact between different organizations. My gratefulness also extend to Metahra Sugar Factory which providing me ethanol that used in the experimental investigation. On the last my appreciation goes to Dilla University, Mechanical and Automotive Engineering Department for providing engine that used in experimental investigation and technical support.

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ACRONYMS AND SYMBOLS

ΔU	change of internal energy
AFR	air fuel ratio
BDC	bottom dead center
b_{sfc}	brake specific fuel consumption
CAI	controlled auto ignition
CC	Centimeter cube
CO	carbon monoxide
CO_2	carbon dioxide
EC	engine capacity
EEV	environmentally enhanced vehicle
GEG	gasoline equivalent gallon
HC	hydrocarbon
I_{mep}	indicated mean effective pressure
LEL	lower explosive limit
LHV	lower heating value
m_a	mass of air
m_f	mass of fuel
m_m	mass of air fuel charged to cylinder
MOWE	ministry of water and energy
M_{res}	mass of residual
NIOSH	National institute for occupational safety and health

NO_X	nitrogen oxide
NREL	National renewable energy laboratory
P	pressure
P_b	brake power
P_i	indicated power
Q	heat added
R_c	compression ratio
rpm	revolution per minute
T	temperature
T_b	brake torque
TDC	top dead center
TJ	tera joules
UEL	upper explosive limit
V	volume
V_{cl}	clearance volume
V_D	displacement volume
W	work done
Z_C	number of cylinder

ABSTRACT

The main objective of this thesis is to study thermodynamic analysis and performance evaluation of spark ignition (SI) engines using gasoline-ethanol blend fuel (E15). In this work the thermodynamic analysis of actual engine cycle with ethanol gasoline blend fuel (E15) was done. Depending on the thermodynamic analysis which can be the mathematical model was developed using c++ computer programming language. So the performance parameter such as brake power, brake torque, thermal efficiency, break mean effective pressure, combustion efficiency, brake specific fuel consumption and emission was analyzed. As a result the brake power torque for E15 is increased by 1.6% from E0. But the fuel consumption decreased by 2.4%, 4.35% and 3.15% at lower speed, maximum power and maximum speed respectively for E15 than E0. This program is versatile that can be functional for every type of engine and any type of blend proportion. As a result the brake power, brake torque, combustion efficiency, thermal efficiency and break mean effective pressure are higher for E15 than E0. The brake specific fuel consumption is lower for E15 than E0. The molar fraction emission of CO_2, CO and water vapor is less for E15 than E0 but molar fraction of N_2 is higher for E15 than E0. Experimental investigation was done on the real engine and the computer c++ program was validated and proved. In experimental investigation the brake power and brake torque for E15 is increased by 2.96% and 1.6% respectively. The minimum fuel consumption for E15 is lower by 1.3% than E0 at 2308rpm, higher for E15 by 1.7% than E0 at maximum brake power (2950rpm) and higher for E15 by 2.5% than E0 at maximum speed (3440rpm). The deviation between result of developed computer c++ program and experimental investigation was less than 5% for all analyzed engine parameter.

Key Words: Alternative fuels, Ethanol gasoline blends, thermodynamic analysis, C^{++}, SI engine

CHAPTER ONE

1. INTRODUCTION

The prices of petroleum products are generally on an increasing trend and consequently affecting the general cost of living. This continuous increase has resulted in the prices of some products tripling in the last decade. In some communities, in the developing nations these products are not only expensive but they are not readily available because of the poor road infrastructure necessary for their distribution. The consumption of petroleum products also has other inconveniencies such as environmental pollution and the emission of greenhouse gasses generally believed to be responsible for global warming. The general trend all over the world now is to reduce the over dependence on petroleum products so as to help reduce the effects of global warming. Other possible advantages of abandoning petroleum products is the fact that alternative sources can be produced from renewable resources that are available almost everywhere that there is life. This avoids the employment of heavy infrastructure for long distance transportation and distribution.

1.1. Background and Justification

Fuel additives are very important, since many of these additives can be added to fuel in order to improve its efficiency and its performance (Pandey and Gupta, 2016). Ethanol - gasoline blend with percentage volume of ethanol up to 15% can be used in SI engine without any modifications. The presence of oxygen and high octane number were significant features of using ethanol-gasoline blends.

The fuel consumption and specific fuel consumption showed a marginal decreasing with the use of ethanol– gasoline blends (Mostafa et al., 2017). Adding ethanol to gasoline will lead to a leaner better combustion. It was experimentally demonstrated that adding 5-15% ethanol to the blends led to an increase in the engine brake power, torque and brake thermal efficiency, volumetric efficiency and decreases the brake specific fuel consumption (Yusuf et al., 2010).

When engine was fueled with methanol gasoline blend, engine performance parameters such as brake torque, brake power, brake thermal efficiency, volumetric efficiency increases with increasing methanol amount in the blended fuel while bsfc and equivalence air-fuel ratio decreased (Shayan et al., 2011).

As the percentage of ethanol increases the emission characteristics improved. It is observed that the emission values of the HC and CO decreased when compared with that of pure gasoline. The brake thermal efficiency increases with the increase in the percentage of ethanol; E5 and E10 gave the best result for all measured parameters at a constant speed of 2000 rpm, thus ethanol may be used as an additive for gasoline. Brake thermal efficiency increased as the volume percentage of ethanol fuel is increased in the mixture (saikrishnm et al., 2016). CO, HC and NO_x emissions decreased with ethanol blends due to leaner air fuel mixtures. Catalytic converter showed reduction in all three emissions .The reduction percentage increased with ethanol percentage in the blend. A small increase in NOx emissions was noticed with catalytic converter (Raja et al., 2015).

1.2. Statement of Problem

The reserves of petroleum-based fuels are directly correlated with the increasing demands of humankind for energy production. With growing world populations, industries, vehicles, and equipment, energy demand leads to the search for a substitute for petroleum fuels which can accommodate to the need of people today. Considering the current global economic crisis, the curiosity in alternative fuels is extremely high.

Alternative fuels used as substitute petroleum-based fuels must be produced from renewable sources, and should be devised to use this fuel without bringing any modifications in the geometry of the engine. Alcohols have provided an answer to this problem. Ethanol is thought to be the most fitting fuel for spark ignition engines. Researchers have generated a few alternative and cost-effectively viable fuels which are also environment friendly.

Ethiopia is also one of the country which totally import petroleum from other country and high demand of energy is observing from day to day. As a result the country's economy highly depend on the fuel price and vulnerable to it which shocked by its fluctuation.

According to the Ministry of Water, Irrigation and Electricity the country has a plan to top up an ethanol blended oil to the local market to reach E25 (a 25% ethanol content blended in Benzene and biodiesel) (Melas Teka.2007). Due this government formulate policy to blend locally produced biofuel with petrol and a number of researcher recommend this blend fuel to save hard currency and minimize the emissions of greenhouse gases. Since 2009, the country has been provided a 5% ethanol and 95% benzene blended (technically known as E5) for the market in which a Sudanese owned oil company, Nile Petrol, was a sole agent that was carrying out the blending process and distributing for all local oil station. But now the Ministry of Water, Irrigation and Electricity (MoWIE) declared that Oil Libya is awarded the task of blending the 10% ethanol-Benzene blend (https://www.ezega.com). But a number of researchers conclude that, using E15 (15% ethanol and 85% gasoline) is possible without engine modification. For this reason E15 was selected to execute the work on it.

The way followed by the previous researcher to know the performance of engine and emissions of greenhouse is time consuming and need professional on the area. Despite a number of work is done on this area in other countries there is no common conclusion which is a basic route creating confusion to use the result of their experiment for practical operation and not easy to rely on it. But if the thermodynamic analysis of blend fuel is done and the computer program is developed for it, one can determine the engine performance and emissions rapidly and easily for every blend proportion. So this problem can be solved after completion of this thesis.

1.3. Objectives

1.3.1. General objective

The main aim of the work is to conduct thermodynamic performance and emission analysis of ethanol-gasoline blend fuel (E15) on spark ignition engine.

1.3.2. Specific objectives

The specific objectives of this work are listed as follow:

- To develop computer C++ program that can rapidly and easily determine engine performance and emissions for every amount of blend proportion.
- To validate the program using the practical data.
- To determine Engine performance such as, brake power, brake thermal efficiency, volumetric efficiency, brake torque, brake mean effective pressure and brake specific fuel consumption on different proportions of blend.
- To determine the amount of emission for all selected blend proportions by including combustion products like CO_2, H_2O, O_2, N_2, HC, CO, N_2.

1.4. Significance of the Study

The future researcher can also get benefit while determining the performance of engine and emissions on the actual or real engine by using the computer c++ program developed in this research.

Generally the automobile companies, users and the country as a whole can be benefited from this thesis.

1.5. Scope

This project was studied the thermodynamic analysis of engine performance and emissions of different proportions of blend fuel (ethanol-gasoline) by mathematical model and develop the computer c++ program. The experimental investigation was done on real engine to validate or prove the result of computer c++ program. All the combustion products were not considered while the emission analysis was done. This research work did not include the effect of different proportions of blend fuel on physical property of engine parts and accessories.

CHAPTER TWO

2. LITERATURE REVIEW

2.1. Biofuel

It is defined as liquid fuels produced from biomass; it excludes treatments of solid biomass as a source of energy. It focuses on ethanol and biodiesel based fuels on current available or near commercially developed technologies. The two most important biofuels are ethanol and biodiesel (Melis Teka. (2007).

2.2. Ethanol

Ethanol is manufactured from microbial conversion of biomass materials through fermentation. Ethanol contains 35 percent oxygen. The production process consists of conversion of biomass to fermentable sugars, fermentation of sugars to ethanol, and the separation and purification of the ethanol. Ethanol (ethyl alcohol, grain alcohol) is a clear, colorless liquid with a characteristic, agreeable odor. In dilute aqueous solution, it has a somewhat sweet flavor, but in more concentrated solutions it has a burning taste. Ethanol, CH_3CH_2OH, is an alcohol, a group of chemical compounds whose molecules contain a hydroxyl group, –OH, bonded to a carbon atom (Basile.,et al 2018).

Fermentation initially produces ethanol, and separation and purification of ethanol. Fermentation initial produces ethanol containing a substantial amount of water. Distillation removes the majority of water to yield about 95 percent purity ethanol, the balance being water. This mixture is called hydrous ethanol. If the remaining water is removed in a further process, the ethanol is called anhydrous ethanol and suitable for blending into gasoline. Ethanol is "denatured" prior to leaving the plant to make it unfit for human consumption by addition of small amount of products such as gasoline. (Basile., et al 2018).

2.3. Gasoline

Gasoline or gas for short (American English), or petrol (British English), is a transparent, petroleum-derived liquid that is used primarily as a fuel in internal combustion engines. It consists mostly of organic compounds obtained by the fractional distillation of petroleum, enhanced with a variety of additives. On average, a 42-gallon barrel of crude oil (159 L) yields about 19 US gallons (72 L) of gasoline when processed in an oil refinery, though this varies based on the crude oil source's assay.

The characteristic of a particular gasoline blend to resist igniting too early (which causes knocking and reduces efficiency in reciprocating engines) is measured by its octane rating. Gasoline is produced in several grades of octane rating. Tetra ethyl lead and other lead compounds are no longer used in most areas to regulate and increase octane-rating, but many other additives are put into gasoline to improve its chemical stability, control corrosiveness and provide fuel system 'cleaning,' and determine performance characteristics under intended use. Sometimes, gasoline also contains ethanol as an alternative fuel, for economic, political or environmental reasons.

Gasoline used in internal combustion engines has a significant impact on the environment, both in local effects (e.g., smog) and in global effects (e.g., effect on the climate). Gasoline may also enter the environment un combusted, as liquid and as vapors, from leakage and handling during production, transport and delivery, from storage tanks, from spills, etc

2.3.1. Comparison of gasoline and ethanol

Table 2.1. Compression of gasoline and ethanol

Characteristics	Gasoline	Ethanol
Molecular formula	C_4 - C_{12}	C_2H_5OH
Molecular weight	95-120	46
Oxygen content (%)	0	34.8%
Density (kg/m3)	740	785
LHV (MJ/kg)	44.3	26.7
Octane number	>90	108
Auto-ignition temp. (°C)	228-470	425
Stoichiometric A/F ratio	14.8	9.00
Latent heat of vapor. (kJ/kg)	305	840
Flash Point (°F)	-45	55
Specific Gravity	0.72–0.76	0.79
Vapor Density	3–4	1.49
Vapor Pressure (mmHg)	38–300	44
Boiling Point (°F)	100–400	173
Flammable Range (LEL–UEL) (%)	1.4–7.6	3.3–19
Conductivity	None	Yes
Smoke Character	Black	Slight to none
Toxicity		Lower than gasoline
Solubility	None	Highly

Source: Elfasakhan. (2015) and Zarihan et al., (2017)

Table 2.2. Tested properties of blend fuel

Fuel blend	Density, kg/L@ 15.6 °C	API gravity, dg.	Kinematic viscosity mm2/s@ 30 °C	Flash Point °C	Fire Point , °C	Cloud Point, °C	Heat of Combustion , MJ/L	Octane number
E0	0.74	59.53	0.4872	_	25	-22	34.84	93.2
E10	0.7396	57.1	0.5383	_	29	> 8	33.19	97.1
E15	0.7495	57.09	0.5619	_	29.1	> 8	32.91	98.6
E20	0.7541	55.95	0.6007	29.2	30	> 8	32.43	100.4
E25	0.7571	55.21	0.638	30	32	> 8	31.7	99.5
E30	0.7613	54.3	0.6614	29.2	30.2	> 8	31.53	102.5
E35	0.7653	53.5	0.6914	31	32	> 8	30.92	104.1

Source: Sharma, (2015)

2.3.2. Characteristics of Ethanol-Blended Fuels

Blending ethanol with gasoline has multiple effects. Ethanol increases the heat output of the unleaded gasoline, which produces more complete combustion resulting in slightly lower emissions from unburned hydrocarbons. The higher the concentrations of ethanol, the more the fuel has polar solvent-type characteristics with corresponding effects on conducting fire suppression operations. However, even at high concentrations of ethanol, minimal amounts of water will draw the ethanol out of the blend away from the gasoline. Ethanol and gasoline are very similar in specific gravity. The two differing fuels mix readily with minimal agitation, but the blend is more of a suspension than a true solution. Ethanol has a greater affinity for water than it does for gasoline. Over time, without agitation, gasoline will be found floating on a layer of an ethanol/water solution. The resulting ethanol/water solution is still flammable since the concentration of ethanol is still fairly rich. Phase separation can occur in fuel storage systems where water is known to be present. Blending these fuels together alters the physical and chemical characteristics of the original fuels. However, the resulting changes may be unnoticeable to emergency responders. One of the noticeable differences in the blended fuel versus unblended gasoline is the visual difference of the smoke and flame characteristics. The higher the content of ethanol, the less visible the black smoke content and orange flame production. These characteristics may be masked by other substrates that may also be burning such as vehicle tires. Another noticeable difference of ethanol-blended fuels under fire conditions is that when foam or water has been flowed on the burning product, the gasoline will tend to burn off first, eventually leaving the less volatile ethanol/water solution which may have no visible flame or smoke.

Ethanol is a polar solvent that is simultaneously water-soluble and flammable. Creating a blend of gasoline and ethanol results in a chemical change that can easily go unnoticed by emergency responders. Knowing that the ethanol will be the last fuel to burn and that it may burn without visible smoke or flame is important in determining an approach to take in dealing with ethanol-involved incidents (Thakur.2017).

2.4. Summary of the Main Ethanol Blends Used Around the World

Several common ethanol fuel mixtures are in use around the world. The use of pure hydrous or anhydrous ethanol in internal combustion engines (ICEs) is only possible if the engines are designed or modified for that purpose, and used only in automobiles, light-duty trucks and motorcycles. Anhydrous ethanol can be blended with gasoline (petrol) for use in gasoline engines, but with high ethanol content only after minor engine modifications.

Ethanol fuel mixtures have "E" numbers which describe the percentage of ethanol fuel in the mixture by volume, for example, E85 is 85% anhydrous ethanol and 15% gasoline. Low-ethanol blends, from E5 to E25, are also known as gasohol, though internationally the most common use of the term refers to the E10 blend.

Blends of E10 or less are used in more than 20 countries around the world, led by the United States, where ethanol represented 10% of the U.S. gasoline fuel supply in 2011. Blends from E20 to E25 have been used in Brazil since the late 1970s. E85 is commonly used in the U.S. and Europe for flexible-fuel vehicles. Hydrous ethanol or E100 is used in Brazilian neat ethanol vehicles and flex-fuel light vehicles and hydrous E15 called hE15 for modern petrol cars in the Netherlands.

2.4.1. E10 or less

E10, a fuel mixture of 10% anhydrous ethanol and 90% gasoline sometimes called gasohol, can be used in the internal combustion engines of most modern automobiles and light-duty vehicles without need for any modification on the engine or fuel system. E10 blends are typically rated as being 2 to 3 octane numbers higher than regular gasoline and are approved for use in all new U.S. automobiles, and mandated in some areas for emissions and other reasons. The E10 blend and lower ethanol content mixtures have been used in several countries, and its use has been primarily driven by the several world energy shortages that have taken place since the 1973 oil crisis.

2.4.2. E15

Typical manufacturer's statement in the car owner's manual regarding the vehicle's capability of using up to E10.

E15 contains 15% ethanol and 85% gasoline. This is generally the highest ratio of ethanol to gasoline that is possible to use in vehicles recommended by some auto manufacturers to run on E10 in the US. This is due to ethanol's hydro philia and solvent power.

As a result of the Energy Independence and Security Act of 2007, which mandates an increase in renewable fuels for the transport sector, the U.S. Department of Energy began assessments for the feasibility of using intermediate ethanol blends in the existing vehicle fleet as a way to allow higher consumption of ethanol fuel. The National Renewable Energy Laboratory (NREL) conducted tests to evaluate the potential impacts of intermediate ethanol blends on legacy vehicles and other engines. In a preliminary report released in October 2008, the NREL presented the results of the first evaluations of the effects of E10, E15 and E20 gasoline blends on tailpipe and evaporative emissions, catalyst and engine durability, vehicle drive ability, engine operability, and vehicle and engine materials. This preliminary report found none of the vehicles displayed a malfunction indicator light as a result of the ethanol blend used; no fuel filter plugging symptoms were observed; no cold start problems were observed at 24 °C (75 °F) and 10 °C (50 °F) laboratory conditions; and as expected, computer technology available in newer model vehicles adapts to the higher octane causing lower emissions with greater horsepower and in some cases greater fuel economy seen in aircraft.

2.4.3. hE15

A 15% hydrous ethanol and 85% gasoline blend, hE15, has been introduced at public gas stations in the Netherlands since 2008. Ethanol fuel specifications worldwide traditionally dictate use of anhydrous ethanol (less than 1% water) for gasoline blending. This results in additional costs, energy usage and environmental impacts associated with the extra processing step required to dehydrate the hydrous ethanol produced via distillation (3.5-4.9 vol. % water) to meet the current anhydrous ethanol specifications. A patented discovery reveals hydrous ethanol can be effectively used in most ethanol/gasoline blending applications.

According to the Brazilian ANP specification, hydrous ethanol contains up to 4.9 vol. % water. In hE15, this would be up to 0.74 vol. % water in the overall mixture. Japanese and German scientific evidence revealed the water is an inhibitor for corrosion by ethanol.

2.4.4. E20, E25

E20 contains 20% ethanol and 80% gasoline, while E25 contains 25% ethanol. These blends have been widely used in Brazil since the late 1970s. As a response to the 1973 oil crisis, the Brazilian government made mandatory the blend of ethanol fuel with gasoline, fluctuating between 10% to 22% from 1976 until 1992. Due to this mandatory minimum gasoline blend, pure gasoline (E0) is no longer sold in Brazil. A federal law was passed in October 1993 establishing a mandatory blend of 22% anhydrous ethanol (E22) in the entire country. This law also authorized the Executive to set different percentages of ethanol within pre-established boundaries, and since 2003, these limits were fixed at a maximum of 25% (E25) and a minimum of 20% (E20) by volume. Since then, the government has set the percentage on the ethanol blend according to the results of the sugarcane harvest and ethanol production from sugarcane, resulting in blend variations even within the same year.

Since July 1, 2007, the mandatory blend was set at 25% of anhydrous ethanol (E25) by executive decree, and this has been the standard gasoline blend sold throughout Brazil most of the time as of 2011. However, as a result of a supply shortage and the resulting high ethanol fuel prices, in 2010, the government mandated a temporary 90-day blend reduction from E25 to E20 beginning February 1, 2010. As prices rose abruptly again due to supply shortages that took place again between the 2010 and 2011 harvest seasons, some ethanol had to be imported from the United States, and in April 2011, the government reduced the minimum mandatory blend to 18%, leaving the mandatory blend range between E18 and E25.

2.4.5. E70, E75

When the vapor pressure in the ethanol blend drops below 45 kPa, fuel ignition cannot be guaranteed on cold winter days, limiting the maximum ethanol blend percentage during the winter months to E75.

E70 contains 70% ethanol and 30% gasoline, while E75 contains 75% ethanol. These winter blends are used in the United States and Sweden for E85 flexible-fuel vehicles during the cold weather, but still sold at the pump labeled as E85. The seasonal reduction of the ethanol content to an E85 winter blend is mandated to avoid cold starting problems at low temperatures.

In the US, this seasonal reduction of the ethanol content to E70 applies only in cold regions, where temperatures fall below 32 °F (0 °C) during the winter. In Wyoming for example, E70 is sold as E85 from October to May. In Sweden, all E85 flexible-fuel vehicles use an E75 winter blend. This blend was introduced since the winter 2006-07 and E75 is used from November until March.

For temperatures below −15 °C (5 °F), all E85 flex vehicles require an engine block heater to avoid cold starting problems. The use of this device is also recommended for gasoline vehicles when temperatures drop below −23 °C (−9 °F). Another option when extreme cold weather is expected is to add more pure gasoline in the tank, thus reducing the ethanol content below the E70 winter blend, or simply not to use E85 during extreme low temperature spells.

2.4.6. E85

E85, a mixture of 85% ethanol and 15% gasoline, is generally the highest ethanol fuel mixture found in the United States and several European countries, particularly in Sweden, as this blend is the standard fuel for flexible-fuel vehicles. This mixture has an octane rating of 94-97, which is significantly lower than pure ethanol, but still higher than normal gasoline (87-93 octane, depending on country).

The 85% limit in the ethanol content was set to reduce ethanol emissions at low temperatures and to avoid cold starting problems during cold weather, at temperatures lower than 11 °C (52 °F). A further reduction in the ethanol content is used during the winter in regions where temperatures fall below 0 °C (32 °F) and this blend is called Winter E85, as the fuel is still sold under the E85 label. A winter blend of E70 is mandated in some regions in the US, while Sweden mandates E75. Some regions in the United States now allow E51 (51% ethanol, 49% gasoline) to be sold as E85 in the winter months.

As of October 2010, nearly 3,000 E85 fuel pumps were in Europe, led by Sweden with 1,699 filling stations. The United States had 3,354 public E85 fuel pumps located in 2,154 cities by August 2014, mostly concentrated in the Midwest.

Thailand introduced E85 fuel by the end of 2008, and by mid-2010, only four E85 filling stations were available, with plans to expand to 15 stations by 2012.

2.4.7. ED95

ED95 designates a blend of 95% ethanol and 5% ignition improver; it is used in modified diesel engines where high compression is used to ignite the fuel, as opposed to the operation of gasoline engines, where spark plugs are used. This fuel was developed by Swedish ethanol producer SEKAB. Because of the high ignition temperatures of pure ethanol, the addition of ignition improver is necessary for successful diesel engine operation. A diesel engine running on ethanol also has a higher compression ratio and an adapted fuel system.

This fuel has been used with success in many Swedish Scania buses since 1985, which has produced around 700 ethanol buses, more than 600 of them to Swedish cities, and more recently has also delivered ethanol buses for commercial service in Great Britain, Spain, Italy, Belgium, and Norway. As of June 2010 Stockholm has the largest ethanol ED95 bus fleet in the world.

As of 2010, the Swedish ED95 engine is in its third generation and already has complied with Euro 5 emission standards, without any kind of post-treatment of the exhaust gases. The ethanol-powered engine is also being certified as environmentally enhanced vehicle (EEV) in the Stockholm municipality.

2.4.8. E100

E100 is pure fuel, straight hydrous ethanol as an automotive fuel has been widely used in Brazil since the late 1970s for neat ethanol vehicles and more recently for flexible-fuel vehicles. The ethanol fuel used in Brazil is distilled close to the azeotrope mixture of 95.63% ethanol and 4.37% water (by weight) which is approximately 3.5% water by volume. The azeotrope is the highest concentration of ethanol that can be achieved by simple fractional distillation. The maximum water concentration according to the ANP specification is 4.9 vol. % (approximately 6.1 weight %). The E nomenclature is not adopted in Brazil, but hydrated ethanol can be tagged as E100, meaning it does not have any gasoline, because the water content is not an additive, but rather a residue from the distillation process. However, straight hydrous ethanol is also called E95 by some authors.

Source: (U.S.Department of Energy 2016).

2.5. Summary of Related Works

Mostafa et al., (2017) this study led to some conclusions, which could be deduced as follow: 1 burning ethanol-gasoline blend in SI engine improves the engine-generator set performance characteristics.2. Ethanol - gasoline blend with percentage volume of ethanol up to 15% can be used in SI engine without any modifications.3. The presence of oxygen and high octane number were significant features of using ethanol-gasoline blends.4. The fuel consumption and specific fuel consumption showed a marginal decreasing with the use of ethanol–gasoline blends.5. Ethanol addition to gasoline resulted in an increase in overall efficiency of the engine - generator set and exhaust gas temperature. For E15 blend, the overall efficiency and exhaust gas temperature at maximum load were increased by 14.88% and 3.85% respectively as compared to base gasoline.

Pikūnas et al., (2003) the engine performance and pollutant emission of a SI engine have been investigated by using ethanol–gasoline blended fuel E10 and pure gasoline. Experimental results indicated that when ethanol–gasoline blend is used, the engine power and fuel consumption of the engine slightly increase; CO emission decreases dramatically as a result of the leaning effect caused by the ethanol addition; HC emission decreases only in some engine working conditions; and CO_2 emission increases because of the improved combustion. In this study, researchers found that using ethanol–gasoline blend, CO emission may be reduced by 10–30%, while CO_2 emission increases by 5–10% depending on engine conditions. The engine power and specific fuel consumption increase approximately by 5% and 2–3%, respectively, in all working conditions.

Gaurav Sharma and Harvinder Lal. (2015) Ethanol's physical and chemical properties show that it can be used as an alternative fuel or additive in SI engine. Ethanol-gasoline blends show very effective results. In this study, four different blends E5, E10, E15 and E20 were run on SI engine for analyzing performances and emissions. It was found that E5 and E10 blend was most effective blend they improved performance and reduced emissions. It was concluded that E5 and E10 blend improved brake thermal efficiency, brake torque reduced brake specific fuel consumption as compared to gasoline. Further both blends also reduced emissions like CO_2, CO, HC and NO_x as compared to gasoline.

Joshi et al. (2015) among the various blends tested, E20 blend was found to have least exhaust and greenhouse gases emissions, also not compromising on engine or vehicle performance.

E20 offers the following property:

- Mechanical efficiency: 92.77%
- Thermal efficiency: 34%
- Mass of fuel consumed: 0.035 Kg
- Heat supplied by the fuel: 1415.05 KJ/min
- CO%: 4.22%
- HC%: 78 ppm
- Materials compatibility: There is no discernible corrosion in fuel-wetted metal parts such as fuel tanks, lines, pressure regulators, etc.
- Engine wear: There is no additional or unusual wear to that normally expected and there is no additional increase in wear metals or decrease in total base number (TBN) of the lubricating oil.
- Water tolerance: Quality of ethanol produced and stored should be extremely pure and the water content should be less than 1.25%. An ethanol compatible water detecting paste must be used to establish the water content of underground storage tanks (standard paste is not suitable) and the water content must be kept to a minimum and, older vehicles are more prone to suffer from phase separation when first fueled with ethanol–petrol blend; however, subsequent continuous use of ethanol–petrol blend prevents water accumulation within the fuel tank.

Kheiralla et al., (2014) the following conclusions could be drawn from this study: Ethanol-gasoline blends can be used as an alternative fuel for variable speed spark ignition up to 35% blends without engine modification. − Fuel properties of the tested ethanol-gasoline blends such as density and viscosity increased continuously and linearly with increasing percentage of ethanol while API gravity and heat value decreased with decreasing percentage of ethanol increase. Furthermore, cloud point, flash and fire points were found to be higher than pure gasoline fuel, while distillation curves proved to be lower. − The tested blends Octane rating based Research Octane Number (RON) increased continuously and linearly with the increasing percentage of ethanol.

Based on the results obtained during this study work it can be suggested that: Comprehensive and extensive testing on fuel properties, engine performance, combustion, and emissions characteristics of ethanol-gasoline blends on SI engines should be conducted for long time. Research collaboration should be undertaken with sugarcane industry, and GIAD Automotive Group regarding using ethanol as bio-fuel for SI engines.

Thakur et al., (2017) this study reveals a clear view of the progress made to improve the performance parameters of spark ignition engines using blends of gasoline-ethanol, gasoline with all other alcohol derivatives, and subsequent alternative fuels. Some key findings of this study are as follows:

- Ethanol blends in lower proportions showed improvement in engine torque and brake power as observed during study. Using E5, E10, E20 an increment of 2.31, 2.77 and 4.16% in engine brake power and 0.29, 0.59 and 4.77% in torque was noted.
- Brake specific fuel consumption increased for higher volume of ethanol content as observed. Brake specific fuel consumption increased by 5.17, 10, 20, 37, and 56% using E20, E25, E30, E75, and E100 respectively.
- Varying the compression ratio and operating the engine at various speed also affected the performance parameters of the spark ignition engine. Using E50 and E85 as fuel at compression ratio 10:1 and 11:1; increment of 20.3 and 45.6% and 16.15 and 36.4%, respectively, was observed for brake specific fuel consumption.

- Brake thermal efficiency was slightly on the high side when an ethanol-gasoline blend was used. Maximum brake thermal efficiency of 3.5% for E20, 2.5% for E10, and 6% for E40 was concluded.

Raja et al., (2015) Gasoline-ethanol blends were experimented at part-throttle operation of motorcycle engine, without modifying compression ratio, intake system, fuel system and ignition timing, without and with catalytic converter. The following conclusions are derived from the results.

Blends up to E15 showed smooth and satisfactory engine operation and E20 blend resulted in choking and stalling at higher engine speeds.

Brake specific fuel consumption was higher for blends due to lower heating value of the blends compared to neat gasoline and correspondingly there was reduction in brake thermal efficiency. Among the blends, E5 and E10 were closer to neat gasoline. Volumetric efficiency and excess air factor increased with ethanol percentage in the blend due to higher heat of vaporization and oxygen in ethanol.

Exhaust temperature decreased with blends. This is because of lower engine temperature and higher excess air in the air-fuel mixture. CO, HC and NO_x emissions decreased with ethanol blends due to leaner air fuel mixtures. Catalytic converter showed reduction in all three emissions. The reduction percentage increased with ethanol percentage in the blend. A small increase in NO_x emissions was noticed with catalytic converter. As E5 and E10 blends show equivalent performance characteristics and superior emission behavior compared to neat gasoline in part-throttle conditions, they can be used as substitute fuels for neat gasoline in motorcycle engine with no modifications.

HuuTruyen et al., (2012) the performance and pollutant emission of the in-use carbureted and fuel injected cars fueled by ethanol/gasoline blends were studied. The results indicated that effects of the ethanol/gasoline fuels on cars were not the same if the cars equipped with different fuel systems. For the carbureted car, using E10, E15 and E20 could increase power, reduce fuel consumption, and decrease significantly HC and CO emissions, whereas increase NO_x emissions. For the fuel injected car, these blends nearly did not affect power and fuel consumption. HC and CO emissions were reduced for E10 but increased for E15 and E20, while NO_x was higher for E10 and E15 but lower for E20 as compared to base gasoline RON92. In this study, it found that utilization of E10 possibly gave benefit for both of the cars.

Ozsezen et al.,(2012) In this study, it was seen that the wheel power with the use of alcohol- gasoline blends slightly increased compared to pure gasoline. When the vehicle was fueled with alcohol-gasoline blends, more fuel was consumed to achieve the same wheel power with pure gasoline, and it caused an increase in the fuel consumption. Generally, the alcohol-gasoline blends at all vehicle speeds provided slightly higher combustion efficiency relative to pure gasoline. The best combustion efficiency was obtained with the use of M5 at 40 km h1 and 80 km h1 vehicle speeds, with the use of E10 at 60 km h1 and 100 km h1 vehicle speeds. The combustion efficiency and the equivalence ratio increased with increasing vehicle speed. For all test fuels, a decreasing in HC emission and an increasing in CO emission took place with the increasing vehicle speed. Generally, a slightly increasing in CO emissions and a decreasing in HC emissions occurred when using alcohol-gasoline blends. The decreases in unburned HC emission levels with use of the alcohol-gasoline blends led to increase in the combustion efficiency. A table trend in CO_2 emissions was not observed, but an increase and a decrease in NO_x emissions was measured at the vehicle speed of 40 km h1 and 100 km h1, respectively, with the use of alcohol-gasoline blends.

Tangka et al., (2011) Bio-ethanol production from the chosen crops was feasible. Bio-ethanol production is highest after three days of fermentation using natural methods. The highest bio-ethanol production came from maize although the values got were lower than those obtained in the United States. Addition of ethanol to gasoline reduces the energy density of the volume. Engine speed increases and engine torque decreases. However overall break power increases. There is loss of fuel economy and consequently mileage with blended rations of bio-ethanol gasoline because the alcohol has a lower energy value than gasoline. Blended fuel is not yet a good option for less developed countries that are not manufacturing resistant parts. Anti-fuel properties of bio-ethanol blended fuels calls for serious engine modifications to avoid knock and unwanted emissions. These modifications include the advancing of engine timing, provision of airtight conduit lines, increasing the compression ratio and roughening of the piston head.

Yusuf et al., (2014) adding ethanol to gasoline will lead to a leaner better combustion. It was experimentally demonstrated that adding 5-15% ethanol to the blends led to an increase in the engine brake power, torque and brake thermal efficiency, volumetric efficiency and decreases the brake specific fuel consumption. The lean combustion improves the completeness of combustion and therefore the CO emission is expected to be decreased. The oxygen enrichment generated from ethanol increased the oxygen ratio in the charge and lead to lean combustion.

The CO_2 emission increased because of the improvement of the combustion and the chemical properties of Ethanol.

Unburned HC is a product of incomplete combustion which is related to A/F ratio. It can be concluded that that adding ethanol to the blends will reduces the HC emission because of oxygen enhancement.

When the combustion process is closer to stoichiometric, flame temperature increases, therefore, NO_x formation is expected to be increased.

The results obtained with presented theoretical model are in acceptable agreement with those experimental ones. An agreement of 11% was determined between experimental and theoretical results.

Kumar et al. (2016) in this study, the effect of unleaded gasoline and unleaded gasoline blended with 5%, 10 % and 15% of ethanol on the performance of a spark-ignition engine were experimentally investigated. Performance tests were conducted for fuel consumption, engine torque, brake power, brake thermal efficiency and brake specific fuel consumption, using unleaded gasoline-ethanol blends with different percentage of fuel at wide open throttle opening position and variable engine speed ranging from 900 to 3000 rpm. The results showed that blending unleaded gasoline with ethanol increase the brake power, torque, brake thermal efficiency and fuel consumption, while it decreases the brake specific fuel consumption. The 15 vol% ethanol in the fuel blend gave the best results for all measured parameters at all engine speed.

Nallamothu1 et al., (2013)

In this research work, procedures of measuring fuels have been used to blend the ethanol produced from sugar cane with gasoline and base fuels used for the experiment.

Properties of ethanol-gasoline blended and base fuel were first examined by the standard ASTM test methods D86, D130, ES626:2008 (ANNEXB), ES640:2001 (ANNEXA), D323, D1298 and fuel was blended in different volume rates E0, E5and E10. Moreover, the experimental comparative performance evaluation are tested and evaluated at 8:1 compression ratios. The performance and exhaust emission were carried out on gasoline engines by using TD43F variable compression engine test rig and Exhaust gas analyzer 5000 and the following test results were summarized. Best performance with maximum reduction is 2.9% Pb is obtained for all samples for the compression ratio of 8:1 at speed of2000 rpm. Blending increases ηb for compression ratio of 8:1. Compression ratio of 8:1 is recommended to use E10.

Shayan et al.,(2011) In this study, it was seen that when engine was fueled with methanol–gasoline blend, engine performance parameters such as brake torque, brake power, brake thermal efficiency, volumetric efficiency increases with increasing methanol amount in the blended fuel while bsfc and equivalence air-fuel ratio decreased. Since the latent heat of evaporation of ethanol is higher than that of gasoline, during compression process, the fuels containing methanol will absorb more heat from combustion chamber and eventually, the pressure of the combustion chamber will be decreased accordingly. Relying on above statements, during the compression process, the pressure of such combustion chamber will be decreased compared with when pure gasoline is used in combustion. On the other hand, due to presence of oxygen entered the combustion chamber during expansion process and after combustion of fuel and upon improvement of combustion, the pressure of the expansion process will be increased as well. Hence, the work of compression process, which is a negative work, will be decreased and that of the expansion process that is a positive work, will be increased for that reason.

Consequently, upon increase of enclosed area in the pressure-volume curve, the work done by the engine will be increased in case of use of the fuel containing methanol and finally, the indicated mean effective pressure will be increased as well. Therefore, brake power will be increased. Using methanol–gasoline blends lead to a significant reduction in exhaust emissions by about 24.9% and 23.7% of the mean average values of HC and CO emissions, respectively, for all engine speeds. On the other hand CO_2 and NO_x emissions increases by about 7.5% and17.5% respectively.

Zhang et al., (2013) in this paper, CAI combustion has been demonstrated on a poppet valve DI gasoline engine operating in the 2-stroke cycle. Gasoline and its mixture with ethanol, E15 and E85, were used and their ranges of CAI operations were determined as a function of the engine speed and load. Their combustion and heat release characteristics, emissions, and their combustion and thermodynamic efficiencies were determined and analyzed.

The results show that:

- 2-stroke CAI combustion operation can be achieved over a wide range of engine speed and load conditions, including idle operation that could not be achieved with 4- stroke operations.
- The presence of ethanol allowed CAI combustion to be extended to higher load conditions. InthecaseofE85themaximum IMEP of 8.4 bar was obtained at 800 rpm, significantly higher than the 4-stroke equivalent. Further improvement in the high load range at higher engine speeds can be achieved with a faster camless system or mechanical cam shafts.
- CO, HC and NO_x emissions are significantly reduced by injecting ethanol blended fuels. E85 has greater effect on the emission reduction than E15.

- Both combustion efficiency and thermodynamic efficiency are improved by the presence of ethanol because of the optimum combustion phasing and lower heat loss.
- E85 improved indicate fuel conversion efficiency by over 5% at 2000 rpm.
- The above results were obtained by keeping the relative air to fuel ratio to stoichiometric in the engine exhaust. In addition, future improvement in the fuel injector and combustion chamber design are expected to further reduce the HC and CO emissions and improve the thermal efficiency and indicated fuel conversion efficiency, making the 2-stroke poppet valve engine a viable low emission and high efficiency IC engine for automotive applications.

Wang et al., (2015) the focus of this study was to evaluate the effects of hydrous ethanol gasoline on the combustion and emission characteristics on a port injection gasoline engine. Based on the experimental results, the conclusions are as follows:

- Compared with E0, E10W showed higher peak in-cylinder pressure at high load. Increases in peak heat rise rates were observed for E10W fuel at all the operating conditions. Besides, the usage of E10W increased NO_X emissions at a wide load range. However, at low load conditions, E10W reduced HC, missions, while CO_2 emissions were not significantly affected at higher operating points.
- The practical consequence of burning E10W fuel showed a better combustion process, especially at higher loads. In addition, E10W fuel had lower NO_X emissions than E10 at all operating conditions and lower HC, CO and CO_2 emissions significantly and simultaneously than E0 at low load conditions. From the above, it can be concluded that E10W fuel can be regarded as a potential alternative fuel for gasoline engine applications.

CHAPTER THREE

3. MATERIALS AND METHODS

In this chapter the required data was collected, the theoretical thermodynamic analysis of real engine cycle was done, depending on the thermodynamic analysis and mathematical model computer c++ program was written and the written computer program is compiled and run. To validate the written computer c++ program experimental analysis conducted on the selected real engine.

3.1. Experimental Site

The experimental analysis was conducted at Dilla University. Dilla is a market town and separate woreda in southern Ethiopia. The administrative center of the Gedeo Zone in the Southern Nations, Nationalities, and Peoples Region (SNNPR), it is located on the main road from Addis Ababa to Nairobi. The town has a longitude and latitude of 6°24′30″N 38°18′30″ECoordinates: 6°24′30″N 38°18′30″E, with an elevation of 1570 meters above sea level. It was part of Wenagoworeda and is currently surrounded by DilaZuriaworeda.

3.2. Collection of Required Data

To validate the output of c++ computer program written in this work one engines was selected and its specification is summarized as follow.in addition to engine specifications the both fuel means gasoline and ethanol physical and chemical characteristics was collected and summarized in chapter 2 (two) of this paper.

Not only each characteristic's the blend's characteristics also summarized in chapter 2(two) of this paper.

3.2.1. Engine specifications

Table 3.1. Selected engine specifications

Engine Specification	
Engine type: -	Air-cooled 1 cylinder, 4-stroke petrol engine
Dimensions: -	L x W x H: 500 x 345 x 410 mm
Weight: approx.	34 kg
Compression: -	8.5:1
Bore: -	89 mm
Stroke: -	63 mm
Connecting rod length: -	110mm
Crank throw: -	31.5 mm
Max. Power: -	8.2 kW
Max. Speed: -	3600 rpm
Max. Torque: -	24.6 Nm at 2200 rpm
Oil capacity: -	1.4 liters
Ignition: -	Magneto ignition
Sound level(distanc)-	96dB(A)
Intake valve open bTDC	15°
Intake valve close aBDC	50°
Start of combustion bTDC	15°
Duration of combustion	55°
exhaust valve open bBDC	50°
exhaust valve close aTDC	15°

Figure 3.1. Engine used for experimental analysis

3.2.2. Technical Information of exhaust gas analyzer

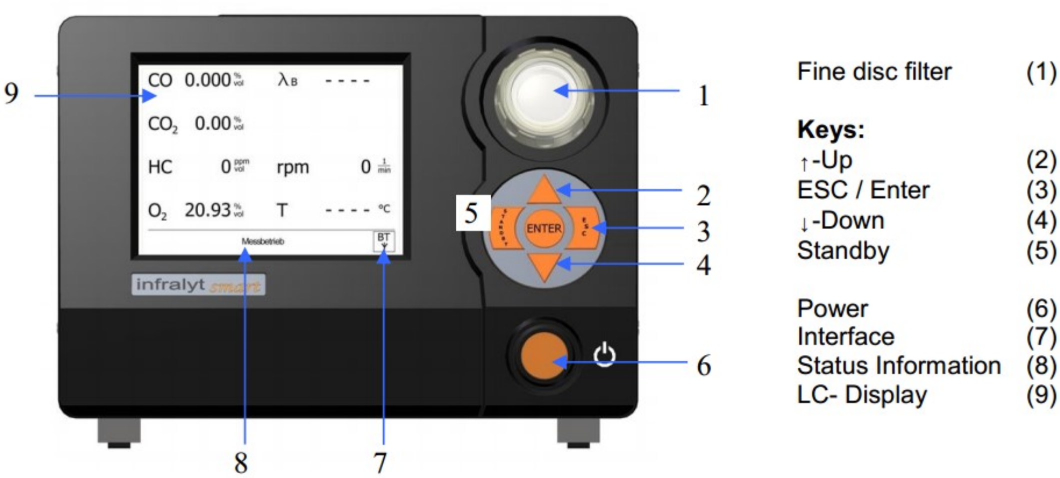


Figure 3.2. Infralyt Smart Exhaust Gas Analyzer

3.2.2.1. Measuring Principle

The measuring principle of CO, CO_2 and HC is based up on the interference correlation filter principle. Infrared light is radiated through the measuring chamber and is detected by sensor that operates an interference correlation filter. The Sensor signals are processed by microcontroller directly. Oxygen and optional nitrogen monoxide is measured using chemical sensors.

3.2.2.2. Technical Data

Table 3.2. Environmental requirements for exhaust gas analyzer

Environment parameters	requirements
Ambient temperature	5 - 45°C
Ambient air pressure	860 - 1060 hPa(hecto pascal)
Max. temperature/humidity ratio	35/95

Table 3.3 technical data of exhaust gas analyzer

Gas concentration resolution	4 Digits
Warming up time	30 s
Automatic zero point check	Every 60 min, 3,2°C temperature change or negative measuring values
Zero Gas	Fresh air, free of HC, CO and CO2.
Response time	IR-Components:T95<15s O2: T95 <60 s @ Flow rate >90 dm3/h
Position of use	horizontal, required because of the operation of the condensate separator; Slope <8 deg
Dimensions	Width 258mm,Depth 330mm,Heigh 203 mm (including foots + printer)
Mass	ca. 6,5 kg
Degree of protection	IP 20

Table 3.4. Functional requirements of exhaust gas analyzer

Measuring components und ranges	CO 0-10.00%vol, CO2 0-20.00% vol, HC 0-2500ppmvol(C_6H_{14}),O2 0-22.00%vol NO 0-5000 ppm vol (available as an option) NOx 0-5000 ppm vol (available as an option)
Measuring components absolute tolerances	CO ± 0.03 % vol, CO2 ± 0,5 % vol HC ± 10 ppmvol(C6H14),O2 ± 0.1 % vol NO ± 2 ppm vol (available as an option) NOx ±25 ppm vol (available as an option)
Measuring components relative tolerances	± 5 %

3.3. Mathematical Model for Actual Engine Cycle

For one cylinder: (Willard W. Pulkrabek) **and** (AKolchin and R Demidw)

1) **Displacement volume (V_D):**

$$V_D = \frac{E_C}{N_C} \quad \text{......3-1}$$

Where, E_C= engine capacity

Z_c=number of cylinder

V_D= 392cm^3

2) **Clearance volume (V_C)**

$$R_C = \frac{V_1}{V_2} = \left(\frac{V_D + V_C}{V_C}\right) \quad \text{......3-2}$$

Where,

V_1 =volume of cylinder above BDC

V_2=volume of cylinder above TDC

$$8.5 = \left(\frac{392\ cm^3 + V_C}{V_C}\right)$$

$$V_C = 52.26\ cm^3$$

$$V_T = V_D + V_C$$

$$= 392 + 52.26$$

$$= 444.26cm^3$$

State1

The intake stroke of the Otto cycle starts with the piston at TDC and is a constant pressure process at an inlet pressure of one atmosphere. The temperature of the air during the inlet stroke is increased as the air passes through the hot intake manifold. The temperature at point TDC will generally be on the order of 25°to 35°C hotter than the surrounding air temperature (Pulkrabek.1988).

$$T_1 = 50^oc = 323$$

$$P_1 = 100kpa = 1atm = 1bar$$

$$V_1 = V_D + V_C = 392 + 52.26 = 444.26cm^3$$

Mass of gas mixture in cylinder can be calculated at state 1. The mass with in the cylinder will then remain the same for the entire cycle.

In air-standard cycles, air is considered an ideal gas such that the following ideal gas relationships can be used in actual engine cycle.

$$PV = Z \cdot m_m \cdot R \cdot T \text{ .... 3-3}$$

Where, Z= 0.99 for intake and 0.98 for exhaust Smith (2002).

$$P_1 \cdot V_1 = Z \cdot m_m \cdot R \cdot T_1$$

$$m_m = \frac{P_1 \cdot V_1}{R \cdot T_1 \cdot Z} = \frac{100 \text{ kpa} \cdot 0.000444 \text{ m}^3}{0.99 \times 323 \times 0.287\text{kJ}/_{\text{kg\_k}}}$$

$$= 0.000483kg$$

Where,

P_1 is atmospheric pressure, T_1 is ambient temperature, V_1 is total volume, R is universal gas constant and m_m is fresh mass charged to cylinder

There is residual mass in cylinder $M_{residual}$ which is from 2% to 8% of m_m (Akolchin and Demidv .1984).

Let take 5% residual mass which is the average.

$$M_{res} = 5/100 * m_m$$

$$= 0.05*0.000483Kg$$

$$= 0.000024kg$$

State 2

The compression stroke is isentropic. The following ideal gas relationships can be used to find pressure and temperature.

The compression stroke in the real engine is different from ideal one. This means the intake valve is not close at BDC rather after the BDC.so compression can be classified to two.

1. Polytropic compression

   This is the stage from BDC to the point intake valve close after BDC.

2. Polytropic effective compression

   This is the stage from intake valve close after BDC to the point at start of combustion.

$$P_2 = P_1 \times (R_C)^K \quad \text{3-4}$$

$= 100kpa\ (8.5)^{1.35}$

$= 1797.69kpa$

$$T_2 = T_1 \times (R_C)^{K-1} \quad \text{3-5}$$

$= 323k\ (8.5)^{1.35-1}$

$= 683.12k$

$$M_{m1} = m_m + M_{res} \quad \text{3-6}$$

$= 0.000483kg + 0.000024kg$

$= 0.00051kg$

Where, k is compression index and R_c is compression ratio.

The type of fuel used in this work is blend of alcohol-gasoline (E15).

Table 3.5 mass and molar fraction blend proportion

Fuel	Mass, m(kg)	Molecular weight, M	Moles N = m/M	Mole fraction
C_2H_5OH	0.15	46	0.00326	0.299
C_8H_{15}	0.85	111	0.00765	0.701
summation	1	157	0.01091	1

$$\text{Mole fraction} = \frac{N_i}{\int N_i} \quad \text{3-7}$$

$0.299C_2H_5OH + 0.701C_8H_{15} + 9.13(O_2 + 3.76N_2) = 6.206CO_2 + 6.156H_2O + 33.49N_2$

$$AFR = \frac{m_a}{m_f} = \frac{(9.13\times(32+3.76\times28))}{(0.3\times46)+(0.7\times111)}$$

$$=13.69$$

The mass of gas mixture m_m in the cylinder made up of air m_a, fuel m_f (mass of ethanol + mass of gasoline).

$$\text{Mass of air } m_a = \left(\frac{AFR}{AFR+1}\right)*(1-XR)*(M_{m1}) \text{ ....3-8}$$

Where, XR is percentage of residual

$$= (13.69/14.69)*(1-0.05)*(0.00051kg)$$

$$= 0.00045kg$$

$$\text{Mass of fuel } m_f = \left(\frac{1}{AFR+1}\right)*(1-XR)*(M_{m1}) \text{ ....3-9}$$

$$= (1/14.69)*0.95*(0.00051kg)$$

$$= 0.000033kg$$

State 3

Heat addition from hydrocarbon fuel.

$$Q_{in} = m_f\times Q_{LHV}\times\eta_c \text{ ....3-10}$$

Where, $m_f = 0.000033kg$

$\eta_c = 90\%$

$$Q_{LHV}\text{ blend fuel} = 0.15\,(Q_{LHV}\text{ alcohol}) + 0.85(Q_{LHV}\text{ gasoline}) \text{ ....3-11}$$

$$= 0.15(27MJ/kg) + 0.85(44MJ/kg)$$

$$= (4.05+37.4)MJ/kg$$

$$= 41.45MJ/kg$$

Where, Q_{LHV} blend fuel is lower heating value of blend fuel

Q_{LHV} alcohol is lower heating value of alcohol

Q_{LHV} gasoline is lower heating value of gasoline

η_c is combustion efficiency

$$Q_{in} = m_f\times Q_{LHV}\times\eta_c$$

$= (0.000033kg) (41.45MJ/kg) (0.9)$

$= 0.0015MJ = 1.5KJ$

Heat release rate

The heat addition for spark ignition engines may be a prescribed function of crank angle and related by the following formula Smith (2002).

$$Q_1 = Q_{in} \times xb \text{ .... 3-12}$$

Where,

$$xb = 0.5\left[1 - \left(cos\frac{\theta - \theta scbTDC}{\theta dc} \times \pi\right)\right] \text{..........................3-13}$$

xb = is the fraction of the heat release, theta refers to crank angle

$\theta scbTDC$ = is theta (degree) start of combustion before top dead center

θdc = is theta duration of combustion

The above equation is applied only during the time of heat release;

$$Q_1 = Q_{in} \times xb$$

$$= m_f \times Q_{LHV} \times \eta_c \times 0.5\left[1 - \left(\cos\frac{\theta - \theta scbTDC}{\theta dc} \times \pi\right)\right]$$

$$= (1.5KJ)*(0.99)$$

$$= 1.504KJ$$

$$W_1 = PV \text{..3-14}$$

Where, $P= P_2$ that means pressure at the end of compression

$V= V_D$ is the swept volume

W_1 = work done

$$W_1 = PV$$

$$= 1797.69kpa \times 0.000444m^3$$

$$= 0.798KJ$$

From:

$$Q - W = \Delta U \text{... 3-15}$$

Where, Q is heat added

W is work done

ΔU is change of internal energy

$$Q_1 - W_1 = M_{m1} \times cv \times (T_3 - T_2) \ldots\ldots\ldots\ldots\ldots \text{3-16}$$

Where, cv = 0.821KJ/Kg-k

$$Q_1 - 0.798\text{KJ} = 0.00076\text{kg} \times 0.821 \frac{\text{KJ}}{\text{Kg\_k}} \times (T_3 - T_2)$$

$$T_3 = \left[\frac{(1.504\text{KJ} - 0.798\text{KJ})}{0.00051\text{Kg} \times \left(\frac{0.821\text{KJ}}{\text{KG\_K}}\right)}\right] + T_2$$

= 1474.5+668.78k

= 2369K (this is maximum temperature)

$$P_3 = P_2 \times \left(\frac{T_3}{T_2}\right) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{3-17}$$

$$= 1797.69\text{Kpa} \times \frac{2369}{683.12}$$

= 6234.89kpa = P_{max} (maximum pressure)

State 4

Power stroke.

The following ideal gas relationships can be used to find pressure and temperature.

$$P_4 = P_3 \times \left(\frac{1}{R_C}\right)^K \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{3-18}$$

$$= 6234.89\text{kpa} \times \left(\frac{1}{8.5}\right)^{1.35}$$

= 346.82Kpa

$$T_4 = T_3 \times \left(\frac{1}{R_C}\right)^{K-1} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{3-19}$$

$$= 2369 \times \left(\frac{1}{8.5}\right)^{1.35-1}$$

= 1120.13k

Work produced in isentropic power stroke for one cylinder during one cycle:

$$W_{power} = \frac{M_{m1} \times R \times (T_4 - T_3)}{1 - K} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{3-20}$$

$$= \frac{0.00051\text{kg} \times 0.287\text{KJ/kg\_k} \times (1120\text{k} - 2369\text{k})}{-0.35}$$

= 0.52KJ

Where, W_{power} is work of power stroke

Work absorbed during isentropic compression stroke for one cylinder during one cycle.

$$W_{compression} = \frac{M_{m1} \times R \times (T_2 - T_1)}{1-K} \text{.......................} 3\text{-}21$$

$$= \frac{0.00051\text{kg} \times 0.287\text{KJ/kg\_k} \times (683.12\text{k} - 323)}{-0.35}$$

= - 0.15KJ

Where, $W_{Compression}$ is work of compression stroke

Work of the intake stroke is canceled by work of the exhaust stroke.

Net indicated work for one cylinder during one cycle is:

$W_{net} = W_{power} + W_{compression}$

= 0.52KJ – 0.15KJ

= 0.37KJ

Where, Wnet work net

From this indicated thermal efficiency can be calculated as follow:

$$\eta_t = \frac{W_{net}}{Q_{in}} \text{...} 3\text{-}22$$

$$= \frac{0.37\text{KJ}}{1.504\text{KJ}}$$

= 0.246

= 24.6%

Where, η_t is thermal efficiency

Indicated mean effective pressure (Imep) can be calculated from the following:

$$\text{Imep} = \frac{W_{net}}{V_1 - V_2} \text{...} 3\text{-}23$$

$$= \frac{0.37\text{KJ}}{0.000392\text{m}^3}$$

= 943.87kpa

From this:

$$P_i = \frac{W_{net} \times N}{n} \text{...} 3\text{-}24$$

= [0.37KJ/cylinder-cycle (3000/60 rev/sec)/ (2rev/cycle)] 1cylinder

= 9.25kw

Where, P_i is indicate power

N is engine speed (rpm)

n is number of stroke

$$P_b = \eta_m \times P_i \text{...3-25}$$

= 0.85 *9.25kw

= 7.86kw

Where, mechanical efficiency (η_m) is 85%

$$T_b = \frac{60 \times P_b}{2 \times \pi \times N} \text{...3-26}$$

= 60 x 7.86/2*3.14*3000

= 25N-m

$$b_{sfc} = \frac{m^\circ{}_f}{P_b} \text{...3-27}$$

= [0.000033kg/cylin-cycle) *(50rev/sec)*(0.5 cycle /rev)*1cyli]/ 7.86kw

= 0.0001kg/sec/kw

= 0.38kg/kWh

Where, P_b is brake power

T_b is brake torque

b_{sfc} is brake specific fuel consumption

$m^\circ{}_f$ is mass of fuel flow rate

3.4. Mathematical Model for Engine Performance

The following 'Empirical Relations' are obtained from experimental data of 'engine performance test' using a 'curve fitting method'.

- for SI (Carburetion) Engines:

$$P_b = P_{b\_max}\left(-x^3 + x^2 + x\right) \quad \text{where, } x = \frac{N}{N_{@P_b\_max}} \quad \text{..........} \quad \text{'N' in 'rpm'; 'P}_b\text{' in 'kW'}$$

$$N_{min} = 500 - 1000\ \text{rpm} \quad \text{and} \quad N_{max} = (1.05 \ \text{to}\ 1.15) \times N_{@Pb\text{-}max} \ ;$$

$$\Delta N = \frac{N_{max} - N_{min}}{I_N} \quad \rightarrow \quad I_N \approx 10$$

$$T_b = \frac{P_b}{\omega} \quad \text{..........} \quad \text{in 'Nm'} \quad \text{where, } \omega = \frac{2\pi \times N}{60}$$

$$bscf = bscf_{@Pb\_max}\left(x^2 - 1.2x + 1.2\right) \quad \text{..........} \quad \text{in 'kg/kWh'}$$

$$bmep = \frac{P_b}{E_c \times n_{cps}} \quad \rightarrow \quad \text{in 'bar'} \quad \text{where, } n_{cps} = \frac{N}{Z_{rpc} \times 60}$$

Note: Using a computer Programming Language (C++,) in appendix 1 and Application Software's (MS-Excel,), the expected 'Engine Performance Curves' can be plotted.

3.5. Mathematical Model for Incomplete Combustion Emission of Blend Fuel.

The chemical reaction of blend fuel for not stoichiometric reaction is as follows.in this reaction there is additional oxygen and carbon monoxide. But since it is rich fuel type mixture there is no oxygen in product part. If there is dissociation reaction in the combustion chamber the product part element is not limited to five rather ten (10) products.

To calculate the molar fraction of all combustion product the very complex numerical mathematical model is needed.

For this reason this work is limited to only five combustion products.

If all carbon in hydrocarbon and ethanol is combusted and there is no carbon in combustion product as in equation 3.29 the reaction is known to be stoichiometric reaction.

$$mfa(C_nH_mOH_y) + mfg(C_bH_b) + atha(O_2 + 3.76N_2) = n_1CO_2 + n_2H_2O + n_3N_2 \quad \text{......3-28}$$

$$mfa(C_nH_mOH_y) + mfg(C_aH_b) + ath(O_2 + 3.76N_2) = n_1CO_2 + n_2H_2O + n_3N_2 + n_4O_2 + n_5CO \quad \text{......3-29}$$

Where, mfa is molar of alcohol

mfg is molar fraction of gasoline

atha is molar of air for stoichiometric air fuel ratio.

ath = λ* atha

Where, ath is molar of air for not stoichiometric reaction.

λ is equivalence ratio of air fuel

The equilibrium of reactants and product parts are done as follow.

$$\text{C: } (\text{n} \times \text{mfa}) - (\text{a} \times \text{mfg}) = \text{n}_1 + \text{n}_5$$

$$\text{H: } ((\text{m} + \text{y}) \times \text{mfa}) + (\text{b} \times \text{mfg}) = 2 \times \text{n}_2$$

$$\text{O: } \text{mfa} + 2 \times \text{ath} = 2 \times \text{n}_1 + \text{n}_2 + \text{n}_5$$

$$\text{N: } 2 \times \text{ath} \times 3.76 = 2 \times \text{n}_3$$

$$\text{n}_1 = (\text{mfa} + 2 \times \text{ath}) - \left(\frac{((\text{m} + \text{y}) \times \text{mfa}) - (\text{b} \times \text{mfg})}{2}\right) - (\text{n} \times \text{mfa}) - (\text{a} \times \text{mfg})$$

$$\text{n}_2 = \frac{(\text{m} \times \text{mfa}) + (\text{b} \times \text{mfg})}{2}$$

$$\text{n}_3 = \frac{(2 \times \text{ath} \times 3.76)}{2}$$

$n_4 \approx 0$, Because the mixture is rich type and O_2 not expected.

$$\text{n}_5 = ((\text{n} \times \text{mfa}) + (\text{a} \times \text{mfg})) - \text{n}_1$$

$$\text{n}_t = \text{n}_1 + \text{n}_2 + \text{n}_3 + \text{n}_4 + \text{n}_5$$

$$\text{y}_1 = \frac{\text{n}_1}{\text{n}_t}, \text{y}_2 = \frac{\text{n}_2}{\text{n}_t}, \text{y}_3 = \frac{\text{n}_3}{\text{n}_t}, \text{y}_4 = \frac{\text{n}_4}{\text{n}_t}, \text{y}_5 = \frac{\text{n}_5}{\text{n}_t}$$

Where, n_1, n_2, n_3, n_4, n_5 are molar of carbon dioxide, water vapor, nitrogen, carbon monoxide respectively and y_1, y_2, y_3, y_4, y_5 are molar fraction of carbon dioxide, water vapor, nitrogen, carbon monoxide respectively.

3.6. Experimental Analysis for E0 and E15

To Plot a power curve and the torque curve at full load the following procedures are under taken.

1. Setting up

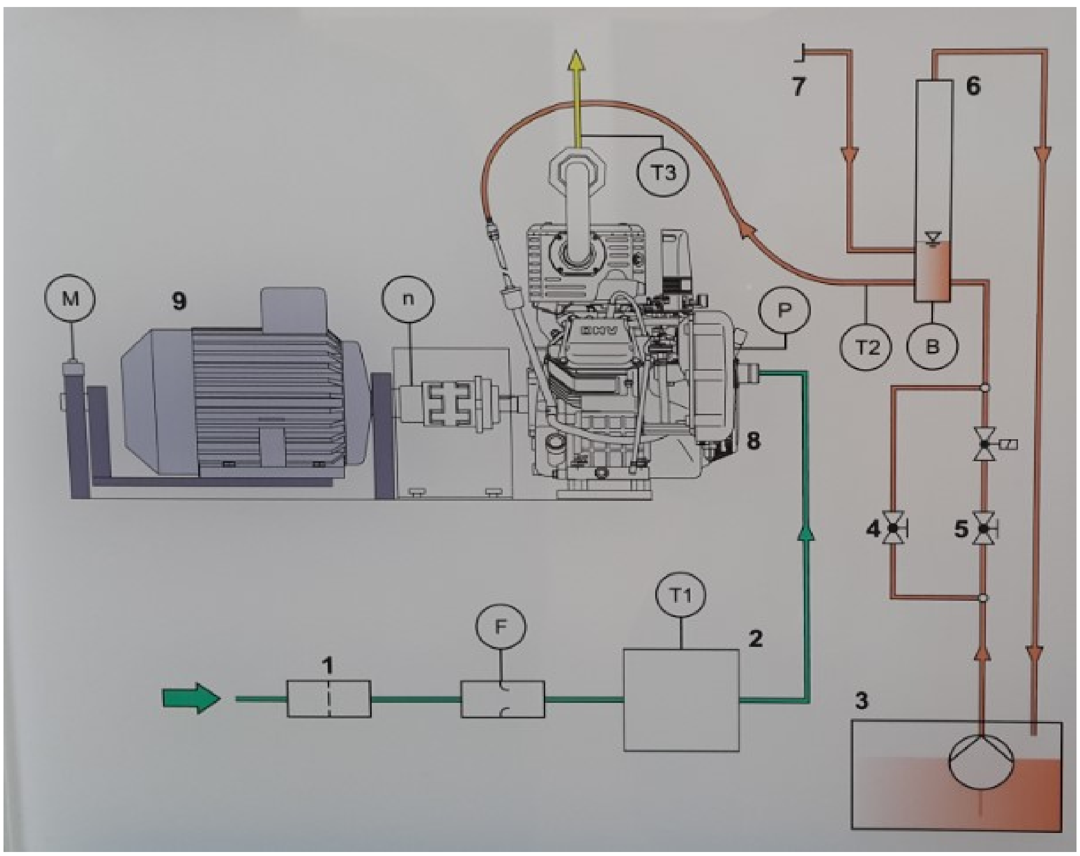


Figure 3.3. Test stand for single cylinder air cooled engine 7.5 Kw

Main component parts of test setup **Instrumentation and control**

1. Air filter
2. Settling tank
3. Fuel tank with pump

B. Fuel consumption

F. Air flow rate

M. Torque

4. Drain valve — n. Speed
5. Filling valve — P. Intake pressure
6. Fuel measurement tube — T1. Intake temperature
7. Diesel return line — T2. Fuel temperature
8. Combustion engine — T3. Exhaust temperature
9. Asynchronous motor for starting and loading

1. Installing and connecting the engine

- The engine and base plate are installed in the CT 110 Test Stand for Small Combustion Engines up to 7.5kW.
- The engine is aligned with the braking device such that after inserting the center piece of the coupling, the inserts for the two halves of the coupling are aligned (see also the instructions for the CT 110)
- ATTENTION! Imprecise alignment of the halves of the coupling can result in irreparable damage to the plastic inserts, particularly at high speeds!
- After alignment the fixing bolts for the engine are tightened.
- The safety cover for the coupling is fitted.
- The fuel line to the engine is connected to the supply connection on the CT 110 test stand.
- The other lines on the engine (ignition, vacuum hose and temperature sensor) should be connected to the connections on the test stand as shown below.
- The air hose and the exhaust gas hose from the test stand are connected to the engine.

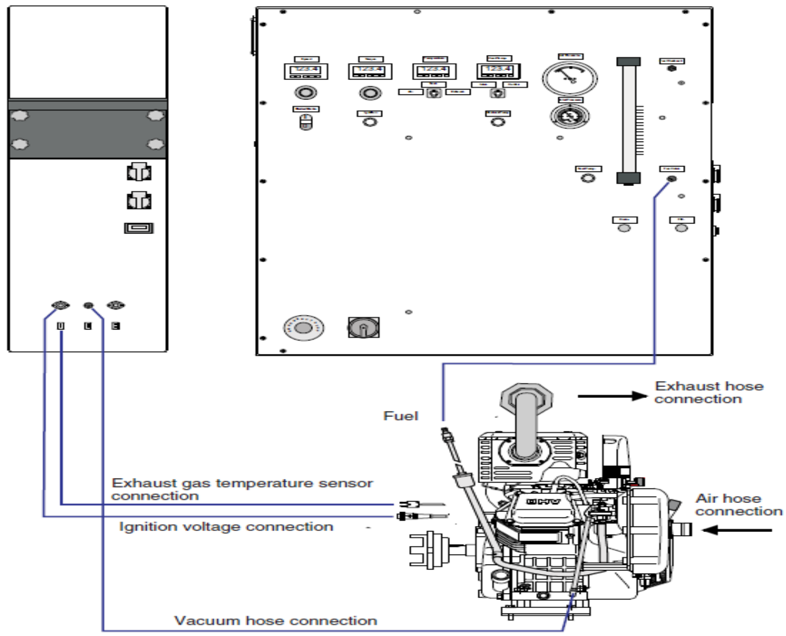


Figure 3.4. Connection between CT110 test stand and CT 100.20 engine.

3.7. Performing the experiment

- Once the engine has warmed up, operate it at full throttle.
- Apply a load to the engine:
- To do this, set the torque adjusting knob on the CT 110 to a maximum value of “10".
- Set the speed adjusting knob so that the maximum speed in the engine’s speed range is reached.
- Operate the “Starter/Brake” button.
- Record the measured values (speed, torque and fuel consumption), i.e. read off the speed and torque on the displays. Calculate the fuel consumption using the scale on the measuring tube and a stopwatch.
- To move to the next measuring point, reduce the speed with the adjusting know on the CT 110.
- Record the measured values.
- Repeat the procedure until a speed of around 2200 rpm is reached or the engine comes to a stop.

CHAPTER FOUR

4. RESULT AND DISCUSSION

4.1. Output of computer c++ program for E0

Table 4.1. "'V', 'P' and 'T' as a function of 'theta' "for E0

theta (deg.)	V (cc)	p (bar)	T (K)	theta (deg.)	V (cc)	p (bar)	T (K)
20	68.43	0.9	313	360	52.27	31.67	1.22E+03
30	87.88	0.9	313	370	56.36	49.18	2.03E+03
40	113.77	0.9	313	380	68.43	55.67	2.80E+03
50	144.87	0.9	313	390	87.88	49.26	3.18E+03
60	179.78	0.9	313	400	113.77	37.48	3.13E+03
70	217	0.9	313	410	144.87	27.04	2.88E+03
80	255.05	0.9	313	420	179.78	20.21	2.67E+03
90	292.53	0.9	313	430	217	15.67	2.50E+03
100	328.24	0.9	313	440	255.05	12.6	2.36E+03
110	361.17	0.9	313	450	292.54	10.47	2.25E+03
120	390.54	0.9	313	460	328.24	8.96	2.16E+03
130	415.82	0.9	313	470	361.17	7.88	2.09E+03
140	436.67	0.9	313	480	390.54	7.09	2.03E+03
150	452.93	0.9	313	490	415.82	6.51	1.99E+03
160	464.53	0.9	313	500	436.67	6.1	1.95E+03
170	471.47	0.9	313	510	452.93	5.8	1.93E+03
180	473.79	0.9	313	520	464.53	5.61	1.91E+03
190	471.47	0.91	313.54	530	471.47	5.5	1.90E+03
200	464.53	0.92	315.17	540	473.79	5.46	1.90E+03
210	452.92	0.96	317.97	550	471.47	1.07	327.63
220	436.67	1	322.07	560	464.53	1.07	327.63
230	415.81	1.07	327.63	570	452.92	1.07	327.63
240	390.53	1.17	334.9	580	436.67	1.07	327.63
250	361.16	1.3	344.19	590	415.81	1.07	327.63
260	328.24	1.48	355.9	600	390.53	1.07	327.63
270	292.53	1.73	370.54	610	361.16	1.07	327.63
280	255.04	2.08	388.76	620	328.24	1.07	327.63
290	216.99	2.58	411.38	630	292.53	1.07	327.63
300	179.77	3.33	439.38	640	255.04	1.07	327.63
310	144.87	4.46	473.87	650	216.99	1.07	327.63
320	113.76	6.18	515.7	660	179.77	1.07	327.63
330	87.88	8.75	564.47	670	144.86	1.07	327.63
340	68.43	12.27	616.12	680	113.76	1.07	327.63
350	56.36	16.4	678.53	690	87.88	1.07	327.63
				700	68.43	1.07	327.63

The table 4.1 is the result of computer c++ program developed which taken from appendix 1. This program can easily calculate volume, pressure, temperature, the input amount of mass charged to cylinder, the mole and molar fraction of emissions, the brake power, brake torque, brake specific fuel consumption, thermal efficiency, and brake mean effective pressure. Most of the result was summarized and analyzed as follow:

- **Engine Size**

$E_C = 392$ cc, $V_S = 392$ cc, $V_{cl} = 52.27$ cc, $V_T = 444.27$cc

Where, E_C is engine capacity V_{cl} is clearance volume

V_S is swept volume V_T is total volume

- **The input amount in the combustion reaction**

atha $= 11.75$, LAMBDA (λ) =0.8, ath $= 9.4$, $X_{AL} = 0$, $X_G = 1$,$Y_{AL} = 0$,$Y_G = 1$

AFR $= 11.63$; $ETA_{-C} = 75.54\%$, $Y_{AL-Per} = 0\%$.

m_m = 0.00045 kg, m_a = 0.000393 kg,

m_f = 3.383015e-05 kg, M_{res} = 2.247992e-05 kg

- **The output amount in the combustion reaction**

W_{net} = 0.34 kJ; Qin = 1.21 kJ; ETA_ith = 27.65 %; Imep = 8.55 bar

Pb = 7.13 kW, Tb = 22.69 NM; bsfc = 0.46 kg/kWh

Where,

atha is mole of air for stoichiometric, reaction, m_m is total mass charged to engine ,m_a is mass of air, m_f is mass of fuel, M_{res} is mass residual, AFR is air fuel ratio, ETA_c is combustion efficiency, λ is equivalence air fuel ratio, Imep indicated mean effective pressure, Pb is brake power, Tb is brake torque, ath is mole of air for not stoichiometric reaction, x_{Al} is mole of alcohol, x_G is mole of gasoline, y_{Al} is molar fraction of alcohol, y_G is molar fraction of gasoline, y_{Al_per} is percentage of alcohol, W_{net} is work net, Qin is heat added to combustion, ETA_ith is thermal efficiency bsfc is brake specific fuel consumption

4.1.1. Temperature

The maximum temperature is created at 390° of crank angle which can be adjusted initially on the engine. This temperature can be affected by ambient temperature of environment. So it depends initially on the environmental temperature. The angle at which combustion started before TDC and the duration of combustion determine the crank angle at which maximum temperature occurred. This temperature (Tmax) can affect the work net from the engine which has direct relation with brake power and other engine performance parameter. The computer c++ program written for actual engine cycle virtually consider all factors like in real engine.

For this specific engine and environment the maximum temperature created in engine was 3180k.

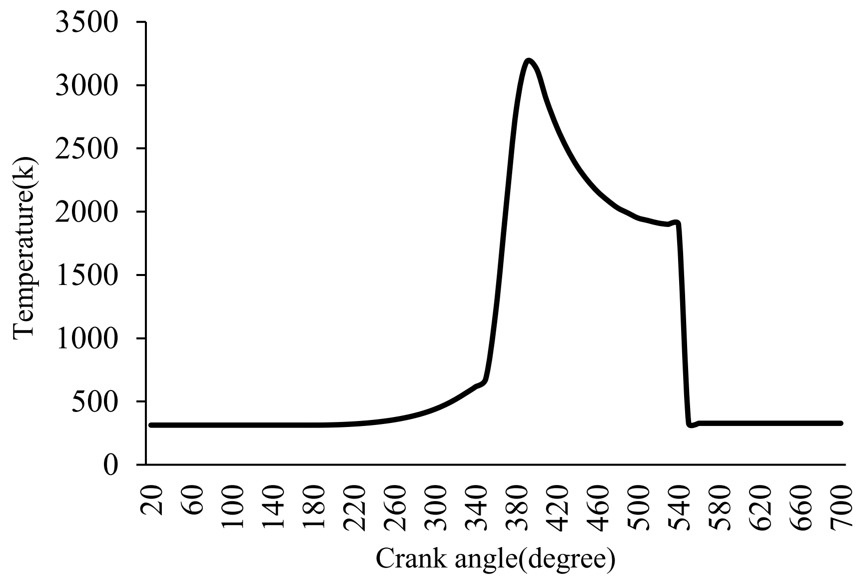


Figure 4.1. Temperature versus crank angle for E0 (pure gasoline)

4.1.2. Pressure

Temperature developed in engine and pressure created in engine are interdependent. The maximum pressure is created at 380° of crank angle which can be adjusted on the engine and not constant for all engine. The angle at which combustion started before TDC and the duration of combustion determine the crank angle at which maximum pressure occurred. Like that of temperature the maximum pressure also depends on the environmental pressure. For this specific engine and environment the maximum pressure created in engine was 55.67 bar.

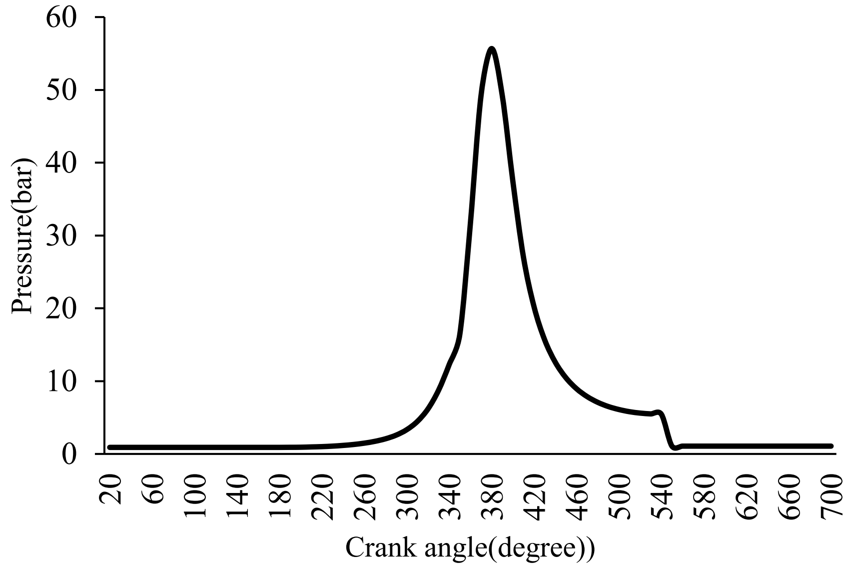


Figure 4.2. Pressure versus crank angle for E0 (pure gasoline)

4.1.3. Volume

The volume created in the engine depends on the compression ratio, bore, stroke and the length of connecting rod. Due to positive and negative displacement in the cylinder, the volume created changes between total volume of cylinder and clearance volume of cylinder. This computer c++ program can calculate the volume at every 720^{o} of crank angle. If compression is the same for pure gasoline and blend fuel the same volume can be created in the engine cylinder.

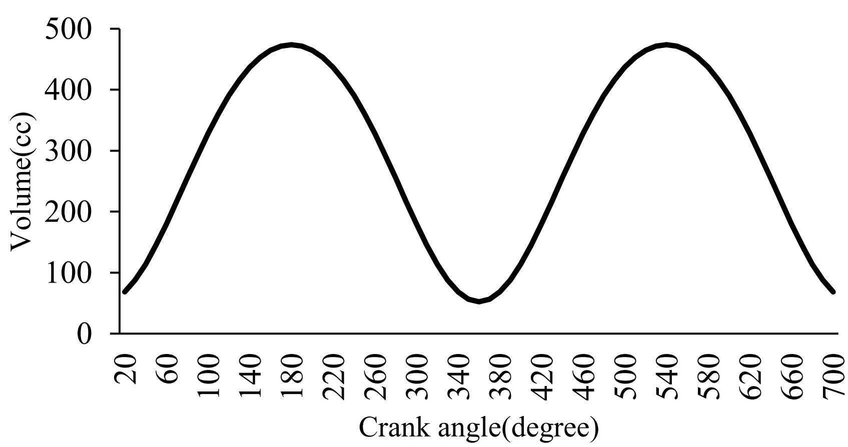


Figure 4.3. Volume versus crank angle for E0 (pure gasoline)

4.1.4. Pressure vs volume diagram

The pressure vs volume diagram is very important for the design of every engine. The peak pressure is one of the determinant factor for brake power and other engine performance parameter. This peak pressure depends on compression ratio, environmental pressure and the nature of fuel used. The value of this peak pressure is the same with maximum pressure (55.67 bar) developed in engine. This peak pressure mostly depends on the type of fuel used and ignition timing.

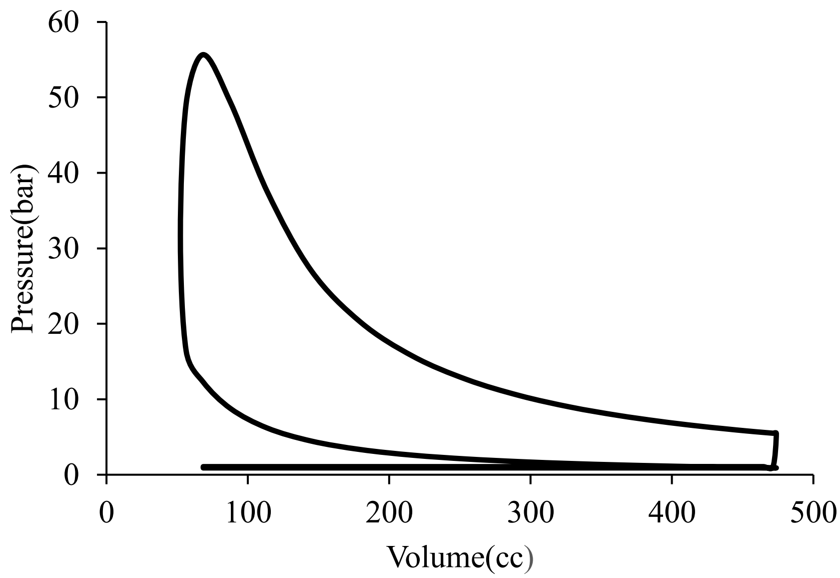


Figure 4.4. Volume versus crank angle for E0 (pure gasoline)

4.1.5. Brake power

Brake power is the function of brake mean effective pressure, engine capacity, and engine revolution per minute and mechanical efficiency. There is two possible ways of calculating brake power. The first one is analytical method which can be used to calculate the maximum power for thermodynamic analysis of actual engine cycle and by using only this method one cannot draw the following diagram. The second one is empirical method which can be used to calculate the maximum brake power by observing engine specification. This method is used for drawing the following diagram which is almost the same with the output of computer simulator of dynamometer.

As a result the maximum brake power for selected engine by using pure gasoline was 7.13kW

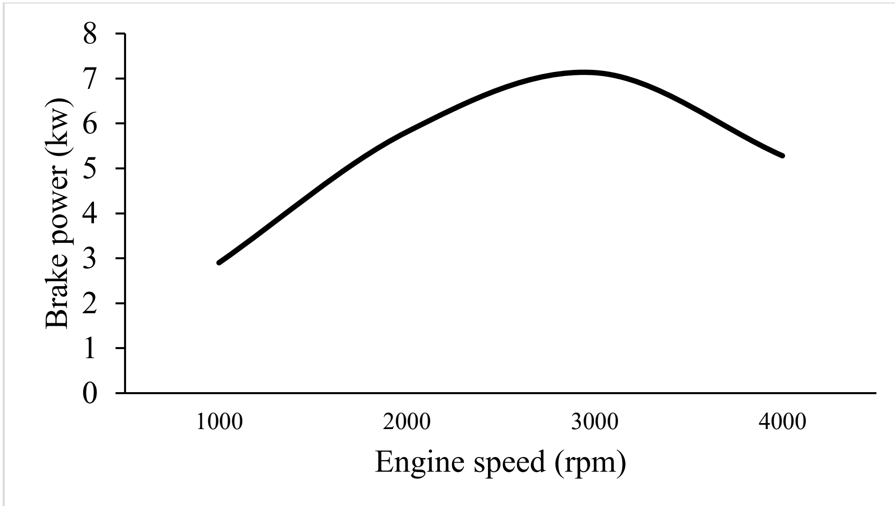


Figure 4.5. Brake power versus engine revolution per minute for E0 at 3000rpm.

4.1.6. Brake torque

Brake torque is the function of brake power, engine capacity, and engine rpm. The factor which affect the brake power has significant role on brake torque. Both of them has directly proportional to each other. The two possible way used to calculate brake power is also functional for brake torque. The maximum brake torque created was 27.75Nm at 2000rpm. But this maximum brake torque is greater than the engine maximum brake torque. For this reason running the engine at lower speed is impossible and can damage engine. So computer c++ program can indicate at which speed engine can run safely. As shown on the following diagram running the engine above broken line area can possibly damage it. Like that of the graph of brake power the following diagram is drawn by using empirical relation of maximum power, speed at maximum power and speed which is the result of experimental analysis.

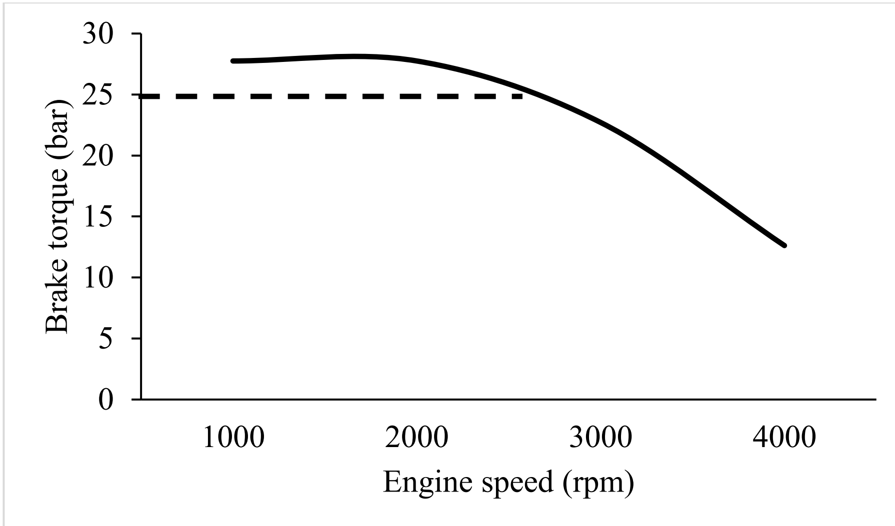


Figure 4.6. Brake torque versus engine revolution per minute for E0

4.1.7. Brake specific fuel consumption

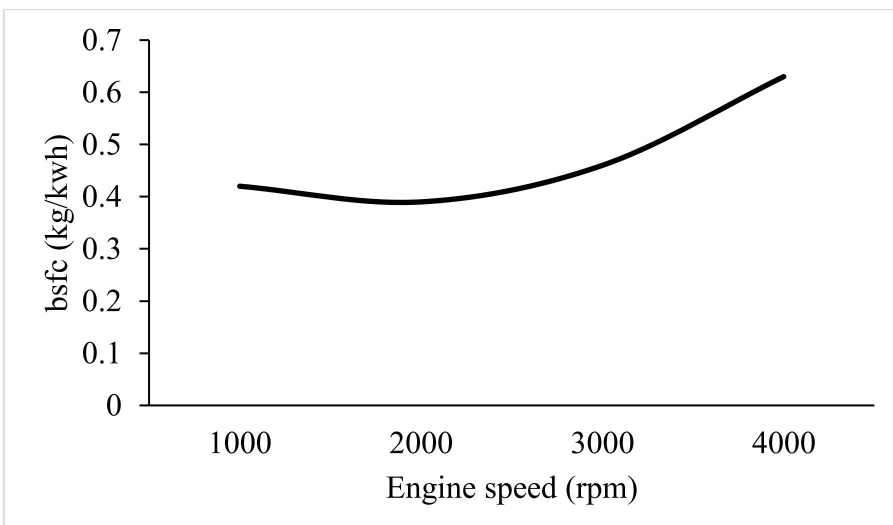


Figure 4.7. Brake specific fuel consumption versus engine revolution per minute for E0

Brake specific fuel consumption is the function of mass of fuel flow rate charged to combustion chamber, speed of engine, number of cylinder, number of stroke and brake power of engine. This parameter has also related to the pressure and temperature of the environment. The type of engine and fuel can affect the amount of brake specific fuel consumption. Brake specific fuel consumption is high at lower speed and decreasing as speed increase and become optimum at the medium and increase as speed increase. As that of brake power and torque brake specific fuel consumption diagram is drawn by using empirical method.as result the minimum fuel consumption is 0.39kg/kWh at speed between 2000rpm.

4.1.8. Brake mean effective pressure

Brake mean effective pressure is the function of brake power, engine capacity and engine revelation per minute. Brake mean effective pressure is directly proportional to brake power and brake torque. So all factors which affect brake power can directly affect brake mean effective pressure. This pressure is crucial point for designer of engine and as other performance engine parameter graph brake mean effective pressure is also drawn by using empirical method. So as shown on the graph the maximum brake mean effective pressure is 8.89bar at 1000rpm.

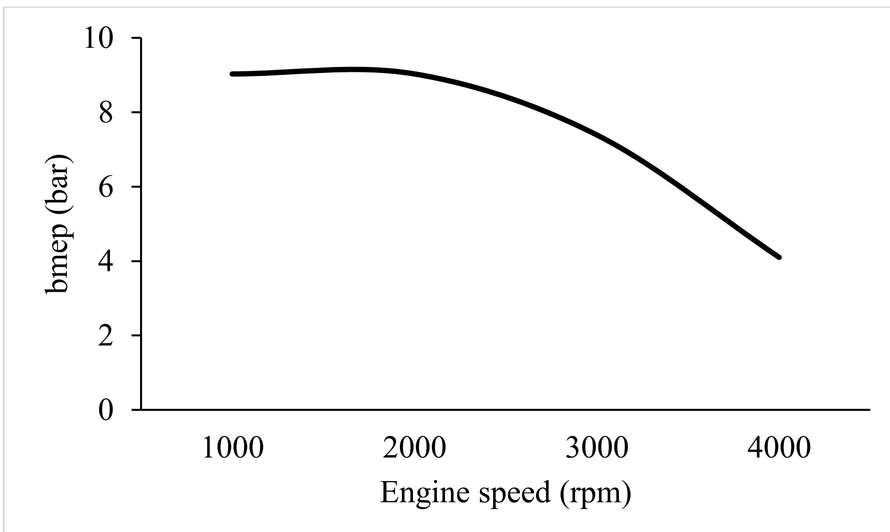


Figure 4.8. Brake mean effective pressure versus engine revolution per minute for E0

4.2. Output of computer c++ program for E15

Table 4.2 'V', 'p' and 'T' as a function of 'theta' "for E15

theta (deg.)	V (cc)	p (bar)	T (K)	theta (deg.)	V (cc)	p (bar)	T (K)
20	68.43	0.9	313	360	52.27	31.88	1.22E+03
30	87.88	0.9	313	370	56.36	49.67	2.05E+03
40	113.77	0.9	313	380	68.43	56.32	2.83E+03
50	144.87	0.9	313	390	87.88	49.86	3.22E+03
60	179.78	0.9	313	400	113.77	37.95	3.17E+03
70	217	0.9	313	410	144.87	27.38	2.91E+03
80	255.05	0.9	313	420	179.78	20.46	2.70E+03
90	292.53	0.9	313	430	217	15.87	2.53E+03
100	328.24	0.9	313	440	255.05	12.76	2.39E+03
110	361.17	0.9	313	450	292.54	10.6	2.28E+03
120	390.54	0.9	313	460	328.24	9.08	2.19E+03
130	415.82	0.9	313	470	361.17	7.98	2.12E+03
140	436.67	0.9	313	480	390.54	7.18	2.06E+03
150	452.93	0.9	313	490	415.82	6.6	2.01E+03
160	464.53	0.9	313	500	436.67	6.17	1.98E+03
170	471.47	0.9	313	510	452.93	5.88	1.95E+03
180	473.79	0.9	313	520	464.53	5.68	1.94E+03
190	471.47	0.91	313.54	530	471.47	5.57	1.93E+03
200	464.53	0.92	315.17	540	473.79	5.53	1.92E+03
210	452.92	0.96	317.97	550	471.47	1.07	327.63
220	436.67	1	322.07	560	464.53	1.07	327.63
230	415.81	1.07	327.63	570	452.92	1.07	327.63
240	390.53	1.17	334.9	580	436.67	1.07	327.63
250	361.16	1.3	344.19	590	415.81	1.07	327.63
260	328.24	1.48	355.9	600	390.53	1.07	327.63
270	292.53	1.73	370.54	610	361.16	1.07	327.63
280	255.04	2.08	388.76	620	328.24	1.07	327.63
290	216.99	2.58	411.38	630	292.53	1.07	327.63
300	179.77	3.33	439.38	640	255.04	1.07	327.63
310	144.87	4.46	473.87	650	216.99	1.07	327.63
320	113.76	6.18	515.7	660	179.77	1.07	327.63
330	87.88	8.75	564.47	670	144.86	1.07	327.63
340	68.43	12.27	616.12	680	113.76	1.07	327.63
350	56.36	16.42	679.46	690	87.88	1.07	327.63
				700	68.43	1.07	327.63

- **Engine Size**

E_c = 392 cc, V_s = 392 cc , V_{cl} = 52.27 cc, V_T = 444.27 cc

- **The input amount in the combustion reaction**

atha =9.14, LAMBDA =0.86, ath =7.86, x_{Al}=0.15, xG =0.85, y_{Al} =0.3

yG =0.7; AF = 11.78; ETA_c = 82.46 %; y_{Al_per} = 29.87 %

mm = 0.00045 kg, ma = 0.000394 kg, mf = 3.342708e-05 kg

mres = 2.247992e-05 kg, mm = 0.00045 kg

- **The output amount in the combustion reaction**

W_{net} = 0.34 kJ; Qin = 1.23 kJ; ETA_ith = 27.7 %; Imep = 8.7 bar

Pb = 7.24 kW

Tb = 23.07 NM; bsfc = 0.44 kg/kwh

4.2.1. Temperature

The maximum temperature created by E15 (15% ethanol and 85% of gasoline) was 3220k at 390° crank angle. This show that the angle at which maximum temperature created is not depends on the type fuel rather ignition timing.

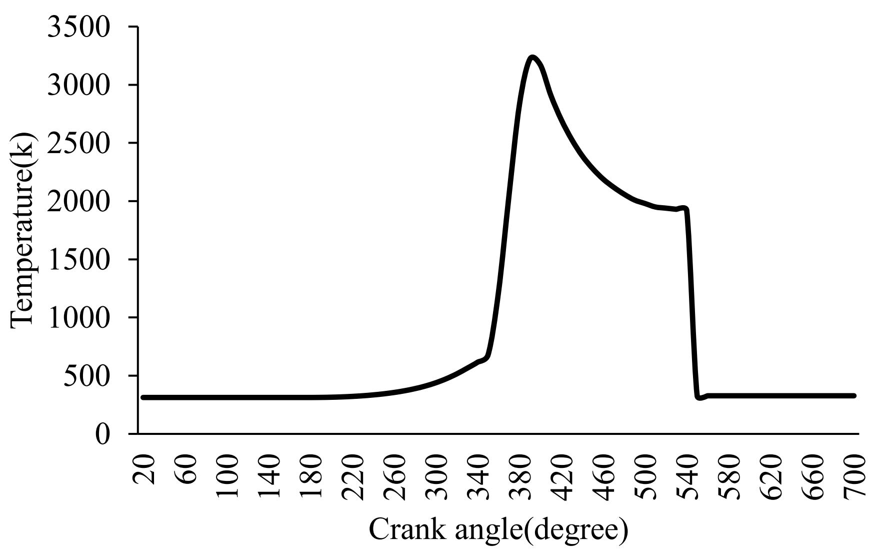


Figure 4.9. Temperature versus crank angle for E15

All factors which listed in pure gasoline can affect the value of maximum temperature in E15.Due to higher air-fuel equivalence ration or λ (LAMBDA) value in ethanol, the thermal efficiency of blend fuel is higher than pure gasoline, as result the combustion efficiency of blend fuel is higher than pure gasoline. Due to this the maximum temperature created in engine by blend fuel is higher than pure gasoline.

4.2.2. Pressure

The maximum pressure created by this blend fuel means E15 is 56.32 bar. This pressure is created at the same crank angle 380º with E0 (pure gasoline). Due to high thermal efficiency of oxygenated fuel the blend fuel has higher maximum pressure than pure gasoline. The maximum temperature created in the engine is directly proportional to maximum pressure created in engine.so the all factors which can affect the amount of temperature while combustion can directly affect the pressure created in engine.

This maximum pressure is very important design point of new engine and it is crucial for the operation of engine.

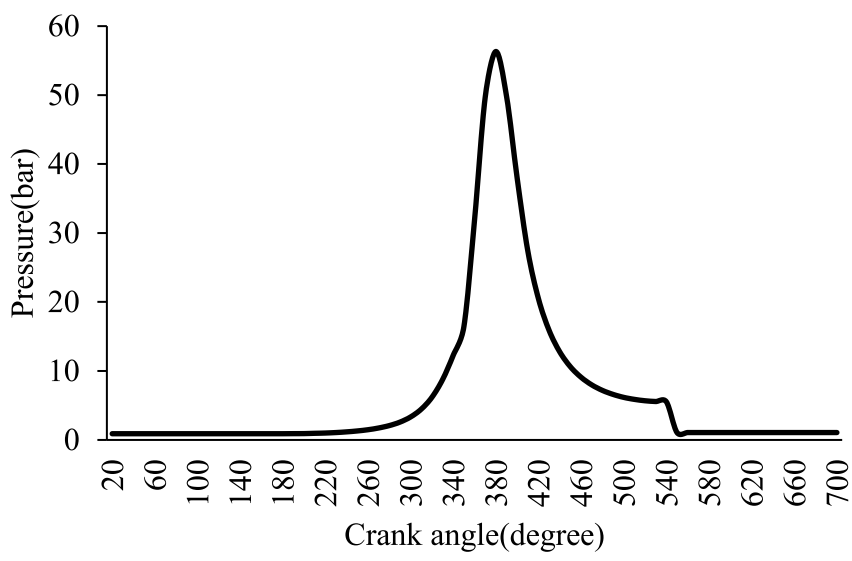


Figure 4.10. Pressure versus crank angle for E15

4.2.3. Volume

In this specific study only one compression ratio is taken as input from the engine specification.one of the determinant factor for the value of swept and clearance volume is compression ratio. Theoretically the compression of oxygenated fuel is higher than hydrocarbon fuel but for this work the compression ratio for both blend and pure gasoline is the same. Due to this there is no the variation of swept volume and clearance volume created in both blend and pure gasoline. Despite this variation can highly affect engine performance practically, the compression ratio for both blend and pure gasoline is not the same.

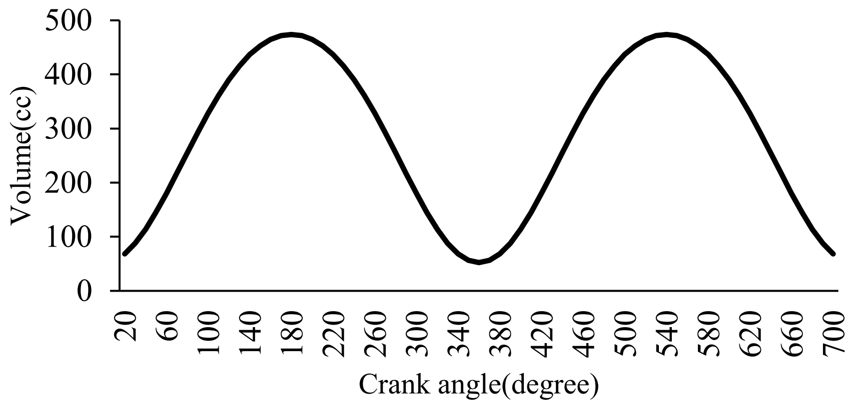


Figure 4.11. Volume versus crank angle for E15

4.2.4. Pressure vs volume diagram

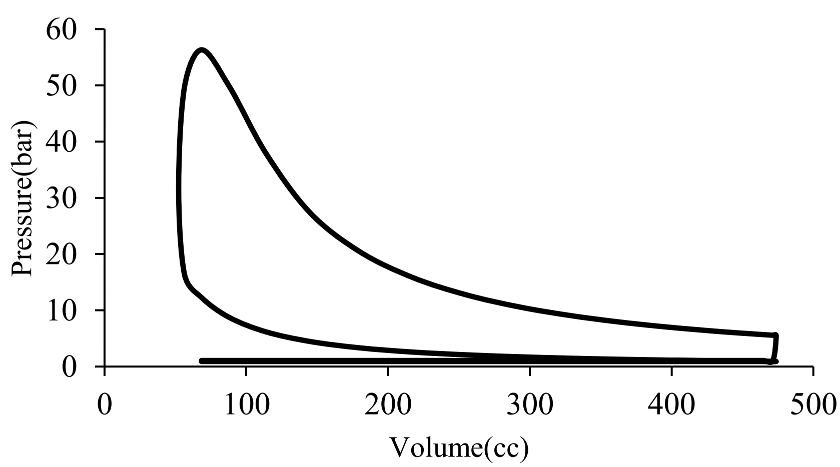


Figure 4.12. Pressure versus volume diagram for E15

The pressure vs volume diagram is very important for the design of every engine. The peak pressure is one of the determinant factor for brake power and other engine performance parameter. This peak pressure depends on compression ratio, environmental pressure and the nature of fuel used. Due to this the maximum pressure created by blend fuel is higher than pure gasoline. The value of this peak pressure is the same with maximum pressure in E15 (56.32 bar). For this reason the type of fuel used in producing power has significant role in affecting the brake power of engine.

4.2.5. Brake power

The brake power produced by E15 is higher than E0. This increment is also for E5 and E10 and as a result the power produced by E15>E10>E5>E0. This is due to high thermal efficiency of oxygenated fuel. This thermal efficiency is dependent on equivalence ratio of air-fuel (λ value).

There is no oxygen in hydrocarbon fuel but in alcohol. Due to this ethanol has higher equivalence ratio of air fuel than pure gasoline. For this reason the brake power produced by E15 is greater than E0.

So the maximum brake power produced by E15 in this specific engine is 7.24kw at 3000rpm.This graph is drawn by using empirical method which is the result of experimental result.

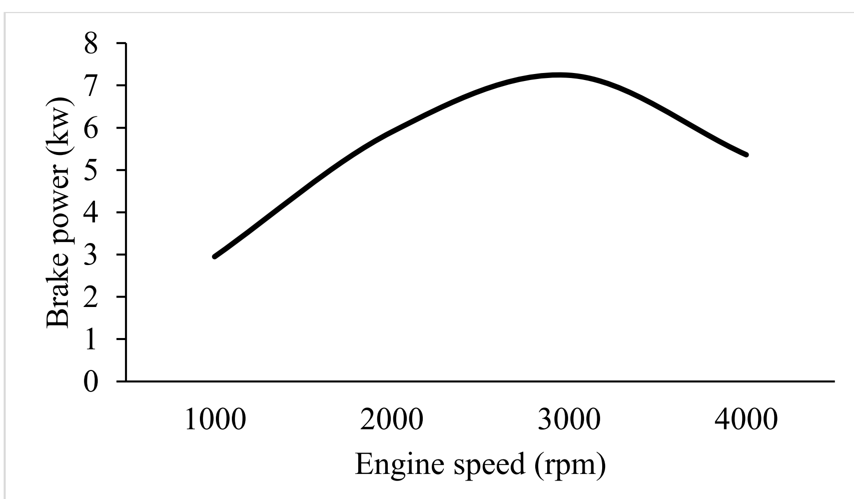


Figure 4.13. Brake power versus engine speed for E15

4.2.6. Brake torque

The brake torque is directly proportional to brake power.as a result the increment of brake power increase the brake torque. This increment is due to the increase of thermal efficiency and break mean effective pressure.

The basic cause for this change is equivalence ratio of air- fuel (λ value) and this means the λ value of blend fuel is greater than pure gasoline because of oxygen content of alcohol.

On this specific study the maximum brake torque as the result of computer c++ program is 28.18 Nm at 2000rpm. But this maximum brake torque is greater than the engine maximum brake torque. For this reason running the engine at lower speed is impossible that can damage engine. So computer c++ program can indicate at which speed engine can run safely. As shown on the following diagram running the engine above dash line area can possibly damage it.

The following diagram computer gram is drawn by using empirical method which is the result of experimental analysis.

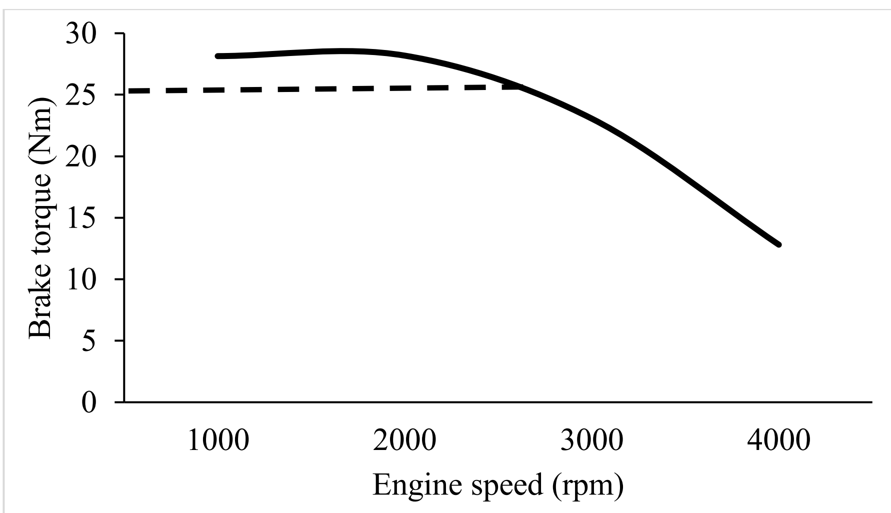


Figure 4.14. Brake torque versus engine revolution per minute for E15.

4.2.7. Brake specific fuel consumption

Brake specific fuel consumption is the function of mass of fuel delivered to engine and brake power produced by the engine.

The brake specific fuel consumption is inversely proportional to brake power. According to computer c++ program that written depending on the empirical relation between fuel consumption and speed, the following diagram is drawn.

As shown on the graph the minimum fuel consumption is 0.37kg/kWh which less if compared with pure gasoline.

The shape diagram for both blend fuel and pure gasoline is the same but brake specific fuel consumption is less for E15 at all speed if compared with E0 (pure gasoline).

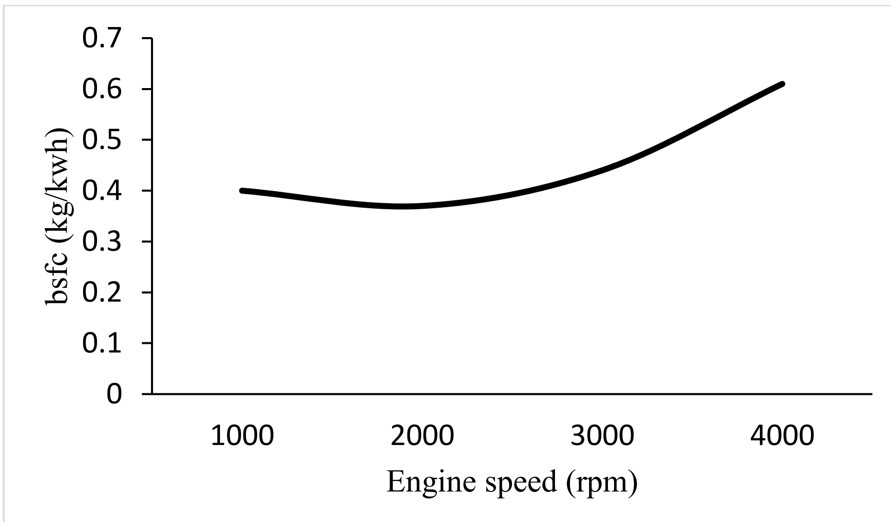


Figure 4.15. Brake specific fuel consumption versus engine speed for E15

4.2.8. Brake mean effective pressure

Break mean effective parameter is one of the important parameter used to determine the brake power and other parameter which is used for perform evaluation.

As break mean effective increase brake power and torque increase. This means they are directly proportional to each other. As shown on the following diagram the maximum brake mean effective is 9.03 bar at 1000rpm.Due to higher thermal efficiency and λ value of blend fuel than pure gasoline the break mean effective pressure of blend fuel is higher than pure gasoline.

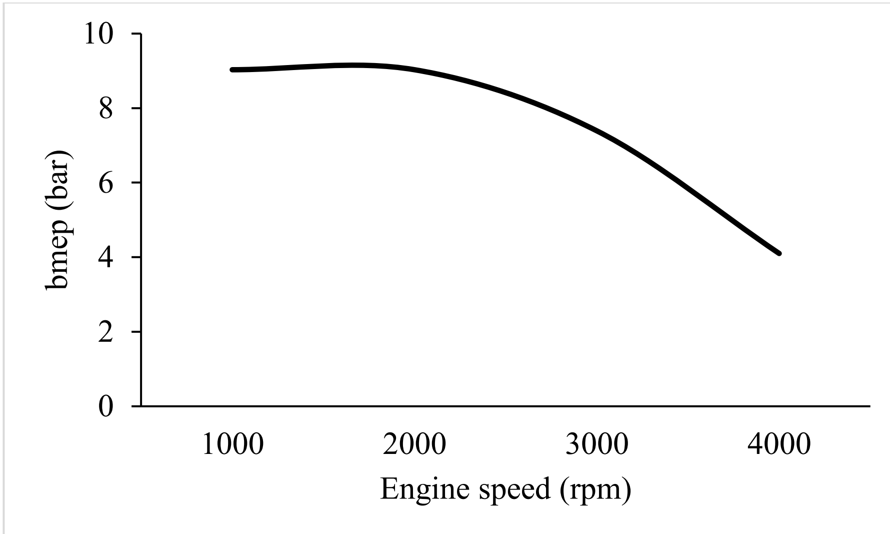


Figure 4.16. Break mean effective pressure vs engine revolution per minute for E15

4.3. Summary of Computer C++ Program Output for E0 - E15

4.3.1. Combustion and thermal efficiency

As shown on the graph below combustion and thermal efficiency increase as the amount of ethanol increase in blend proportion. But according to this computer c++ program it continuously increase only up to E85 and then commence to decrease.

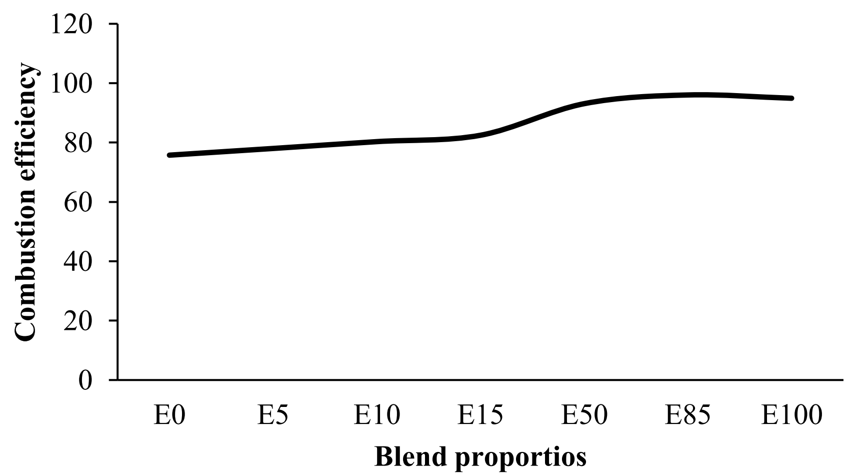


Figure 4.17. Combustion efficiency versus blend proportions

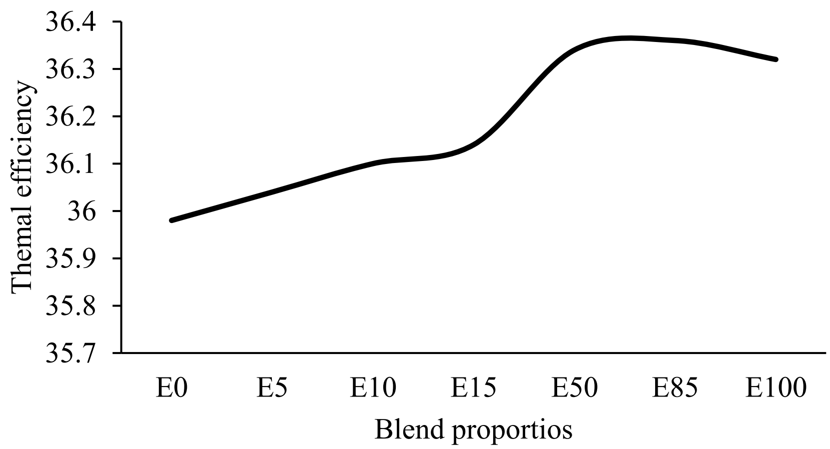


Figure 4.18. Thermal efficiency versus blend proportion

Combustion and thermal efficiency are interdependent to each other. Due to this thermal efficiency also follow the path of combustion efficiency.

4.3.2. Temperature

Because of the high equivalence air-fuel ratio (λ value) of alcohol, the thermal efficiency of blend fuel is greater than pure gasoline. As shown on the figure 4.19 the maximum temperature created by E0, E5, E10 and E15 is 3180k, 3200k, 3110k and 3220k respectively.

4.3.3. Pressure

As the amount of ethanol in gasoline increase the maximum pressure created also increase. As shown on the figure 4.20 the maximum pressure created by E0, E5, E10 and E15 are 55.67bar, 55.99bar, 56bar and 56.32bar respectively.

4.3.4. Pressure vs volume diagram

The pressure volume diagram which depends on the value of peak pressure for E0, E5, E10 and E15 are summarized as fig 4.21.

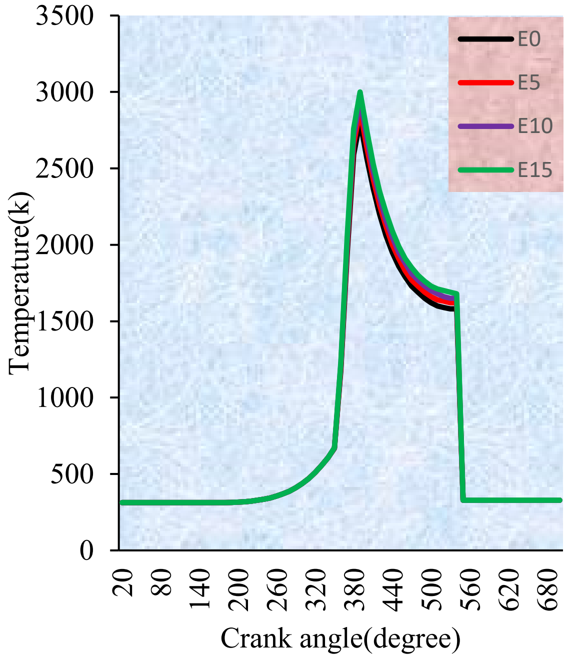


Figure 4.19.Summary of maximum temperature result versus engine crank angle

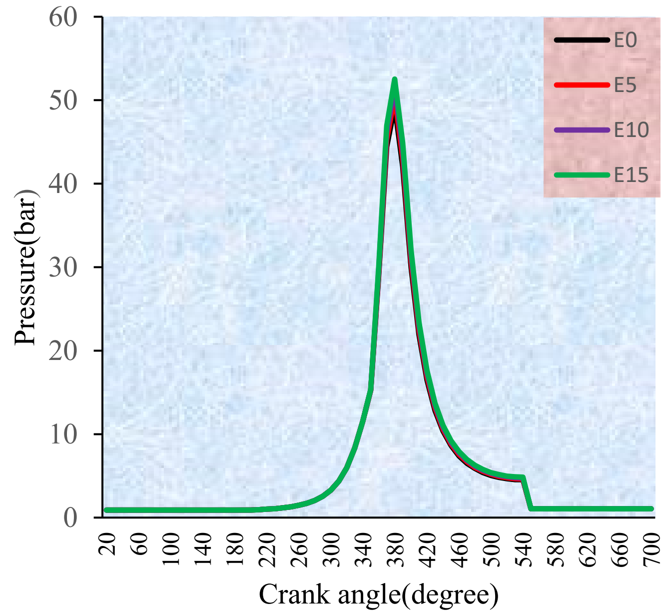


Figure 4.20. Summary of maximum pressure result versus engine crank angle

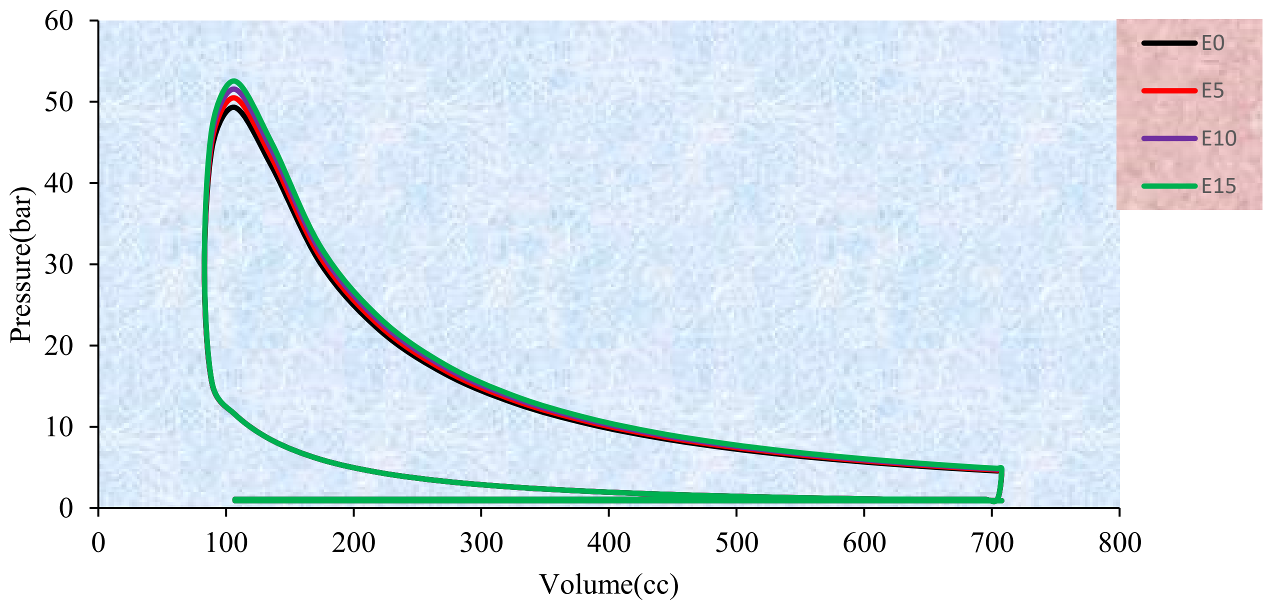


Figure 4.21. Summary of maximum pressure volume diagram

4.3.5. Brake power

According to computer c++ program developed, as the amount of ethanol proportion increase in gasoline the brake power increase. As a result the brake power produced by E0, E5, E10 and E15 are 7.13kw, 7.18kw, 7.22kw and 7.24kw at 3000rpm respectively. One of the factor affecting brake power is the thermal efficiency of the fuel, due to this the increment of the brake power follow the path of it.

Table 4.3. Summary of brake power result from computer c++ program

Speed (N)	Brake power(Pb)			
	E0	E5	E10	E15
0	0	0	0	0
1000	2.9	2.93	2.94	2.95
2000	5.81	5.85	5.88	5.9
3000	7.13	7.18	7.22	7.24
4000	5.28	5.32	5.35	5.36

4.3.6. Brake torque

The brake torque and brake power are directly proportional to each other and they are interdependent. As a result the brake torque produced by E0, E5, E10 and E15 are 27.75Nm, 27.95Nm, 28.1Nmand 28.18Nm at 2000rpm respectively which are above the engine maximum brake toque. So this indicate that the selected engine for test cannot run to lower engine speed.

Table 4.4. Summary of brake torque result from computer c++ program

Speed(N)	Brake torque (Tb)			
	E0	E5	E10	E15
0	0	0	0	0
1000	27.75	27.95	28.1	28.14
2000	27.75	27.95	28.1	28.18
3000	22.71	22.87	22.99	23.06
4000	12.62	12.7	12.77	12.81

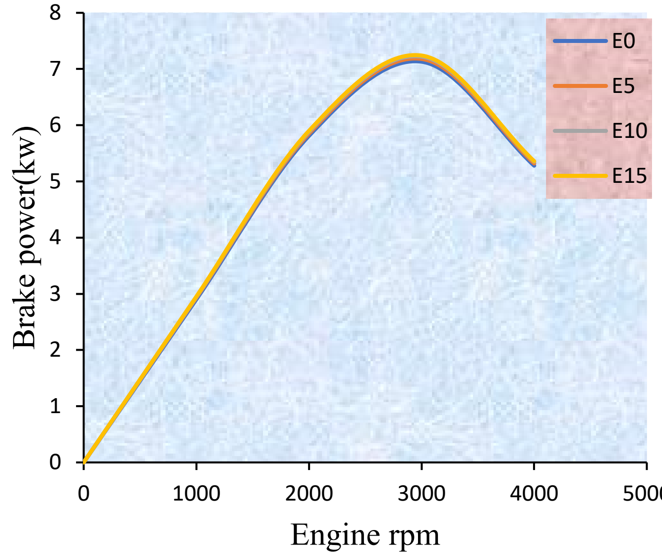


Figure 4.22. Summary of brake power versus engine speed from computer c++ program

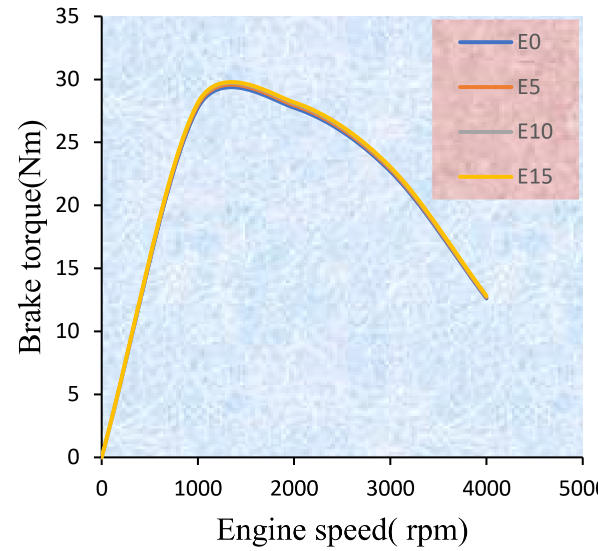


Figure 4.23. Summary brake torque versus engine speed

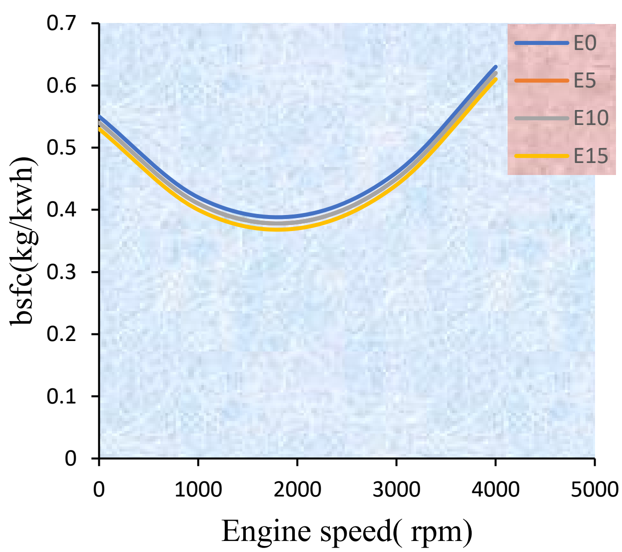


Figure 4.24. Summary brake specific fuel consumption versus engine speed

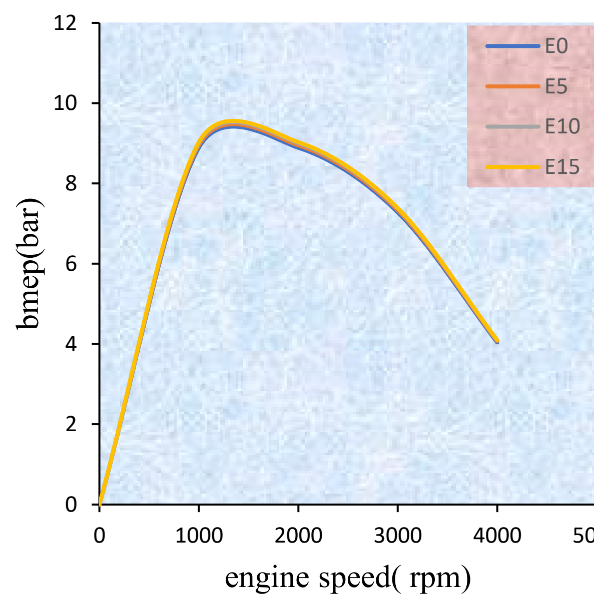


Figure 4.25. Summary brake mean effective pressure versus engine speed

4.3.7. Brake specific fuel consumption

According to the output of this computer c++ program the brake specific fuel consumption decrease if ethanol is blended to gasoline. The minimum brake specific fuel consumption of E0 is 0.39kg/kWh and E5, E10 and E15 are 0.37kg/kWh respectively.

Table 4.5. Summary of brake specific fuel consumption result from computer c++ program

Speed(N)	Brake specific fuel consumption(bsfc)			
	E0	E5	E10	E15
0	0.55	0.54	0.54	0.53
1000	0.42	0.41	0.41	0.4
2000	0.39	0.38	0.38	0.37
3000	0.46	0.45	0.45	0.44
4000	0.63	0.62	0.62	0.61

4.3.8. Brake mean effective pressure

The break mean effective pressure increase as the amount of ethanol increase in blend proportion. Brake mean effective pressure also follow path of the thermal efficiency.

As a result the brake mean effective pressure produced by E0, E5, E10 and E15 are 8.89bar, 8.95bar, 9bar, 9.03bar and at 2000rpm respectively.

Table 4.6. Summary of brake mean effective pressure result from computer c++ program

Speed(N)	Brake mean effective pressure(bmep)			
	E0	E5	E10	E15
0	0	0	0	0
1000	8.89	8.95	9	9.03
2000	8.89	8.95	9	9.03
3000	7.28	7.33	7.37	7.39
4000	4.04	4.07	4.09	4.1

4.3.9. Emissions

In this work five incomplete combustion product was considered even though there is ten product which emitted from internal combustion engine. Considering all combustion product is good, but it needs very complex numerical method. So by using simple simultaneous equation one cannot find the amount of all combustion products. Due to this in this paper only five combustion products are like CO_2, CO, H_2O, N_2 and O_2 are considered. Due to this the mathematical model and the computer c++ is developed for the only five combustion products listed above. As the amount of ethanol increase in blend proportion, the molar fraction emission of CO_2 decrease. This emission depends on the λ value of fuel used. As shown on the figure 4.26 CO_2 emission deceased because of the improvement of the chemical properties of blend fuel.

One of the most dangerous emission from internal combustion engine is carbon monoxide (CO) which is very harmful gas for living things.

This gas is not only harmful but also plays great role in global warming. According to this computer c++ program the emission of carbon monoxide decrease as the amount of ethanol in blend proportion increase.

This is due to the change of λ value as blend proportion increases. Carbon monoxide is a product of incomplete combustion due to insufficient amount of air in the air–fuel mixture.

Table 4.7. Molar fraction of emissions product

blend amount	molar fraction of emission				type of mixture	λ value
	CO2	NOX	CO	O2		
E0	0.19	0.82	0.11	0		0.8
E5	0.18	0.83	0.1	0		0.82
E10	0.18	0.83	0.08	0		0.84
E15	0.17	0.83	0.07	0	rich	0.86
E50	0.08	0.84	0	0	stoichiometric	1
E85	0.11	0.71	0	0.04		1.12
E100	0.1	0.71	0	0.14	lean	1.2

When oxygen ethanol containing is mixed with gasoline, the combustion of the engine becomes better, and therefore CO emission is reduced. As shown on the fig 4.27 the concentration of CO decreased as the volume percentage of ethanol fuel is increased in the fuel mixture.

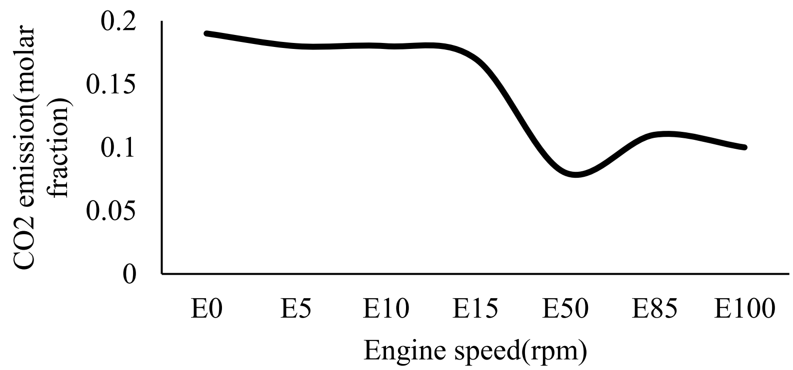


Figure 4.26. Diagram of Co_2 emission versus blend proportions

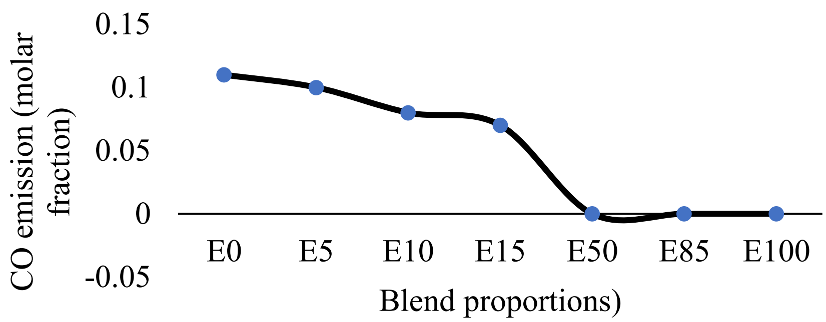


Figure 4.27. Diagram of CO emission versus blend proportions

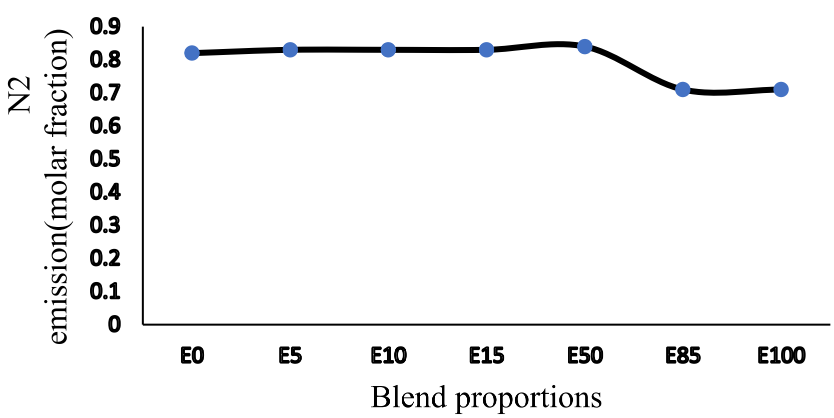


Figure 4.28. Diagram of N_2 emission versus blend proportion

One of the advantage of blend fuel is minimizing positive emission to the environment and promising to keep natural environment.

The molar fraction of N_2 is increased if the type of reaction is rich and deceased if the type of reaction is lean.as shown on the fig 4.28 when the combustion process is closer to stoichiometric, flame temperature increases, therefore, NO_2 formation is expected to be increased.

4.4. Result from experimental analysis for E0 and E15

4.4.1. Result from experimental analysis for E0

Table 4.8. Output of experimental investigation for E0

Worksheet for data acquisition on CT 100.20

Name of tech: Tamiru Tesfaye Date: 22.08.18

Ambient temperature: 24°C

Experiment: Plotting the full load characteristic curve

load	Speed	Torque	Power	Fuel consumption
Full,3/4, 1/2, 1/4	In rpm	In Nm	In kW	in kg/h
Full Load	2200	23.27	5.4	0.6957
	2308	24.51	5.8	0.6573
	2420	25.32	6.1	0.7457
	2560	25.1	6.4	0.7537
	2640	24.16	6.5	0.7645
	2780	23.59	6.6	0.7268
	2908	23.48	7	0.8499
	2950	22.31	7.19	0.8501
	3100	22.12	6.9	0.8657
	3220	21.1	6.3	0.7832
	3360	16.39	4.6	0.8324
	3400	13.56	2.4	0.8858
	3420	12.62	1.7	0.9092
	3440	12.21	1.2	1.2882

4.4.1.1. Brake power

The maximum brake power recorded by using pure gasoline is 7.1kw.This value was the result of experimental investigation and directly taken from the engine test stand.

This maximum brake power was obtained at the 2950rpm.staterting from 2200rpm the brake power increase up to 2950rpm which is the speed at maximum brake power recorded and then decreases up to maximum speed of the engine.

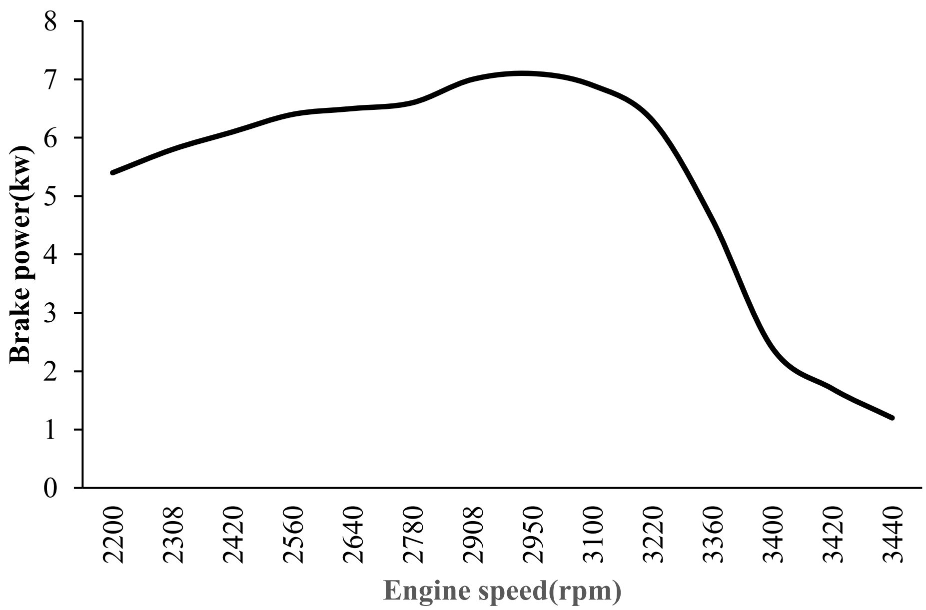


Figure 4.29. Diagram brake power versus engine speed of experimental investigation for E0

4.4.1.2. Brake torque

The brake torque is one of the engine performance parameter that can directly measure from engine dynamometer. Experimental analysis was conducted on the selected engine by using pure gasoline fuel type. As a result the maximum brake torque recorded is 25.32Nm at 2420rpm. This value is somewhat greater than the engine rated brake torque, and happens since engine run at lower engine revolution per minute which is not recommended by manufacturer.

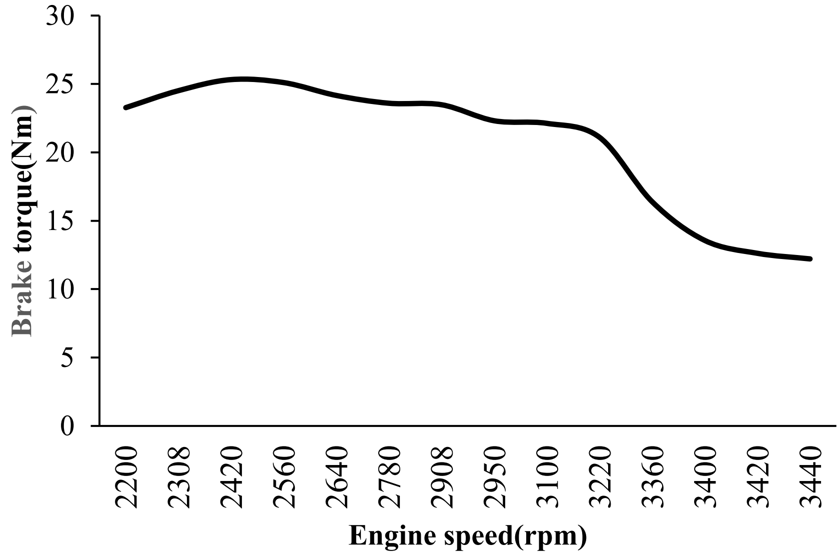


Figure 4.30. Diagram brake toque versus engine speed of experimental investigation for E0

4.4.1.3. Fuel consumption

As shown on the figure 4.31 the amount of fuel consumption fluctuate unlike the expected theoretical value. So to get the value which related true the experiment was conducted three times, and data was recorded by using the same fuel type on selected engine.

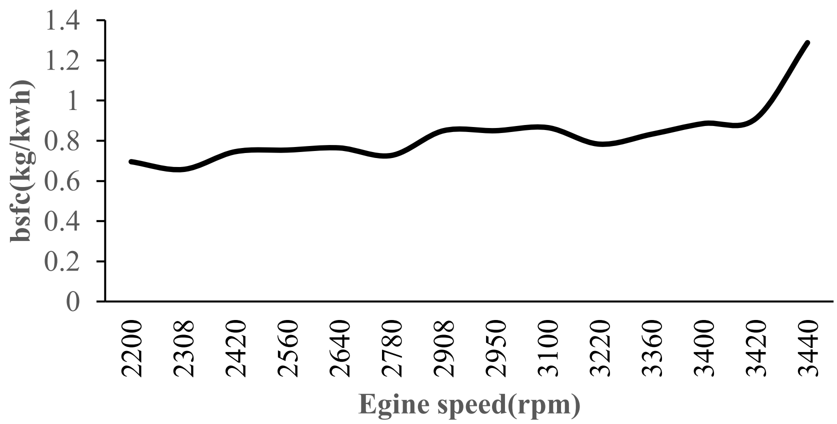


Figure 4.31. Diagram bsfc versus engine speed of experimental investigation for E0

4.4.2. Result from experimental analysis for E15

Table 4.9. Output of experimental investigation for E15

Worksheet for data acquisition on CT 100.20

Name of tech: Tamiru Tesfaye Date: 22.08.18

Ambient temperature: 24°C

Experiment: Plotting the full load characteristic curve with Ethanol Blend 15%

load Full throttle,	Speed In rpm	Torque In Nm	Power In kW	Fuel consumption in kg/h
Full Load	2200	23.57	5.87	0.6832
	2308	24.62	6.1	0.6487
	2420	25	6.21	0.7238
	2560	25.51	6.68	0.7485
	2640	24.34	7.08	0.7632
	2780	23.9	7.21	0.7686
	2908	23.45	7.34	0.8573
	2950	22.28	7.41	0.865
	3100	22.23	7.3	0.873
	3220	21	6.89	0.8768
	3360	16.79	4.98	0.882
	3400	14.11	3.35	0.8998
	3420	13.32	1.62	1.1992
	3440	12.47	1.28	1.322

4.4.2.1. Brake power

As shown fig 4.32 the maximum brake power recorded by using pure gasoline is 7.41kw.This value was investigated experimentally, performance of a single cylinder four-stroke spark-ignition engine fueled with ethanol-gasoline blends. The tests was conducted at thirteen different engine speeds ranging from 2200 to 3440 rpm at same compression ratio (8.5:1) and maximum brake power was obtained at the 2950rpm.

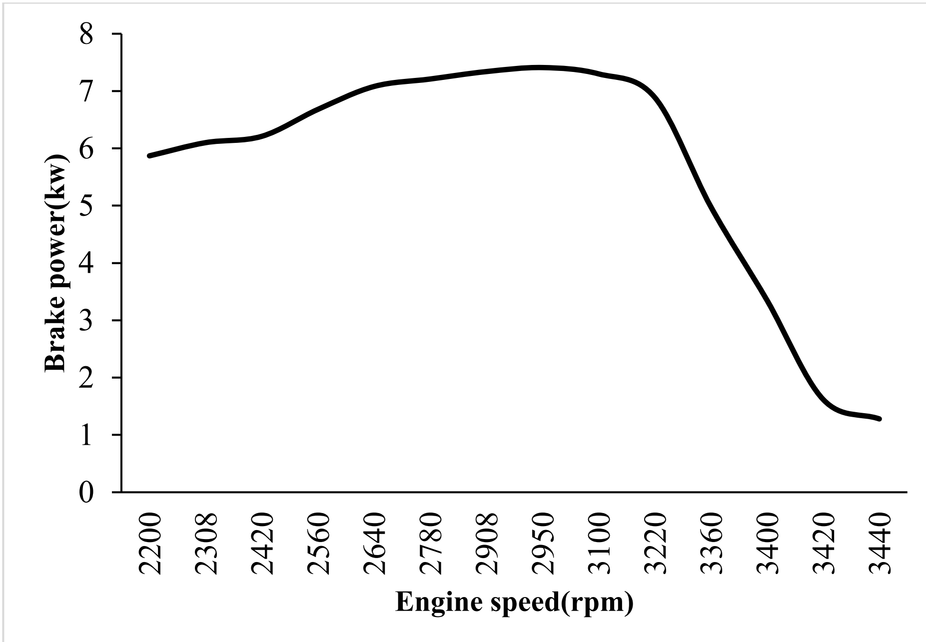


Figure 4.32. Brake power versus engine speed experimental investigation for E15

4.4.2.2. Brake torque

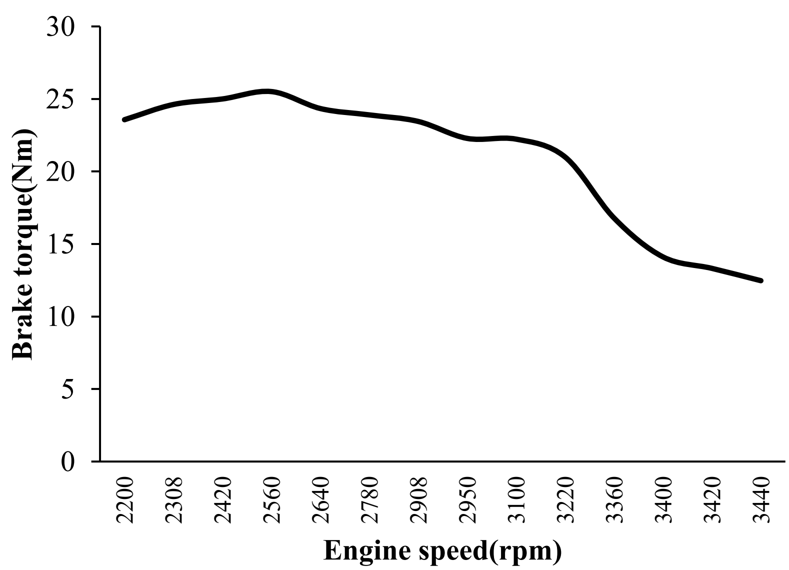


Figure 4.33. Brake torque versus engine speed experimentally investigated for E15

Like that of pure gasoline the brake torque for blend fuel was experimentally investigated. The maximum brake torque created on this selected engine by using E15 (15% ethanol and 85% gasoline) was 25.51Nm at 2560rpm. Due to higher thermal efficiency for blend fuel the brake power of E15 is higher than E0. This brake torque is directly recorded from engine test stand and investigated at all engine speed tested for E0. The fig 4.33 shows the relation between brake torque and engine speed experimentally investigated for E15.

4.4.2.3. Fuel consumption

As shown on the figure 4.34 the fuel consumption for E15 was experimentally investigated and recorded for all speed. The minimum fuel consumption recorded was 0.64kg/h at 2308 rpm.

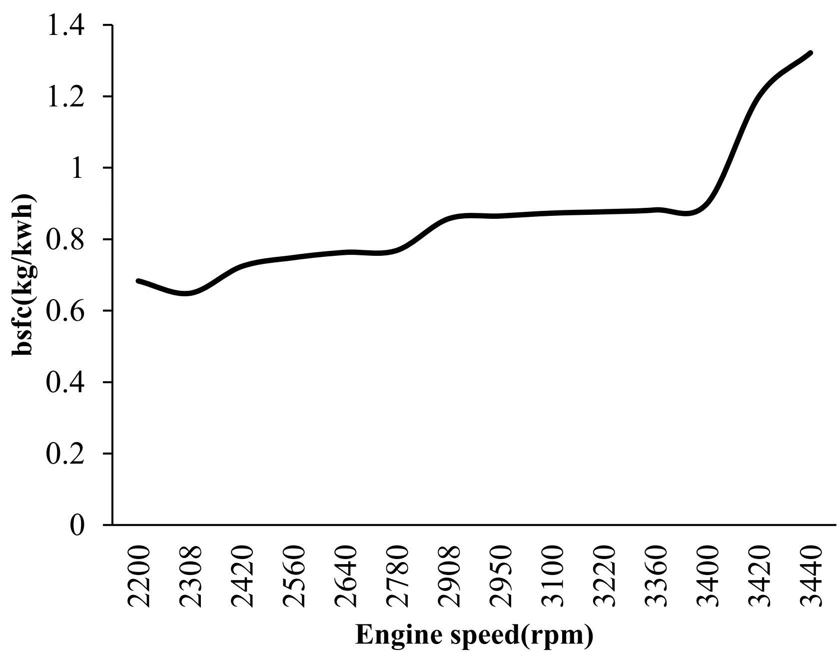


Figure 4.34. Fuel consumption versus engine speed experimental investigation for E15

4.4.3. Summary from experimental analysis for E0 and E15

4.4.3.1. Brake power

When the ethanol content in the blend fuel is increased, the engine brake power is somewhat increased for all engine speeds (Fig.4.35). The improvement of the engine power was due to the rise of the indicated mean effective pressure and the in cylinder pressure due to the higher ethanol content in the blends. The heat of evaporation of ethanol is higher than that of gasoline, this provides fuel air charge cooling and increases the density of the charge, and thus higher power output is obtained shrma. (2015). The maximum brake power for both blend fuel and pure petrol occurred at the same engine rpm.

4.4.3.2. Brake torque

The raise of ethanol content will increase the torque of the engine. Added ethanol will produce lean mixture that increases the relative air-fuel ratio (λ) to a higher value and makes the burning more efficient. The improved anti-knock behavior (due to the addition ethanol, which raised the octane number) allowed a more advanced timing that result in higher combustion pressure and thus higher torques. As shown on fig 4.36 the maximum brake torque of E15 blend fuel is higher than pure gasoline at all engine speed.

4.4.3.3. Fuel consumption

The effects of ethanol–gasoline blends on fuel consumption vs. engine speed are shown in Figure 4.37. As shown in this figure, the fuel consumption decreases as the ethanol content increase from E0 to E15. This minimization of fuel consumption is continues only up to 2640 rpm and commence to increase up to maximum engine rpm. This is due to the increase in brake thermal efficiency and decreases in equivalence air fuel ratio (φ).

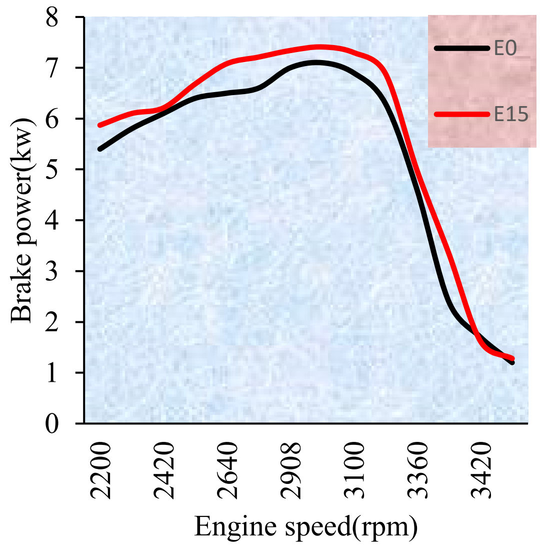


Figure 4.35. E0 and E15 brake power versus engine speed for experimental investigation for E0 and E15

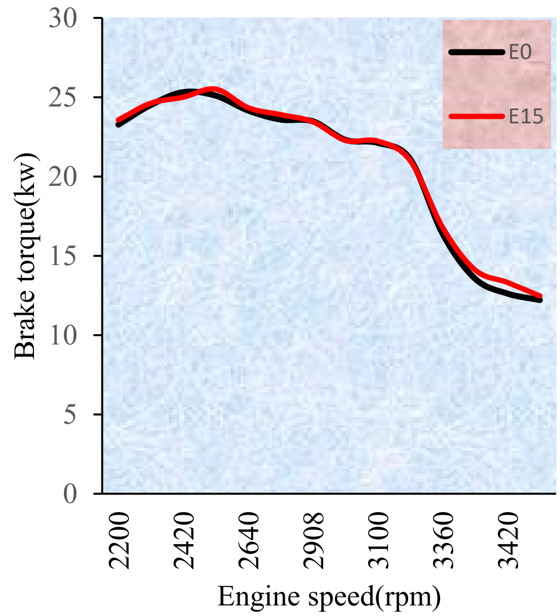


Figure 4.36. Summary of brake torque versus engine speed from experimental investigation for E0 and E15

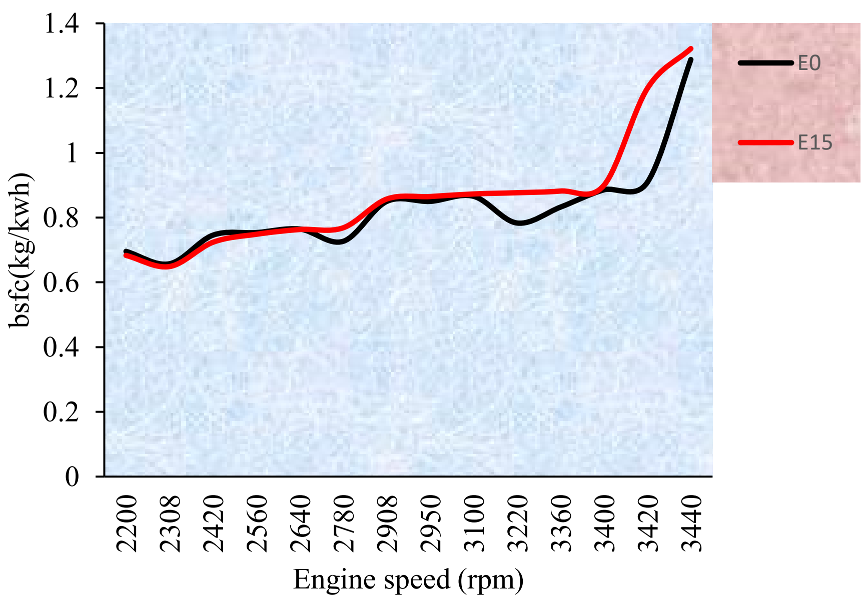


Figure 4.37. Fuel consumption vs engine speed experimental investigation for E0 and E15

4.4.4. Emission result from experimental investigation of E0 and E15

The experimental investigation on emission was conducted on both E0 and E15 and the result is tabulated in table 4.6 and 4.7 as follows.

Table 4.10. Emission experimental investigation of E0

Exhaust Emission (%)				
Engine Speed (rpm)	Fuel Sample			
	E-0			
	CO	CO_2	HC(ppm)	O_2
3450	0.132	1.58	17	16.99
3200	0.099	1.52	15	17.20
3000	0.092	1.45	13	17.25
2800	0.082	1.37	13	17.34
2600	0.073	1.44	13	17.44
2400	0.065	1.23	13	17.57
2200	0.053	1.22	15	17.68

Blending ethanol to gasoline is the best way to minimize the amount of green house and harmful gases emission to the environment.

According to experimental investigation and shown on the figure 4.38 and 4.39, emission of CO and CO_2 increase with speed. But emission of HC is higher at lower and higher speed while lower at the medium speed.

Table 4.11. Emission experimental investigation of E15

Exhaust Emission (%)				
Engine Speed (rpm)	Fuel Sample			
	E-15			
	CO	CO_2	HC	O_2
3450	0.112	1.45	16	16.88
3200	0.093	1.37	15	16.98
3000	0.087	1.30	13	17.15
2800	0.070	1.28	12	17.29
2600	0.071	1.22	12	17.37
2400	0.059	1.23	13	17.51
2200	0.052	1.16	14	17.70

CO:

The most important parameter that affects CO emission is the air fuel equivalence ratios. With air fuel equivalence ratio approaching unity, CO emission diminishes, and even becomes zero in lean condition.CO emissions are likely to be reduced for various blends due to the oxygen enrichment coming from ethanol.

CO_2:

Figure 4.39 represents the variation of carbon dioxide vs. engine speed. It is clear from figure that the CO_2 concentration decreases as the ethanol percentage increased. CO2 emissions depend on relative air-fuel ratio.

HC:

Figure 4.40 represents the hydrocarbon vs. Engine speed. It is clear from the figure that HC emission was reduced with increasing speed. Further, with increasing percentage of ethanol in gasoline results reduced HC emissions. The concentration of HC emission decreases with the increase of the relative air fuel ratio. The air–fuel mixing in case of ethanol blend enhances the combustion and complete combustion take place which result lower HC concentration.

O_2:

The emission of O_2 decreases with increasing engine speed. This imply that as the engine speed increase the equivalence air fuel ratio decrease due to this the emission of O_2 also decrease. Generally according to experimental investigation the emission of CO, CO_2, HC and O_2 is lower for E15 than E0.

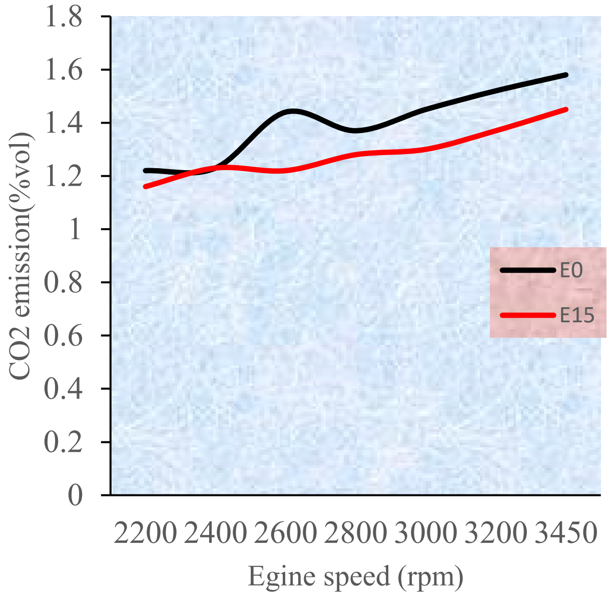


Figure 4.38. Experimental investigation of CO_2 versus engine speed for E0 and E15

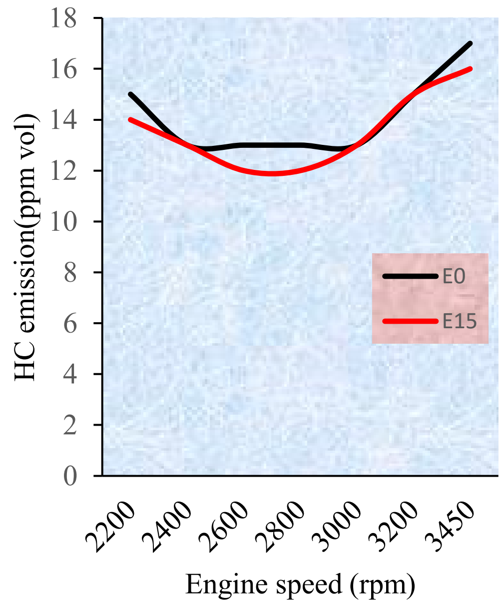


Figure 4.39. Experimental investigation HC versus engine speed for E0 and E15

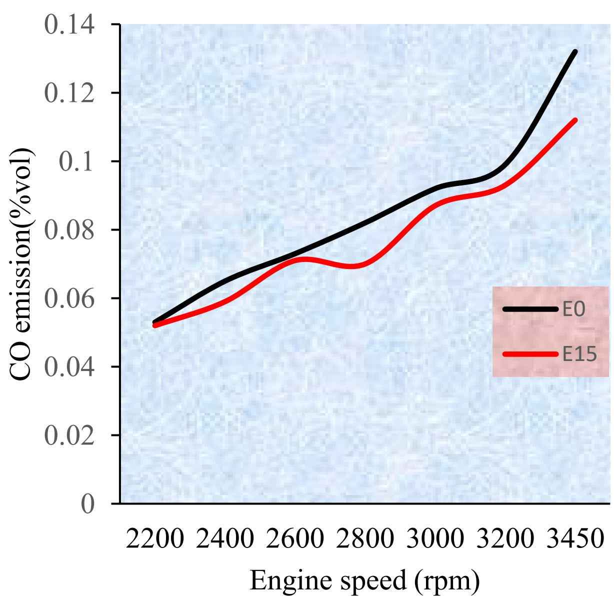


Figure 4.40. Experimental investigation of CO versus engine speed for E0 and E15

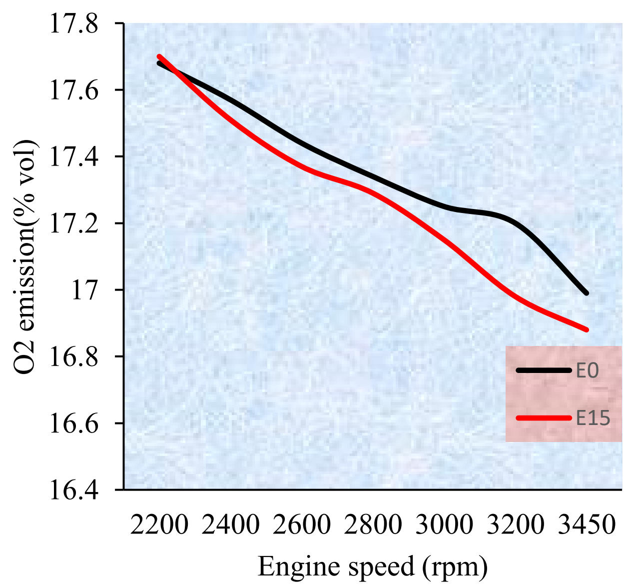
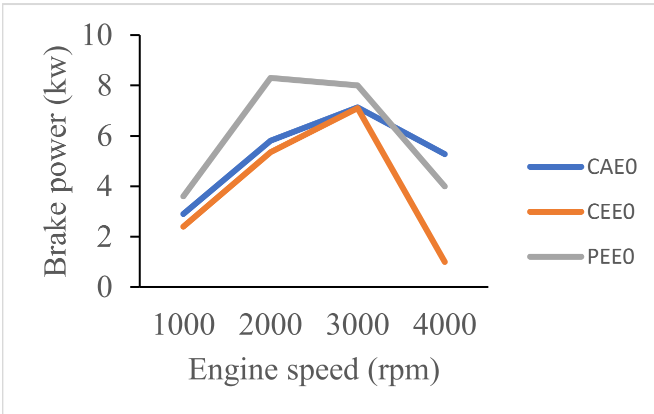
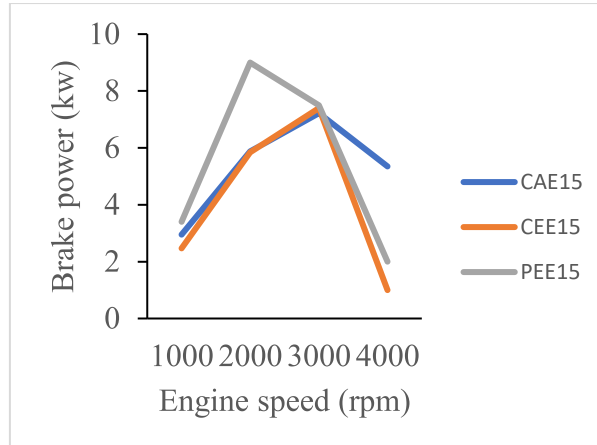


Figure 4.41. Experimental investigation of O_2(% vol) versus engine speed(rpm)

4.5. Validation of the computer c++ program and experimental investigation result with previous work.



Brake power for E0.



Brake power for E15.

Where,

CAE0 is current analytical work for E0

CEE0 is current experimental work for E0

PEE0 is previous experimental work for E0 Kumar et al.(2016)

Where,

CAE15 is current analytical work for E15

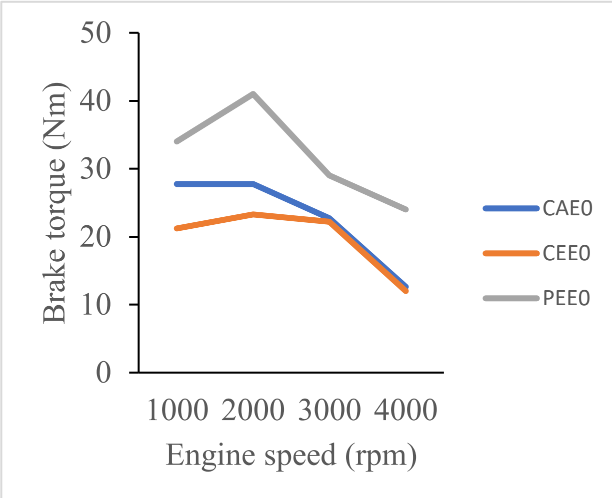
CEE15 is current experimental work for E15

PEE0 is previous experimental work for E15 Kumar et al.(2016)

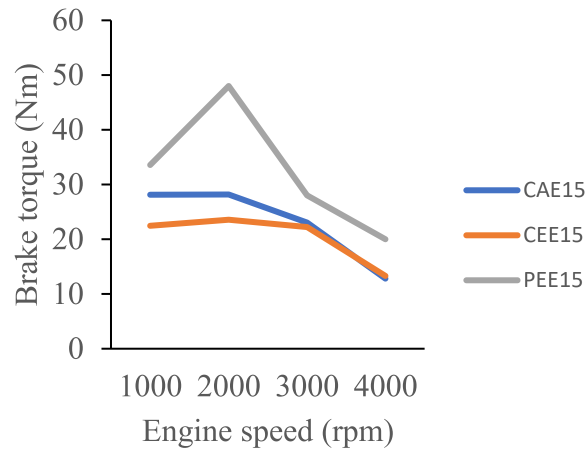
To validte the output of both computer c++ program and expermental invistigation the similar previous work was selected.

Kumar et al.,(2016) When the ethanol content in the blend fuel is slightly increased for all engine speed, the gain of the engine power was due to the increases of the indicated mean effective pressure.

As shown on the above figures the analytical, experimental for the current work and previous work are following the related trend. But since the engine of the previous work and the current one is not the same the output of them are not same due to this the line of the graph were not coincide each other.



Brake torque for E0



Brake torque for E15

Where,

CAE0 is current analytical work for E0

CEE0 is current experimental work for E0

PEE0 is previous experimental work for E0 Kumar et al.(2016)

Where,

CAE15 is current analytical work for E15

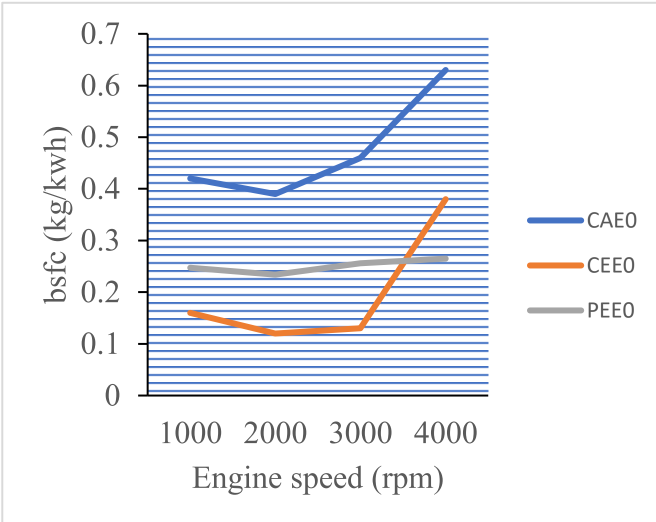
CEE15 is current experimental work for E15

PEE0 is previous experimental work for E15 Kumar et al.(2016)

To validte the output of both computer c++ program and expermental invistigation the similar previous work was selected.

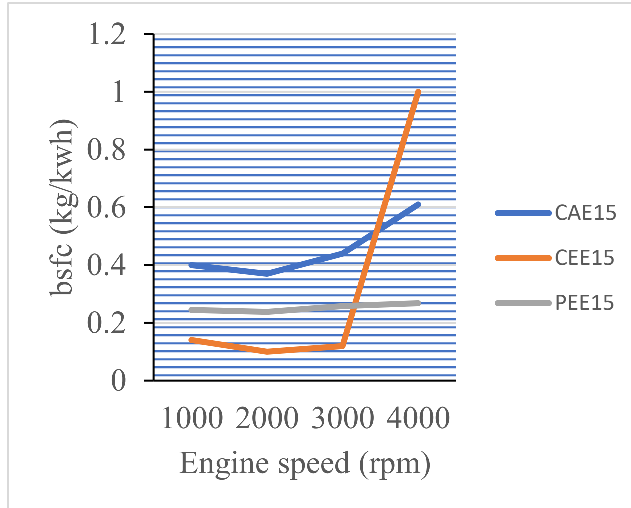
Kumar et al.,(2016) When the ethanol content in the blend fuel is slightly increased for all engine speed, the gain of the engine maximum torque was increase.

As shown on the above figures the analytical, experimental for the current work and previous work are following the related trend. But since the engine of the previous work and the current one is not the same the output of them are not same due to this the line of the graph were not coincide each other.



Brake specific fuel consumption for E0

Where,
CAE0 is current analytical work for E0
CEE0 is current experimental work for E0
PEE0 is previous experimental work for E0
Kumar et al.(2016)



Brake specific fuel consumption for E15

Where,
CAE15 is current analytical work for E15
CEE15 is current experimental work for E15
PEE0 is previous experimental work for E15
Kumar et al.(2016)

To validte the output of both computer c++ program and expermental invistigation the similar previous work was selected.

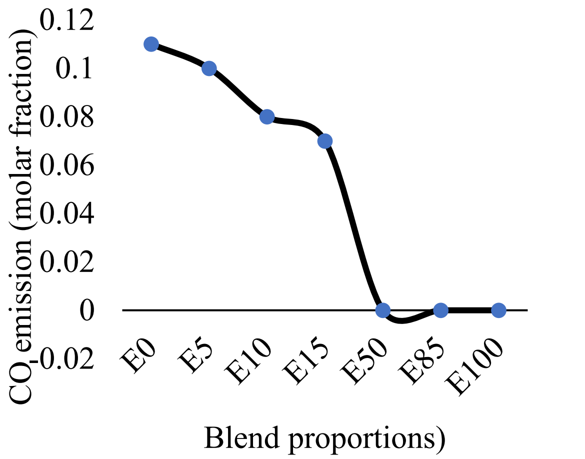
Kumar et al.,(2016) When the ethanol content in the blend fuel is slightly increased the minimum bsfc decreases up to 15% of ethanol in blend fuel.

As shown on the above figures the analytical, experimental for the current work and previous work are following the related trend. But since the engine of the previous work and the current one is not the same the output of them are not same due to this the line of the graph were not coincide each other. The value of bsfc from previous work related to each other on selected engine speed due to this the seams straight line but is not. The result of analytical and experimental work are not related to each other because filling data on experimental one is manual and possibly cause for recording wrong value. In this case the stop watch and analog fuel metering device used to record the fuel flow rate due to this the data was recorded by wrong way and the variation with analytical one is created.

Source of previous work was kumar et al.,(2016).

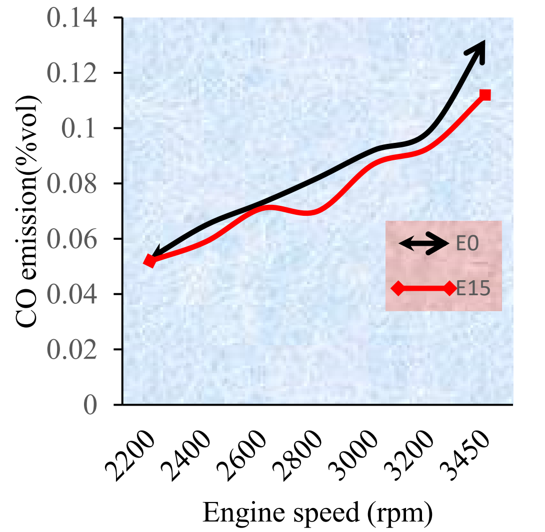
Table 4.12. Validation of brake torque out with previous work.

<table>
<tr>
<td>



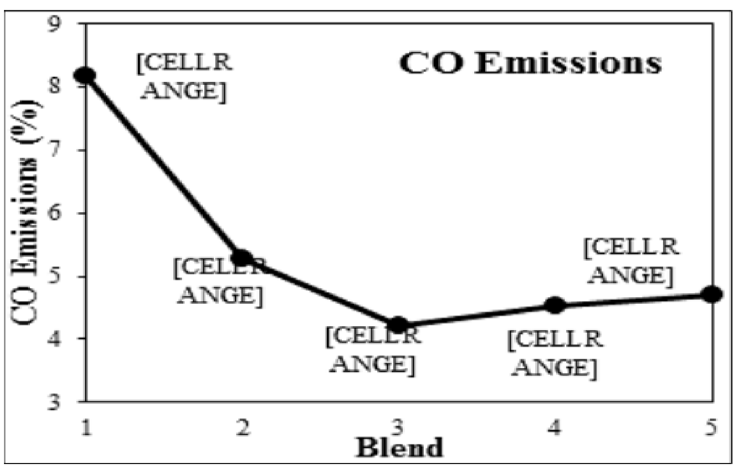
This graph is the output of computer c++ program and show the emission of CO decreases as the amount of ethanol increase in blend proportion and become zero if the equivalence air fuel ratio is unity and above it.

</td>
<td>



This graph is the output of experimental investigation and shows that, the emission of CO decreases as blend proportion increase from E0 to E15.

</td>
</tr>
<tr>
<td colspan="2">



Joshi et al., (2015) with air–fuel equivalence ratio approaching unity, CO emission diminishes, and even becomes zero in lean condition. CO emissions are likely to be reduced for various blends due to the oxygen enrichment coming from ethanol.

</td>
</tr>
</table>

The emission of hydrocarbon is not analyzed by computer c++ program and NO_X is not experimentally investigated due to this both of them are not compared with the previous research result.

CHAPTER FIVE

5. CONCLUSION AND RECOMMENDATION

Under this chapter summary of the all work in this paper, conclusion of the result and discussion and the recommendation for the future researcher are explained.

5.1. Summary

After the title was identified depending on the existing gap or problem the objective of research means general and specific objective of research was identified. The benefit and scope of this work was also recognized. Different literature which has relation with selected title was reviewed and summarized. To accomplish the objective of research, different methods such as preparing mathematical model, developing computer c++ program and executing experimental investigation are used. Depending on the result from both computer c++ program and experimental investigation enough discussion was done on each topic. Finally the result was concluded and the recommendation was listed for the future researcher.

5.2. Conclusion

In this work the thermodynamic analysis of actual engine cycle with ethanol gasoline blend fuel (E15) is done. Depending on the thermodynamic analysis which can be the mathematical model, the computer c++ program was developed and performance parameter such as brake power, brake torque, thermal efficiency, break mean effective pressure, combustion efficiency, brake specific fuel consumption and emission was analyzed by using developed computer c++ program. This program is versatile that can be functional for every type of engine and any type of blend proportions. As a result the brake power, brake torque, combustion efficiency, thermal efficiency and break mean effective pressure are higher for E15 than E0 and the brake specific fuel consumption is lower for E15 than E0. Therefore the brake power torque for E15 is increased by 1.6% from E0. But the fuel consumption decreased by 2.4%, 4.35% and 3.15% at lower speed, maximum power and maximum speed respectively for E15 than E0.

The molar fraction emission of CO_2, CO and water vapor is less for E15 than E0 but molar fraction of N_2 is higher for E15 than E0.

To validate and prove the result of computer c++ program the experimental investigation was done on selected engine.

The motivation of performance parameter gained by computer c++ program is also proper for experimental investigation. According to the experimental investigation the brake power and brake torque for E15 is higher than that of E0, but the fuel consumption of E15 is less than E0. In experimental investigation the brake power and brake torque for E15 is increased by 2.96% and 1.6% respectively. The minimum fuel consumption for E15 is lower by 1.3% than E0 at 2308rpm, higher for E15 by 1.7% than E0 at maximum brake power (2950rpm) and higher for E15 by 2.5% than E0 at maximum speed (3440rpm).

The emission value which gained by computer c++ program and experimental investigation was almost similar.

According to experimental investigation the emission of CO, CO_2, HC and O_2 is lower for E15 than E0. The emission of CO, CO_2, HC increase with increase of engine speed and O_2 emission decreases.

The deviation between result of developed computer c++ program and experimental investigation was less than 5% for all analyzed engine parameter.

5.3. Recommendation

After conclusion of this work the following listed points are very important that can enhance the trustworthiness of the outcome of research.

- For the future work it is possible to recommend the researcher to investigate further on this topic because there is no common conclusion between researchers executed work concerning the topic.
- Computer c++ program developed in this work can easily identify the performance parameter, calculate the value of material charged to combustion

and identify the amount of emission so before testing on real engine using computer c++ program is cost efficient and highly support the exponential investigation process.

- Depending on the real factor which may be didn't consider in this work while preparing mathematical model and develop computer c++ program, the future researcher can reshuffle and come out with perfect result.
- Using the engine of higher torque is better to experimentally investigate at lower engine rpm.
- At one test time one cannot get the precise value so testing a number of time is very important.
- Before experimental investigation calibrating the engine and knowing the physical and chemical characteristic of fuel (for both ethanol and gasoline) can support for the correct reasoning of result.
- Studying all factor affecting engine performance parameter is very important to identify crucial one and optimize the result.
- The emission is not limited to only five combustion product so studying all of them are enhance the reliability of emission amount.

REFERENCES

A.Samuel Raja , A. Valan Arasu, T. Sornakumar (2015). Effect of gasoline ethanol blends on performance and emission characteristics of a single Cylinder Air Cooled Motor bike si engine. Journal of Engineering Science and Technology , Vol. 10(No.12), 1540 - 1552.

Ahmed Mostafa Ahmed, Ismail Youssef, Mohamed Mourad. (2017). The Influence of Ethanol–Gasoline Blends on Performance Characteristics of Engine Generator Set. American Journal of Engineering Research (AJER), Volume-6(Issue-9), p-ISSN : 2320-0936.

Ahmet Necati Ozsezen . (2012). Performance and combustion characteristics of alcohol-gasoline blends at wide-open throttle.

AKolchin and R Demidw. (1984). Design Of Automotive Engines, pp.47-78.,

Alvydas Pikūnas, Saugirdas Pukalskas, Juozas Grabys. (2003). Nfluence Of Composition Of Gasoline – Ethanol Blends On Parameters Of Internal Combustion Engines. Journal of KONES Internal Combustion Engines, vol. 10(3-4).

Amit Kumar Thakur, Ajay Kumar Kaviti, Roopesh Mehra & K. K. S. Mer (2017) Performance analysis of ethanol gasoline blends on SI engine:review,Biofuels,8:1,91112,DOI:10.1080/17597269.2016.120 4586

Angelo Basile, Adolfo Lulianelli, Francesso Dalena, T.Nejat Veziroglu. (2018). Ethanol science and Engineering 1st Edition, eBook ISBN: 9780128115367.

Asad, Islam. (2016). Simulation of Four Stroke Internal Combustion Engine. International Journal of Scientific& Engineering Research, ISSN 2229-5518.

Ashraf Elfasakhany. (2015). Investigations on the effects of ethanol-methanol-gasoline blends in a spark-ignition engine: international jouranal of science and technology,no 18,pp 713-719.

Deepak Kumar and Ajay Trehan.(2016). Experimental Investigation of SI Engine Performance using Ethanol-Gasoline Blended Fuels. International Journal of Scientific & Engineering Research, Volume 7 (Issue 12), ISSN 2229-5518.

Julian Happian-Smith (2002). An Introduction to Modern Vehicle Design.pp.371-401.

F. Kheiralla, Mohamed El-Awad, Mathani Y. Hassan, M. A. Hussen, and Hind I. Osman.(October2011)). Effect of Ethanol−Gasoline Blends on Fuel Properties Characteristics of Spark Ignition Engines. UofKEJ, Vol. 1(ssue 2), I pp. 22-28.

Gaurav Sharma* and Harvinder lal. (2015). Effects of Ethanol-Gasoline Blends on Engine Performance and Exhaust Emissions in a Spark Ignition, International Journal on Emerging Technologies., 6(2), 184-188.

Pham Huu Truyen, Pham Huu Truyen, Pham Huu Tuyen, Pham Minh Tuan, Le Anh Tuan .(2012). Influence Of E10, E15 And E20 Fuels Onperformance And Emissions Of In-Use Gasoline Passenger Cars. ASEAN Engineering Journal Part C, Vol 4(No 2), ISSN 2286-8150.

Willard W. Pulkrabek. (1988). Engineering Fundamentals of the Internal Combustion,pp.81-86.

M. S. M. Zaharin, N. R. Abdullah, M. Imran Dahalan, Hazim Sharudin, A.R. Asiah, M. Beriache. (2017). Experimental Study on the Effects of Methanol and Ethanol on Gasoline Engine Performance and Exhaust Emissions. Journal of Mechanical Engineering, Vol 2(SI), 27-44.

Juan Miguel, Diego Alexande, Carlos Humberto. (2015). Multivariate analysis of performance and emissions for internal combustion engines running with gasoline-ethanol blends. Ingeniería Energética, XXXVI(3), p.232-242 ISSN 1815 - 5901.

K., T. J. (2011). Physico-chemical properties of bio-ethanol/gasoline blends and the qualitative effect of different blends on gasoline quality and engine performance. Journal of Petroleum Technology and Alternative Fuels, Vol. 2(3), pp. 35-44.

Melis Teka . (2007). Biofuel Strategy of Ethiopia. East African Power Industry Convention . Addis Ababa.

Ramachandran, S. (2009). Rapid Thermodynamic Simulation Model of an Internal Combustion Engine on Alternate Fuels. International Multi Conference of Engineers and Computer Scientists. Hong Kong.

Ramesh Babu Nallamothu,Geleta Fekadu, & Prof B.V. Appa Rao. (2013). Comparative Performance Evaluation Of Gasoline And Its Blends With Ethanol In Gasoline Engine. Global Jornal Of Biology, Agriculture,And Health Science,, ISSN: 2319 – 5584.

S. Babazadeh Shayan, M. Seyedpour, F. Ommi, S. H. Moosavy and M. Alizadeh. (2011). Impact of Methanol–Gasoline Fuel Blends on the Performance and Exhaust Emissions of a SI Engine. International Journal of Automotive Engineering, Vol. 1,(Number 3,).

Sandeep M. Joshi1, Raahul Krishna, Balagouda A. Patil, Aneesh C. Gangal,Performance.(2014) Evaluation of Petrol-Ethanol Blends and Testing on Multi cylinder 4-Stroke SI Engine ,Journal of Alternate Energy Sources and Technologies Volume 6, Issue 1ISSN: 2230-7982(online), ISSN: 2321-5186

TalalYusaf, G Najafi, and David Buttsworth. (2014). Theoretical and experimental investigation of SI engine performance and exhaust emissions using ethanol-gasoline bend. (http://www.research gate).

United State Department Energy. (2016) Handbook for Handling, Storing, and Dispensing E85 and Other Ethanol Blends, Doe/go-102016-4854.

V. Saikrishnan, A. Karthikeyan & J. Jayaprabakar. (2018) Analysis of ethanol blends on spark ignition engines, International Journal of Ambient Energy,39:2,103107,DOI:10.1080/01430750.2016.1269678

Viral K Pandya, Shailesh N Chaudhary, Bakul T Patel, Parth D Patel. (Nov 2011). Experimental Study On The Effect Of Methanol Gasoline ,Ethano-Lgasoline and N-Butanolgasoline Blends On The Performance Of 2-Stroke Petrol Engine. International Journal of Advances in Engineering & Technology.

Vivek Pandey and V.K.Gupta. (2016). Experimental analysis of performance characteristics of an SI engine on ethanol-gasoline blended fuel. International Journal of Engineering Development and Research, Volume 4, Issue 3 | ISSN: 232.

Wikipedia, (2007). Ethanol fuel and Ethanol Fuel in Brazil last modified 8 January 2007, last modified 10 January 2007.

Wikipedia, (2009). Corn ethanol. http://en.wikipedia.org/wiki/Corn_ethanol.

Xiaochen Wang , Zhenbin Chen , Jimin Ni , Saiwu Liu , Haijie Zhou .(2015) The effects of hydrous ethanol gasoline on combustion and emission characteristics of a port injection gasoline engine. case study in thermal engineering ,vol 6,pp 147-154.

Yan Zhang , Hua Zhao, Mohammed Ojapah, Alasdair Cairns.(2013) .CAI combustion of gasoline and its mixture with ethanol in a 2-stroke poppet valve DI gasoline engine, Fuel 109 ,pp 661–668

APPENDEICES

Appendix. 1

Developed computer c++ program for calculating temperature, pressure and volume

```
// Actual Engine Cycle - SI Engine (AEC-SI1).
#include<iostream.h>
#include<iomanip.h>
#include<conio.h>
#include<math.h>
#include<fstream.h>
 void main()
  {
    float
Ec_cc,Zc,Rsd,Rrl,a,b,RHO_kgplit,QLHV,ZC,ZH,ZO,RHO_Alkgplit,QLHV_Al;
    float
xAl_per,xR_per,LAMBDA,pim,p1,T1,Rc,N,ETA_mper,theta_ivobTDC;
    float
theta_ivcaBDC,theta_scbTDC,theta_dc,theta_evobBDC,theta_evcaTDC;
    float
Zs,Z_rpc,ETA_m,xR,theta_a,theta_b,theta_ivc,theta_1,theta_sc;
    float theta_2,theta_ec,theta_3 =
theta_ec,theta_evo,theta_4,theta_5;
    float
theta_6,theta_0,Ec_liter,Ec,Vs,Vcl,Vt,Vs_cc,Vcl_cc,Vt_cc,d,d_mm;
    float
powd,s,s_mm,r,r_mm,lcon,lcon_mm,Ap,g,Ru,mA,mF,mres,Tim,theta_c;
    float
a1O2r,a2O2r,a3O2r,a4O2r,cpO2,cvO2,a1N2r,a2N2r,a3N2r,a4N2r,cpN2;
    float
cvN2,cpA6b,cvA6b,MmA,cp6b,cv6b,R6b,n6b,RHO6b,v6b,Z6b,Z1,V1,R1;
    float
mm,MmO,MmN,MmC,MmH,yO2,yN2,MmO2,MmN2,xAl,xG,xF,MmG,MmAl,QLHV_G;
    float nAl,nG,nAlG,yAl,yG,yF,yAl_per,MmF,n1th,n2th,atha,ath,n3th;
    float
n10,n3,n1r,n1,n2,nt,y1,y2,y3,y10,MmAth,MmFth,nO2,nN2,nA,AFth,AF,QLHV
_F;
    float
RHO_Fkgplit,RHO_F,ETA_cmax,ETA_c,ETA_cper,T6bi,RHO_Gkgplit;
    float Rm,Zm,Vm,n4,n5,y4,y5,yAL,Mm1,Mm2,Mm3,Mm4,Mm5,nF,Tb_NM;
    const float PI = 3.1416; g = 9.81; Ru = 8.314;
// process 0-b (constant pressure intake process).
    float
V0bi,p0bi,T0bi,m0bi,R0bi,rad,Z0b,xp0b,Vs0b,V0b,p0b,T0b,R0b,m0b;
    float Vb,pb,Tb,mb,thetab,deltheta,theta,mb1i;
// process b-1 (polytropic compression):
    float
Vb1i,pb1i,Tb1i,Rb1i,xpb1,Vsb1,Vb1,delmb1,VbVb1,nb1,nb1T,Rb1;
    float Zb1,pb1,Tb1,mb1,theta1;
// process 1-2 (polytropic effective compression):
    float m12i,V12i,p12i,T12i,xp12,Vs12,V12,V1V12,n12,n12T,R12,Z12;
    float p12,T12,m12,V2,p2,T2,theta2;
```

```
// process 2-3 (polytropic combustion):
    float V23i,p23i,T23i,T23,W23i,xp23,Vs23,V23,m23,theta2dc,rad2dc;
    float xb,Qin,Q1,Q23,cpO2o2,R23;
    float cpN2o2,cpFo2,cpr2,cvr2,Mmr2,cp2,cv2,R2,m2,RHO2;
    float
v2,hfoF,hfoO2,hfoN2,delhFo2,delhO2o2,delhN2o2,UrFo2,UrO2o2,UrN2o2;
    float
Ur2,Ur,a1CO2p,a2CO2p,a3CO2p,a4CO2p,cpCO2,cvCO2,cp123,a1H2Op,a2H2Op;
    float a3H2Op,a4H2Op,cpH2O,cvH2O,cp223,a1N2p,a2N2p,a3N2p,a4N2p;
    float cp323,a1O2p,a2O2p,a3O2p,a4O2p,cp423,a1COp,a2COp,a3COp;
    float
a4COp,cpCO,cvCO,cp523,cp23_kmol,cv23_kmol,Mm23,n23,cp23,cv23;
    float
delh1o23,delh2o23,delh3o23,delh5o23,hfo1,hfo2,hfo3,hfo4,hfo5,U123;
    float
U223,U323,U423,U523,U23,Up,Qin23,mF23,nF23,delUpr,error23,T3;
    float p23,delV23,delW23,W23,V3,p3,m3,theta3,To,Qin1,QW23;
// process 3-4 (polytropic expansion):
    float
V34i,p34i,T34i,m34i,xp34,Vs34,V34,V3V34,n34,n34T,R34,Z34,p34,T34;
    float delm34,m34,V4,p4,T4,theta4;
// process 4-5 (polytropic exhaust blowdown):
    float
V45i,p45i,T45i,m45i,xp45,Vs45,V45,V4V45,n45,n45T,R45,Z45,p45,T45;
    float delm45,m45,V5,p5,T5,theta5;
    // process 5-6 (polytropic exhaust):
    float xp56,Vs56,V56,V5V6,m56,V6,theta6,p6,T6;
// engine parameters
    float
Wnet,ETA_ith,ETA_ithper,imep_bar,W12,W2,W3,Q3,W4,W34,W45,W5;
    float
Wi,n_cps,Pi,Pb_kW,mF_kgps,mF_kgph,bsfc,LAMBDAMAX,LAMBDAMIN;
    float na,ng,nag,na1,n5r,na2;
 clrscr ();
 ofstream outfile ("EAEC-SI3.dat");
 cout << "\n\n\t\t  Adama Science and Technology University (ASTU)
";
 outfile << "\n\n\t\t  Adama Science and Technology University
(ASTU) ";
 cout << " \n\t          Department of Mechanical and Vehicle
Engineering ";
 outfile << " \n\t          Department of Mechanical and Vehicle
Engineering ";
 cout << " \n\t\t          Tamiru Tesfaye (Asst. lecturer) ";
 outfile << " \n\t\t          Tamiru Tesfaye (Asst. lecturer) ";
 cout <<
"\n.............................................................
............... \n";
 outfile <<
"\n.............................................................
..... \n";
 cout <<"\n Enter the 'Engine capacity (Ec_cc)' in 'cc' = ";
```

```
 cin >> Ec_cc;
 cout <<"\n Enter the 'Number of cyliders (Zc)' = ";
 cin >> Zc;
 cout <<"\n Enter the 'Ratio of stroke-to-bore (Rsd)' = ";
 cin >> Rsd;
 cout <<"\n Enter the 'Ratio of crank-to-con rod legnth (Rrl)' = ";
 cin >> Rrl;
 cout <<"\n Enter the 'number of Carbon atoms in Gasoline (a)' = ";
 cin >> a;
 cout <<"\n Enter the 'number of Hydrogen atoms in Gasoline (b)' =
";
 cin >> b;
 cout <<"\n Enter the 'density of Gasoline (RHO_G)' in 'kg/liter' =
";
 cin >> RHO_kgplit;
 cout <<"\n Enter the 'Lower Heating Value of Gasoline (QLHV_G)' in
'MJ/kg' = ";
 cin >> QLHV_G;
 cout <<"\n Enter the 'number of Carbon atoms in Alcohol (ZC)' = ";
 cin >> ZC;
 cout <<"\n Enter the 'number of Hydrogen atoms in Alcohol (ZH)' =
";
 cin >> ZH;
 cout <<"\n Enter the 'number of Oxygen atoms in Alcohol (ZO)' = ";
 cin >> ZO;
 cout <<"\n Enter the 'density of Alcohol (RHO_Al)' in 'kg/liter' =
";
 cin >> RHO_Alkgplit;
 cout <<"\n Enter the 'Lower Heating Value of Alcohol (QLHV_AL)' in
'MJ/kg' = ";
 cin >> QLHV_Al;
 cout <<"\n Enter the 'percentage of Alcohol by mass' = ";
 cin >> xAl_per;
 cout <<"\n Enter the 'residual mass fraction (xres) in '%' = ";
 cin >> xR_per;
 cout <<"\n Enter the 'engine Load (pim)' in 'bar' = ";
 cin >> pim;
 cout <<"\n Enter the 'pressure (p1)' in 'bar' = ";
 cin >> p1;
 cout <<"\n Enter the 'Temperature (T1)' in 'K' = ";
 cin >> T1;
 cout <<"\n Enter the 'compression Ratio (Rc)' = ";
 cin >> Rc;
 cout <<"\n Enter the 'required engine speed (N)' in 'rpm' = ";
 cin >> N;
 cout <<"\n Enter the 'mechanical Efficiency (ETA_m)' in '%' = ";
 cin >> ETA_mper;
 cout <<"\n Enter 'theta_ivobTDC' in 'deg.' = ";
 cin >> theta_ivobTDC;
 cout <<"\n Enter 'theta_ivcaBDC' in 'deg.' = ";
 cin >> theta_ivcaBDC;
 cout <<"\n Enter 'theta_scbTDC' in 'deg.' = ";
```

```
 cin >> theta_scbTDC;
 cout <<"\n Enter 'theta_dc' in 'deg.' = ";
 cin >> theta_dc;
 cout <<"\n Enter 'theta_evobBDC' in 'deg.' = ";
 cin >> theta_evobBDC;
 cout <<"\n Enter 'theta_evcaTDC' in 'deg.' = ";
 cin >> theta_evcaTDC;
   Zs = 4.0; Z_rpc = Zs/2.0; Tim = T1-10.0; To = 293.0;
   ETA_m = ETA_mper/100.0; xR = xR_per/100.0;
   theta_a = 0.0; theta_0 = theta_evcaTDC; theta_b = 180.0;
   theta_ivc = 180.0+theta_ivcaBDC; theta_1 = theta_ivc;
   theta_sc = 360.0-theta_scbTDC; theta_2 = theta_sc; theta_c =
360.0;
   theta_ec = (theta_dc-theta_scbTDC)+360.0; theta_3 = theta_ec;
   theta_evo = 540.0-theta_evobBDC; theta_4 = theta_evo; theta_5 =
540.0;
   theta_6 = 720.0-theta_ivobTDC;
   Ec_liter = Ec_cc*1.0e-3; Ec = Ec_liter*1.0e-3;
   Vs = Ec/Zc; Vcl = Vs/(Rc-1.0); Vt = Vcl+Vs;
   Vs_cc = Vs*1.0e6; Vcl_cc = Vcl*1.0e6; Vt_cc = Vt*1.0e6;
   powd = (4.0*Vs)/(PI*Rsd); d = pow(powd,0.33); d_mm = d*1.0e3; s =
d*Rsd;
   s_mm = s*1.0e3; r = s/2.0; r_mm = r*1.0e-3; lcon = r/Rrl;
   lcon_mm = lcon*1.0e3; Ap = (PI*d*d)/4.0;
 cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
 getch();
 getch();
 clrscr ();
 cout << "\n\n\t\t  Adama Science and Technology University (ASTU)
";
 outfile << "\n\n\t\t  Adama Science and Technology University
(ASTU) ";
 cout << " \n\t\t         Tamiru Tesfaye (Asst. lecturer) ";
 outfile << " \n\t\t         Tamiru Tesfaye (Asst. lecturer) ";
 cout <<
"\n.............................................................
............... \n";
 outfile <<
"\n.............................................................
..... \n";
 cout << "\n  Engine Size ";
 outfile << "\n  Engine Size ";
 cout << "\n ------------- ";
 outfile << "\n ------------- ";
   cout <<setprecision(3)<<"\n  Ec = "<<Ec_cc<<" cc ";
   outfile<<setprecision(3)<<"\n  Ec = "<<Ec_cc<<" cc ";
   cout <<setprecision(2)<<"\n  Vs = "<<Vs_cc<<" cc ";
   outfile<<setprecision(2)<<"\n  Vs = "<<Vs_cc<<" cc ";
   cout <<setprecision(2)<<"\n  Vcl = "<<Vcl_cc<<" cc ";
   outfile<<setprecision(2)<<"\n  Vcl = "<<Vcl_cc<<" cc ";
   cout <<setprecision(2)<<"\n  Vt = "<<Vt_cc<<" cc ";
   outfile<<setprecision(2)<<"\n  Vt = "<<Vt_cc<<" cc ";
```

```
 cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
 getch();
 getch();

       MmO = 16.0; MmN = 14.0; MmC = 12.0; MmH = 1.0;
       yO2 = 0.21; yN2 = 0.79; MmO2 = 2.0*MmO; MmN2 = 2.0*MmN;
       xAl = xAl_per/100.0; xG = 1.0-xAl; xF = xAl+xG;
       MmG = (a*MmC)+(b*MmH); MmAl = (ZC*MmC)+(ZH*MmH)+(ZO*MmO);
       nAl = xAl/MmAl; nG = xG/MmG; nAlG = nAl+nG;
       yAl = nAl/nAlG; yG = 1.0-yAl; yF = yAl+yG; yAl_per =
yAl*100.0;
       MmA = (yO2*MmO2)+(yN2*MmN2); MmF = (yG*MmG)+(yAl*MmAl);
       n1th = (ZC*yAl)+(a*yG); n2th = ((ZH*yAl)+(b*yG))/2.0;
       LAMBDAMAX = 1.2; LAMBDAMIN = 0.8;
       LAMBDA = (LAMBDAMAX*xAl)+(LAMBDAMIN*xG);
       atha = ((2.0*n1th)+n2th-(ZO*yAl))/2; ath = atha*LAMBDA;
       getch();
       clrscr();
 cout<<"\n\t The input amount in the combustion reaction";
 outfile<<"\n\t The input amount in the combustion reaction";
 cout<<"\n\t ***********************************************";
 outfile<<"\n\t ***********************************************";
       cout<<setprecision(2)<<"\n atha ="<<atha;
       outfile<<setprecision(2)<<"\n atha ="<<atha;
       cout<<setprecision(2)<<"\n LAMBDA ="<<LAMBDA;
       outfile<<setprecision(2)<<"\n LAMBDA ="<<LAMBDA;
       cout<<setprecision(2)<<"\n ath ="<<ath;
       outfile<<setprecision(2)<<"\n ath ="<<ath;
       cout<<setprecision(2)<<"\n xAl="<<xAl;
       outfile<<setprecision(2)<<"\n xAl="<<xAl;
       cout<<setprecision(2)<<"\n xG ="<<xG;
       outfile<<setprecision(2)<<"\n xG ="<<xG;
       cout<<setprecision(2)<<"\n yAl ="<<yAl;
       outfile<<setprecision(2)<<"\n yAl ="<<yAl;
       cout<<setprecision(2)<<"\n yG ="<<yG;
       outfile<<setprecision(2)<<"\n yG ="<<yG;
       getch();
       clrscr();
 cout<<"\n\t The emission amount in the combustion reaction";
 outfile<<"\n\t The emission amount in the combustion reaction";
 cout<<"\n\t ***********************************************";
 outfile<<"\n\t ***********************************************";
  if (LAMBDA == 1.0)
   {
// Considering 'one Kilo mole (nF = 1 kmol)' of HydroCarbon Fuel
(CaHb)',
// the 'theoretical combustion equation (LAMBDA = 1)'is:
//   nF*Ca+Hb+LAMBDA*ath*(O2+3.76*N2) - n1*CO2+n2*H2O+n3*N2
      yO2 = 0.21; yN2 = 0.79; MmO2 = 2.0*MmO; MmN2 = 2.0*MmN;
       xAl = xAl_per/100.0; xG = 1.0-xAl; xF = xAl+xG;
       MmG = (a*MmC)+(b*MmH); MmAl = (ZC*MmC)+(ZH*MmH)+(ZO*MmO);
       nAl = xAl/MmAl; nG = xG/MmG; nAlG = nAl+nG;
```

```
       yAl = nAl/nAlG; yG = 1.0-yAl; yF = yAl+yG; yAl_per = 
yAl*100.0;
       MmA = (yO2*MmO2)+(yN2*MmN2); MmF = (yG*MmG)+(yAl*MmAl);
       n1th = (ZC*yAl)+(a*yG); n2th = ((ZH*yAl)+(b*yG))/2.0;
      atha = (2.0*n1th)+n2th-(ZO*yAl); ath = atha*LAMBDA;
      n3 = (2.0*3.76*LAMBDA*atha)/2.0;
      n1 = n1th; n2 = n2th; n3th = n3th; n4 = 0.0; n5 = 0.0;
      nt = n1+n2+n3+n4+n5;
      y1 = n1/nt; y2 = n2/nt; y3 = n3/nt; y4 = n4/nt; y5 = n5/nt;
   cout<<setprecision(2)<<"\n\n  n1= "<<n1;
   outfile<<setprecision(2)<<"\n\n  n1= "<<n1;
   cout <<setprecision(2)<<"\n\n  n2= "<<n2;
   outfile<<setprecision(2)<<"\n\n  n2= "<<n2;
   cout<<setprecision(2)<<"\n\n  n3= "<<n3;
   outfile<<setprecision(2)<<"\n\n  n3= "<<n3;
   cout<<setprecision(2)<<"\n\n  n4= "<<n4;
   outfile<<setprecision(2)<<"\n\n  n4= "<<n4;
   cout<<setprecision(2)<<"\n\n  n5= "<<n5;
   outfile<<setprecision(2)<<"\n\n  n5= "<<n5;
   cout<<setprecision(2)<<"\n\n  y1= "<<y1;
   outfile<<setprecision(2)<<"\n\n y1= "<<y1;
   cout<<setprecision(2)<<"\n\n y2= "<<y2;
   outfile<<setprecision(2)<<"\n\n y2= "<<y2;
   cout<<setprecision(2)<<"\n\n y3= "<<y3;
   outfile<<setprecision(2)<<"\n\n y3= "<<y3;
   cout<<setprecision(2)<<"\n\n  y5 "<<y5;
   outfile<<setprecision(2)<<"\n\n y5= "<<y5;
      }
 if (LAMBDA > 1.0)
   {
// Considering 'one Kilo mole (nF = 1 kmol)' of HydroCarbon Fuel 
(CaHb)',
// the 'theoretical combustion equation (LAMBDA < 1)'is:
//   nF*Ca+Hb+LAMBDA*ath*(O2+3.76*N2) - n1*CO2+n2*H2O+n3*N2+n4*O2
       yO2 = 0.21; yN2 = 0.79; MmO2 = 2.0*MmO; MmN2 = 2.0*MmN;
       xAl = xAl_per/100.0; xG = 1.0-xAl; xF = xAl+xG;
       MmG = (a*MmC)+(b*MmH); MmAl = (ZC*MmC)+(ZH*MmH)+(ZO*MmO);
       nAl = xAl/MmAl; nG = xG/MmG; nAlG = nAl+nG;
       yAl = nAl/nAlG; yG = 1.0-yAl; yF = yAl+yG; yAl_per = 
yAl*100.0;
       MmA = (yO2*MmO2)+(yN2*MmN2); MmF = (yG*MmG)+(yAl*MmAl);
       n2 = ((ZH*yAl)+(b*yG))/2.0; n3 = (2.0*3.76*LAMBDA*atha)/2.0;
       n1r = (yAl+(2.0*ath)); n4 =((yAl+(2.0*ath))-(2*n1)-n2)/2;
       n5 = ((ZC*yAl)+(a*yG))-n1r; nt = n1+n2+n3+n4+n5r;
       n1= ((ZC*yAl)+(a*yG));
       na1 = ZC;na2=(yAl+(2.0*ath))-((ZH*yAl)+(b*yG))/2-
2*((ZC*yAl)+(a*yG));
       na= ZC*yAl;ng=a*yG;nag =na+ng;
       n5r= 0;
       y1 = n1/nt; y2 = n2/nt; y3 = n3/nt; y4 = n4/nt; y5 = n5r/nt;
   cout<<setprecision(2)<<"\n\n  n1= "<<n1;
   outfile<<setprecision(2)<<"\n\n  n1= "<<n1;
```

```
   cout <<setprecision(2)<<"\n\n  na2= "<<na2;
   cout<<setprecision(2)<<"\n\n  n1r= "<<n1r;
   cout <<setprecision(2)<<"\n\n  n2= "<<n2;
   outfile<<setprecision(2)<<"\n\n  n2= "<<n2;
   cout<<setprecision(2)<<"\n\n  n3= "<<n3;
   outfile<<setprecision(2)<<"\n\n  n3= "<<n3;
   cout<<setprecision(2)<<"\n\n  n4= "<<n4;
   outfile<<setprecision(2)<<"\n\n  n4= "<<n4;
   cout<<setprecision(2)<<"\n\n  n5= "<<n5r;
   outfile<<setprecision(2)<<"\n\n  n5= "<<n5r;
   cout<<setprecision(2)<<"\n\n  y1= "<<y1;
   outfile<<setprecision(2)<<"\n\n y1= "<<y1;
   cout<<setprecision(2)<<"\n\n y2= "<<y2;
   outfile<<setprecision(2)<<"\n\n y2= "<<y2;
   cout<<setprecision(2)<<"\n\n y3= "<<y3;
   outfile<<setprecision(2)<<"\n\n y3= "<<y3;
   cout<<setprecision(2)<<"\n\n y4= "<<y4;
   outfile<<setprecision(2)<<"\n\n y4= "<<y4;
   cout<<setprecision(2)<<"\n\n  y5 "<<y5;
   outfile<<setprecision(2)<<"\n\n y5= "<<y5;
   }
  if (LAMBDA < 1.0)
   {
// Considering 'one Kilo mole (nF = 1 kmol)' of HydroCarbon Fuel
(CaHb)',
// the 'theoretical combustion equation (LAMBDA < 1)'is:
//   nF*Ca+Hb+LAMBDA*ath*(O2+3.76*N2) -
n1*CO2+n2*H2O+n3*N2+n4*O2+n5*CO
       yO2 = 0.21; yN2 = 0.79; MmO2 = 2.0*MmO; MmN2 = 2.0*MmN;
       xAl = xAl_per/100.0; xG = 1.0-xAl; xF = xAl+xG;
       MmG = (a*MmC)+(b*MmH); MmAl = (ZC*MmC)+(ZH*MmH)+(ZO*MmO);
       nAl = xAl/MmAl; nG = xG/MmG; nAlG = nAl+nG;
       yAl = nAl/nAlG; yG = 1.0-yAl; yF = yAl+yG; yAl_per =
yAl*100.0;
       MmA = (yO2*MmO2)+(yN2*MmN2); MmF = (yG*MmG)+(yAl*MmAl);
       n2 = ((ZH*yAl)+(b*yG))/2.0; n3 = (2.0*3.76*LAMBDA*atha)/2.0;
       n1r = (yAl+(2.0*ath)); n4 = 0.0;
       n5 = ((ZC*yAl)+(a*yG))-n1r; nt = n1+n2+n3+n4+n5r;
       n1= ((ZC*yAl)+(a*yG));
       na1 = ZC;na2=(yAl+(2.0*ath))-((ZH*yAl)+(b*yG))/2-
((ZC*yAl)+(a*yG));
       na= ZC*yAl;ng=a*yG;nag =na+ng;
       n5r= n1-na2;
       y1 = n1/nt; y2 = n2/nt; y3 = n3/nt; y4 = n4/nt; y5 = n5r/nt;
   cout<<setprecision(2)<<"\n\n  n1= "<<n1;
   outfile<<setprecision(2)<<"\n\n  n1= "<<n1;
   cout <<setprecision(2)<<"\n\n  na2= "<<na2;
   cout<<setprecision(2)<<"\n\n  n1r= "<<n1r;
   cout <<setprecision(2)<<"\n\n  n2= "<<n2;
   outfile<<setprecision(2)<<"\n\n  n2= "<<n2;
   cout<<setprecision(2)<<"\n\n  n3= "<<n3;
   outfile<<setprecision(2)<<"\n\n  n3= "<<n3;
```

```
   cout<<setprecision(2)<<"\n\n  n4= "<<n4;
   outfile<<setprecision(2)<<"\n\n  n4= "<<n4;
   cout<<setprecision(2)<<"\n\n  n5= "<<n5r;
   outfile<<setprecision(2)<<"\n\n  n5= "<<n5r;
   cout<<setprecision(2)<<"\n\n  y1= "<<y1;
   outfile<<setprecision(2)<<"\n\n y1= "<<y1;
   cout<<setprecision(2)<<"\n\n y2= "<<y2;
   outfile<<setprecision(2)<<"\n\n y2= "<<y2;
   cout<<setprecision(2)<<"\n\n y3= "<<y3;
   outfile<<setprecision(2)<<"\n\n y3= "<<y3;
   cout<<setprecision(2)<<"\n\n  y5 "<<y5;
   outfile<<setprecision(2)<<"\n\n y5= "<<y5;
   getch();
 clrscr();
}
       MmAth = MmA; MmFth = MmF; nO2 = atha; nN2 = atha*3.76; nA =
nO2+nN2;
       AFth = (nA*MmAth)/MmFth; AF = LAMBDA*AFth;
       QLHV_F = (QLHV_Al*xAl)+(QLHV_G*xG);
       RHO_Fkgplit = (RHO_Alkgplit*yAl)+(RHO_Gkgplit*yG);
       RHO_F = RHO_Fkgplit*1.0e3; ETA_cmax = 0.95;
       ETA_c = ETA_cmax*(-1.6+(4.65*LAMBDA)-(2.07*LAMBDA*LAMBDA));
       ETA_cper = ETA_c*100.0; Rm = 0.287; Zm = 0.99; Vm = Vt;
       mm = (pim*1.0e5*Vm)/(Zm*Rm*1.0e3*Tim);
       mA = (AF/(AF+1.0))*(1.0-xR)*mm;
       mF = (1.0/(AF+1.0))*(1.0-xR)*mm;
       mres = xR*mm;
   cout <<setprecision(2)<<"\n\n  ath = "<<ath<<"  ;   AF = "<<AF;
   outfile<<setprecision(2)<<"\n\n  ath = "<<ath<<"  ;   AF = "<<AF;
   cout <<setprecision(2)<<" ;  ETA_c = "<<ETA_cper<<" % ";
   outfile<<setprecision(2)<<" ;  ETA_c = "<<ETA_cper<<" % ";
   cout <<setprecision(2)<<" ;  yAl_per = "<<yAl_per<<" % ";
   outfile<<setprecision(2)<<" ;  yAl_per = "<<yAl_per<<" % ";
   cout <<setprecision(6)<<"\n\n  mm = "<<mm<<" kg ";
   outfile<<setprecision(6)<<"\n\n  mm = "<<mm<<" kg ";
   cout <<setprecision(6)<<"\n  mA = "<<mA<<" kg ";
   outfile<<setprecision(6)<<"\n  mA = "<<mA<<" kg ";
   cout <<setprecision(6)<<"\n  mF = "<<mF<<" kg ";
   outfile<<setprecision(6)<<"\n  mF = "<<mF<<" kg ";
   cout <<setprecision(6)<<"\n  mres = "<<mres<<" kg ";
   outfile<<setprecision(6)<<"\n  mres = "<<mres<<" kg ";
   cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
   getch();
   getch();
 clrscr();
 cout << "\n\n\t\t  Adama Science and Technology University (ASTU)
";
 cout << " \n\t         Department of Mechanical and Vehicle
Engineering ";
 cout << " \n\t\t         Tamiru Tesfaye (Asst. lecturer) ";
```

```
 cout <<
"\n...............................................................
.............. \n";
 cout << "\n\n   'V', 'p' and 'T' as a function of 'theta' ";
 outfile << "\n\n  'V', 'p' and 'T' as a function of 'theta' ";
 cout << "\n ------------------------------------------- \n";
 outfile << "\n\t----------------------------------------------------
-- ";
 cout <<"\n\ttheta\t  V\t  p\t  T ";
 cout <<"\n\t(deg.)\t (cc)\t (bar)\t (K) \n";
 outfile <<"\n\t   theta\t   V\t   p\t  T ";
 outfile <<"\n\t   (deg.)\t  (cc)\t  (bar)\t  (K) \n ";
      deltheta = 10.0;
   for (theta = 0.0; theta <= 720.0; theta = theta+deltheta)
    {
      V0bi = Vcl; p0bi = pim; T0bi = Tim; m0bi = mres; R0bi = R6b;
     if (theta >= theta_0 && theta <= theta_b)
      {
// process a-7 (constant pressure intake process).
        rad = theta*(PI/180.0); Z0b = 0.99;
        xp0b = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
        Vs0b = Ap*xp0b; V0b = Vcl+Vs0b; p0b = p0bi; T0b = T0bi; R0b =
0.287;
        m0b = m0bi+((pim*1.0e5*V0b)/(Z0b*R0b*1.0e3*T0b));
        Vb = V0b*1.0e6; pb = p0b; Tb = T0b; mb = m0b; thetab = theta;
          cout << setprecision(2)<<"\n\t"<<thetab;
          outfile << setprecision(2)<<"\n  " <<thetab;
          cout << setprecision(2)<<"\t "<<Vb;
          outfile << setprecision(2)<<"\t"<< Vb;
          cout <<setprecision(2)<<"\t "<<pb;
          outfile <<setprecision(2)<<"\t " << pb;
          cout <<setprecision(2)<<"\t"<<Tb;
          outfile <<setprecision(2)<<"\t " << Tb;
          getch();
           }
      Vb1i = Vb*1.0e-6; pb1i = pb; Tb1i = Tb; mb1i = mb;
     if (theta > theta_b && theta <= theta_1)
      {
// process b-1 (polytropic compression):
        rad = theta*(PI/180.0);
        xpb1 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
        Vsb1 = Ap*xpb1; Vb1 = Vcl+Vsb1; VbVb1 = Vb1i/Vb1;
        nb1 = 1.35; nb1T = nb1-1.0; Rb1 = 0.287; Zb1 = 0.99;
        pb1 = pb1i*pow(VbVb1,nb1); Tb1 = Tb1i*pow(VbVb1,nb1T);
        delmb1 = mb1i-((pb1*1.0e5*Vb1)/(Zb1*Tb1*Rb1*1.0e3));
        mb1 = mb1i-delmb1;
        V1 = Vb1*1.0e6; p1 = pb1; T1 = Tb1; theta1 = theta;
          cout << setprecision(2)<<"\n\t"<<theta1;
          outfile << setprecision(2)<<"\n  " <<theta1;
          cout << setprecision(2)<<"\t "<<V1;
          outfile << setprecision(2)<<"\t"<< V1;
          cout <<setprecision(2)<<"\t "<<p1;
```

```
          outfile <<setprecision(2)<<"\t " << p1;
          cout <<setprecision(2)<<"\t"<<T1;
          outfile <<setprecision(2)<<"\t " << T1;
          getch();
           }
        m12i = mb1;
        V12i = V1*1.0e-6; p12i = p1; T12i = T1; m12i = mb1;
     if (theta > theta_1 && theta <= theta_2)
      {
// process 1-2 (polytropic effective compression):
        rad = theta*(PI/180.0);
        xp12 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
        Vs12 = Ap*xp12; V12 = Vcl+Vs12; V1V12 = V12i/V12;
        n12 = 1.35; n12T = n12-1.0; R12 = 0.287; Z12 = 0.99; m12 =
m12i;
        p12 = p12i*pow(V1V12,n12); T12 = T12i*pow(V1V12,n12T);
        W12 = (m12*R12*(T12-T12i))/(1.0-n12); W2 = W12;
        V2 = V12*1.0e6; p2 = p12; T2 = T12; theta2 = theta;
          cout << setprecision(2)<<"\n\t"<<theta2;
          outfile << setprecision(2)<<"\n  " <<theta2;
          cout << setprecision(2)<<"\t "<<V2;
          outfile << setprecision(2)<<"\t"<< V2;
          cout <<setprecision(2)<<"\t "<<p2;
          outfile <<setprecision(2)<<"\t " << p2;
          cout <<setprecision(2)<<"\t"<<T2;
          outfile <<setprecision(2)<<"\t " << T2;
          cout<<setprecision(2)<<"\t " << W12;
          getch();
           }
          W12 = W2;
          V23i = V2*1.0e-6; p23i = p2; T23i = T2; T23 = T2+5.0;
     if (theta > theta_2 && theta <= theta_3)
      {
        rad = theta*(PI/180.0); p23 = p3;
        xp23 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
        Vs23 = Ap*xp23; V23 = Vcl+Vs23; m23 = m12i;
        theta2dc = ((theta-theta_2)/theta_dc)*PI; rad2dc = theta2dc;
        cv23 = 0.821; R23 = 0.287; n23 = 1.3; // Z23 = 0.98;
        xF = 1.0/(AF+1.0); mF = (xF*(1.0-xR))*m23; xb = 0.5*(1.0-
cos(rad2dc));
        Qin = (mF*QLHV_F*1.0e6)*ETA_c; Q1 = Qin*xb; Q23 = Q1;
        W23 = (p23*1.0e5)*(V23-V23i);
        QW23 = (Q23-W23)/(m23*cv23*1.0e3);
        T23 = QW23+T23i; p23 = ((T23/T23i)*(V23i/V23))*p23i;
        W3 = W23/1000.0; Q3 = Q23/1000.0;
        V3 = V23*1.0e6; p3 = p23; T3 = T23; T23 = T3;
        m3 = m23; theta3 = theta;
          cout << setprecision(2)<<"\n\t"<<theta3;
          outfile << setprecision(2)<<"\n  " <<theta3;
          cout << setprecision(2)<<"\t "<<V3;
          outfile << setprecision(2)<<"\t"<< V3;
          cout <<setprecision(2)<<"\t "<<p3;
```

```
        outfile <<setprecision(2)<<"\t " << p3;
        cout <<setprecision(2)<<"\t"<<T3;
        outfile <<setprecision(2)<<"\t " << T3;
        getch();
         }
    W23 = W3; Q23 = Q3;
    V34i = V3*1.0e-6; p34i = p3; T34i = T3; m34i = m3;
   if (theta > theta_3 && theta <= theta_4)
    {
// process 3-4 (polytropic expansion):
       rad = theta*(PI/180.0);
       xp34 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
       Vs34 = Ap*xp34; V34 = Vcl+Vs34; V3V34 = V34i/V34; m34 = m12i;
       n34 = 1.35; n34T = n34-1.0; R34 = 0.287; Z34 = 0.99;
       p34 = p34i*pow(V3V34,n34); T34 = T34i*pow(V3V34,n34T);
       delm34 = m34i-((p34*1.0e5*V34)/(Z34*T34*R34*1.0e3));
       m34 = m34i-delm34; W34 = (m34*R34*(T34-T34i))/(1.0-n34);
       W4 = W34;
       V4 = V34*1.0e6; p4 = p34; T4 = T34; theta4 = theta;
        cout << setprecision(2)<<"\n\t"<<theta4;
        outfile << setprecision(2)<<"\n  " <<theta4;
        cout << setprecision(2)<<"\t "<<V4;
        outfile << setprecision(2)<<"\t"<< V4;
        cout <<setprecision(2)<<"\t "<<p4;
        outfile <<setprecision(2)<<"\t " << p4;
        cout <<setprecision(2)<<"\t"<<T4;
        outfile <<setprecision(2)<<"\t " << T4;
        cout<<setprecision(2)<<"\t " << W34;
        getch();
         }
    W34 = W4;
    V45i = V4*1.0e-6; p45i = p4; T45i = T4; m45i = m12i;
   if (theta > theta_4 && theta <= theta_5)
    {
// process 4-5 (polytropic exhaust blowdown):
       rad = theta*(PI/180.0);
       xp45 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
       Vs45 = Ap*xp45; V45 = Vcl+Vs45; V4V45 = V45i/V45; m45 = m12i;
       n45 = 1.35; n45T = n45-1.0; R45 = 0.287; Z45 = 0.99;
       p45 = p45i*pow(V4V45,n45); T45 = T45i*pow(V4V45,n45T);
       delm45 = m45i-((p45*1.0e5*V45)/(Z45*T45*R45*1.0e3));
       m45 = m45i-delm45; W45 = (m45*R45*1.0e3*(T45-T45i))/(1.0-
n45);
       W5 = W45/100.0;
       V5 = V45*1.0e6; p5 = p45; T5 = T45; theta5 = theta;
        cout << setprecision(2)<<"\n\t"<<theta5;
        outfile << setprecision(2)<<"\n  " <<theta5;
        cout << setprecision(2)<<"\t "<<V5;
        outfile << setprecision(2)<<"\t"<< V5;
        cout <<setprecision(2)<<"\t "<<p5;
        outfile <<setprecision(2)<<"\t " << p5;
        cout <<setprecision(2)<<"\t"<<T5;
```

```
        outfile <<setprecision(2)<<"\t " << T5;
        getch();
         }
        W45 = W5;
       V45i = V4*1.0e-6; p45i = p4; T45i = T4; m45i = m12i;
    if (theta > theta_5 && theta <= theta_6)
     {
// process 4-5 (polytropic exhaust blowdown):
       rad = theta*(PI/180.0);
       xp56 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
       Vs56 = Ap*xp56; V56 = Vcl+Vs56; V5V6 = V45i/V56; m56 = m12i;
       n45 = 1.35; n45T = n45-1.0; R45 = 0.287; Z45 = 0.99;
       p45 = p45i*pow(V4V45,n45); T45 = T45i*pow(V4V45,n45T);
       delm45 = m45i-((p45*1.0e5*V45)/(Z45*T45*R45*1.0e3));
       m45 = m45i-delm45; W45 = (m45*R45*1.0e3*(T45-T45i))/(1.0-
n45);
       W5 = W45/100.0;
       V6 = V56*1.0e6; p6 = p1; T6 = T1; theta6 = theta;
        cout << setprecision(2)<<"\n\t"<<theta6;
        outfile << setprecision(2)<<"\n  " <<theta6;
        cout << setprecision(2)<<"\t "<<V6;
        outfile << setprecision(2)<<"\t"<< V6;
        cout <<setprecision(2)<<"\t "<<p6;
        outfile <<setprecision(2)<<"\t " << p6;
        cout <<setprecision(2)<<"\t"<<T6;
        outfile <<setprecision(2)<<"\t " << T6;
        getch();
         }
          }
        Wnet = W12+W34; Qin = Q23;
        ETA_ith = Wnet/Qin; ETA_ithper = ETA_ith*100.0;
        imep_bar = ((Wnet*1.0e3)/Vs)*1.0e-5;
        Wi = Wnet*Zc; n_cps = N/(60.0*Z_rpc); Pi = Wi*n_cps;
        Pb_kW = Pi*ETA_m; mF_kgps = mF*Zc*n_cps;
        mF_kgph = mF_kgps*3600.0; bsfc = mF_kgph/Pb_kW;
        Tb_NM = (Pb_kW*60*1000)/(2*3.14*N);
   cout <<setprecision(6)<<"\n\n  mm = "<<mm<<" kg ";
   outfile<<setprecision(6)<<"\n\n  mm = "<<mm<<" kg ";
   cout <<setprecision(6)<<"\n  mb1i = "<<mb1i<<" kg ";
   outfile<<setprecision(6)<<"\n  mb1i = "<<mb1i<<" kg ";
   cout <<setprecision(6)<<"\n  m12i = "<<m12i<<" kg ";
   outfile<<setprecision(6)<<"\n  m12i = "<<m12i<<" kg ";
   cout <<setprecision(2)<<"\n\n  Wnet = "<<Wnet<<" kJ ";
   outfile<<setprecision(2)<<"\n\n  Wnet = "<<Wnet<<" kJ ";
   cout <<setprecision(2)<<";  Qin = "<<Qin<<" kJ ";
   outfile<<setprecision(2)<<";  Qin = "<<Qin<<" kJ ";
   cout <<setprecision(2)<<";  ETA_ith = "<<ETA_ithper<<" % ";
   outfile<<setprecision(2)<<";  ETA_ith = "<<ETA_ithper<<" % ";
   cout <<setprecision(2)<<";  imep = "<<imep_bar<<" bar ";
   outfile<<setprecision(2)<<";  imep = "<<imep_bar<<" bar ";
   cout <<setprecision(2)<<"\n\n  Pb = "<<Pb_kW<<" kW ";
   outfile<<setprecision(2)<<"\n\n  Pb = "<<Pb_kW<<" kW ";
```

```
 cout <<setprecision(2)<<"\n\n  Tb= "<<Tb_NM<<" NM ";
outfile<<setprecision(2)<<"\n\n  Tb = "<<Tb_NM<<" NM ";
cout <<setprecision(2)<<";  bsfc = "<<bsfc<<" kg/kWh ";
outfile<<setprecision(2)<<";  bsfc = "<<bsfc<<" kg/kWh ";
cout << "\n\n\n\t\t Press 'enter' twice to exit ! ";
getch();
getch();
getche();
```

Appendix. 2

For drawing engine performance parameter curve

```
#include<iostream.h>
#include<iomanip.h>
#include<conio.h>
#include<math.h>
#include<fstream.h>
 void main()
  {
  float Pb,Pb_max,x,N,Tb,NaPb_max,pi,bsfc,bsfcaPb_max,Ec,bmep;
  float OMEGA,n_cps,del;
 clrscr ();
 ofstream outfile ("per.dat");
 cout << "\n\n\t\t  Adama Science and Technology University (ASTU)
";
 outfile << "\n\n\t\t  Adama Science and Technology University
(ASTU) ";
 cout << " \n\t         Department of Mechanical and Vehicle
Engineering ";
 outfile << " \n\t         Department of Mechanical and Vehicle
Engineering ";
 cout << " \n\t\t         Tamiru Tesfaye (Asst. lecturer) ";
 outfile << " \n\t\t         Tamiru Tesfaye (Asst. lecturer) ";
 cout <<
"\n...................................................................
............... \n";
 outfile <<
"\n.................................................................";
```

```
  cout <<"\n Enter engine capacity(Ec) in cc = ";
  cin >>Ec;
  cout <<"\n Enter maximum brake power(Pb_max)in kw = ";
  cin >>Pb_max;
  cout << "\n Enter speed at maximum brake power(NaPb_max)in rpm =
";
  cin >>NaPb_max;
  cout << "\n Enter bsfc at maximum brake power(bsfcaPb_max)in
kg/kwh  = ";
  cin >>bsfcaPb_max;
  del = 1000;
  getch();
  getch();
  clrscr();
  for (N =0; N<= 8000; N = N+del)
  {
    pi=3.14;
    x = N/NaPb_max; n_cps = N/(60.0*2); OMEGA = (2.0*pi*N)/60.0;
     Pb = Pb_max*(-(x*x*x)+(x*x)+x); Tb = (Pb*1.0e3)/OMEGA;
    bsfc = bsfcaPb_max*((x*x)-(1.2*x)+1.2);
    bmep = ((Pb*1.0e3)/(Ec*n_cps))*1.0e6;
    getch();clrscr();
    cout << "\n\n\t\t  Adama Science and Technology University
(ASTU) ";
    outfile << "\n\n\t\t  Adama Science and Technology University
(ASTU) ";
    cout << " \n\t\t         Tamiru Tesfaye (Asst. lecturer) ";
    outfile << " \n\t\t       Tamiru Tesfaye(Asst. lecturer) ";
    cout <<
"\n....................................................................
............... \n";
    outfile <<
"\n....................................................................
..... \n";
    cout <<"\n\n  Major 'Performance Parameters':";
```

```
   cout <<"\n --------------------------------";
   cout <<"\n N"; outfile <<"\n N"; cout <<"\t Pb"; outfile <<"\t
Pb";
   cout <<"\t Tb"; outfile <<"\t Tb"; cout <<"\t bsfc"; outfile
<<"\t bsfc";
   cout <<"\t bmep"; outfile <<"\t bmep"; cout <<"\n (rpm)"; outfile
<<"\n (rpm)";
   cout <<"\t (kw)"; outfile <<"\t (kw)"; cout <<"\t (NM)"; outfile
<<"\t (NM)";
   cout <<"\t (kg/kwh)"; outfile <<"\t (kg/kwh)"; cout <<"\t\t
(bar)"; outfile <<"\t (bar)";
   cout <<setprecision(3)<<"\n  N = "<<N<<" rpm ";
   outfile <<setprecision(3)<<"\n  N = "<<N<<" rpm ";
   cout <<setprecision(2)<<"\t  Pb = "<<Pb<<" kW ";
   outfile <<setprecision(2)<<"\t  Pb = "<<Pb<<" kW ";
   cout <<setprecision(2)<<"\t  Tb = "<<Tb<<" Nm ";
   outfile  <<setprecision(2)<<"\t  Tb = "<<Tb<<" Nm ";
   cout <<setprecision(2)<<"\t  bsfc = "<<bsfc<<" kg/kWh ";
   outfile  <<setprecision(2)<<"\t  bsfc = "<<bsfc<<" kg/kWh ";
   cout <<setprecision(2)<<"\t  bmep = "<<bmep<<" bar ";
   outfile <<setprecision(2)<<"\t  bmep = "<<bmep<<" bar ";
   getch();
   getche();
   }
   cout <<"\n press any key twice to 'proceed'";
   getch();
   getch();
   getche();
  }
```