Simulation of diesel engine performances fueled with Diesel-Ethanol blends

Habtamu Belay Afework

A Thesis Submitted to

The Department of Mechanical Systems and Vehicle Engineering

School of Mechanical, Chemical and Materials Engineering

Presented in Partial Fulfillment of the Requirement for the Award of the Degree of Master of Science in Automotive Engineering

Office of Graduate Studies

Adama Science and Technology University

September, 2021GC

Adama, Ethiopia

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DECLARATION

I hereby declare that this Master Thesis entitled "Simulation of diesel engine performances fueled with Diesel-Ethanol blends" is my original work. That is, it has not been submitted for the award of any academic degree, diploma or certificate in any other university. All sources of materials that are used for this thesis have been duly acknowledged through citation.

________________	________________	________________
Name of the student	Signature	Date

RECOMMENDATION

I, the advisor of this thesis, hereby certify that I have read the revised version of the thesis entitled “Simulation of diesel engine performances fueled with Diesel-Ethanol blends” prepared under my guidance by Habtamu Belay Afework submitted in partial fulfillment of the requirements for the degree of Master of Science in Automotive Engineering.

Therefore, I recommend the submission of revised version of the thesis to the department following the applicable procedures.

________________	________________	________________
Advisor	Signature	Date

APPROVAL PAGE

I, the advisor of the thesis entitled "Simulation of diesel engine performances fueled with Diesel-Ethanol blends" and developed by Habtamu Belay, hereby certify that the recommendation and suggestions made by the board of examiners are appropriately incorporated into the final version of the thesis.

Advisor	Signature	Date

We, the undersigned, members of the Board of Examiners of the thesis by Habtamu Belay have read and evaluated the thesis entitled "Simulation of diesel engine performances fueled with Diesel-Ethanol blends" and examined the candidate during open defense. This is, therefore, to certify that the thesis is accepted for partial fulfillment of the requirement of the degree of Master of Science in Automotive Engineering.

Chairperson	Signature	Date
Internal Examiner	Signature	Date
External Examiner	Signature	Date
Department Head	Signature	Date
School Dean	Signature	Date
Office of Postgraduate Studies, Dean	Signature	Date

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List of symbols

a_s	Number of moles of air at stoichiometric condition [kmol]
a	Crank offset or crank radius [m]
AF	Air–fuel ratio,
AF_s	Air–fuel ratio for stoichiometric condition
A_h	Heat transfer area [m^2]
$bsfc$	specific effective fuel consumption [kg/kwh]
B	Bore [m]
C_v	constant volume specific heat [kJ/kg k]
C_p	Specific heat capacity [kJ/kg]
D	Cylinder diameter [m]
E_s	Engine capacity [cm^3]
h_{cv}	Heat transfer coefficient for gases in the cylinder [kJ/m^2K]]
l	Connecting rod length [m]
LHV	Lower heating value [MJ/m^2K]
m	Mass of cylinder contents [kg]
M	Molar mass [kJ/mol]
$P_{b\ max\ @_{N_{max}}}$	Maximum brake power at max speed [kW]
P	Pressure inside cylinder [bar]
Q	Heat transfer [kJ]
Q_{in}	Heat added from burning fuel [kJ]
r_C	Compression ratio
R	Universal gas constant [kJ/kg K]

S	Distance travelled by the piston from top dead center [m]
T_w	Cylinder wall temperature [K]
t	Time [s]
Tb	Brake torque [Nm]
V	Cylinder volume [m^3]
V_c	Clearance volume [m^3]
V_d	Displacement volume [m^3]
x_b	Mass fraction of fuel burnt at given crank angle [deg]
Z_c	Number of cylinders
x	Distance from top dead center [m]
θ	Angle [deg]
θ_{sc}	Duration of combustion [deg]
λ	Air fuel ratio
PM	Particulate matter
NOx	Nitrogen oxide
CO	Carbon mono oxide
HC	Hydro carbon
E5	5% ethanol 95 % diesel
E10	10% ethanol 90 % diesel
E15	15% ethanol 85 % diesel
ω	Angular velocity [rad/s]

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ABSTRACT

Number of researches has been done to study the effects of alcohol (ethanol) as a blend fuel for IC engines. In this particular work the effects of diesel-ethanol blends (E5, E10 and E15) on the performance; combustion and exhaust emission is studied in comparison to the pure diesel fuel (E0). A mathematical model is prepared from the thermodynamic analysis of actual diesel engine cycle and a computer program was developed to plot various important curves. The major performance parameters used were brake power, brake specific fuel consumption, brake thermal efficiency. The maximum in-cylinder gas pressure of E0 is 59.08 bar, and ethanol-blended E5, E10 and E15 are 3.12%, 2.5% and 2.3% higher, respectively, at 60.98 bar, 60.56 and 60.44, respectively and exhaust emission were analyzed. It is observed that the brake power of blend fuels tends to drop slightly as compared to the pure diesel fuel (E0). On the contrary, the brake thermal efficiency of blend fuels tends to decrease slightly as compared to the pure diesel fuel. The major pollutants or emissions: CO, HC, NO_x and PM (Particulate Matters) tend to drop with respect to pure diesel fuel. In this work, combustion model is developed to predict the performance of diesel engine. A computer code using "C++" language was developed for compression ignition (CI) engine. Combustion characteristics such as cylinder pressure, heat release, heat transfer and performance characteristics such as brake power, brake thermal efficiency and emission formation were analyzed.

***Keyword**: Alternative fuels, Diesel-ethanol blends, thermodynamic analysis, computer simulation*

CHAPTER ONE

INTRODUCTION

1.1 Background

Fossil fuel is a dominant player in the supply of energy as the global energy demand rises ex-potentially but the use of fossil fuels as the major source of energy for most countries has caused several negative impacts on their economy (Achmad et al. 2014). The recent sustained increases in world prices of petroleum products become a drag on the world economy, due to escalating price of petroleum products has resulted increase in inflation and reduce economic growth. In terms of inflation, oil prices directly affect the prices of goods made with petroleum products. As mentioned above, higher prices of petroleum products indirectly affect costs such as transportation and manufacturing sector. The increase in these costs can in turn affect the prices of a variety of goods and services in one country. Also, with the continuously increasing energy demand for the transport and other sectors which uses fossil fuel as the main energy sources the world environment enormously affected due to exhaust emissions of greenhouse gasses. The major contributor to air pollution and global warming is considered to be the high levels of greenhouse gas emissions, especially carbon dioxide (CO_2) caused by the burning of fossil fuel. Air pollution causes of many health problems and bring on social and economic negative effects. As concern about global warming and dependence on petroleum products grows, the search for renewable energy sources that reduce effects of global warming becomes a matter of widespread attention. In addition, as the depletion of these petroleum based non-renewable resources is underway around the world the investigations for renewable and sustainable alternative source of energy which can provide the necessary energy demand of the world has gained attention all over the world. Blending a fossil fuel with renewable alternative fuel has advantages of reducing exhaust emissions greenhouse gasses such that CO, NOx and increase the engine break power (Achmad et al. 2014). Air pollution is a major problem to human health and environment. Undoubtedly, the global essential energy production and air pollution are the mostly issues in 21st century. Diesel engines are widely used in the world transportation system (Simeon et al. 2019). Diesel fuel is comprised of a mixture of crude oil components known as hydrocarbons. It is a fossil base that provide mainly of aliphatic hydrocarbons from 8 to 28 with the various boiling points, from 130 to 375℃. Diesel engine develops high power, but cause to increase of the air pollution. Alcohols has a good potential to use as oxygenated fuel additives with diesel fuel in the diesel engine to decrease exhaust emissions, controlling fuel properties and increase the engine performance (Vahid et al. 2016).

This thesis deals with studying of the effect of using ethanol-diesel (E5, E10 and E15) fuel in compression ignition (CI) engine through computer programing based on the thermodynamic analysis of actual diesel engine.

1.2 Statement of the problem

The rising price of petroleum-based fuels coupled with increasing threat to the environment due to exhaust emissions and global warming heightened interest in the adoption and use of alternative fuel. The use of ethanol blended with diesel is receiving more attention by many researchers in recent times in an effort to reduce dependence on fossil fuels with a sustainable and environmentally sound improvement. Ethiopia is one of the developing countries in which energy is one of the most important factors for economic and social development. As a net oil importing country, Ethiopia is highly vulnerable to the supply and price escalation of petroleum in the world market. Therefore, petroleum energy security always has been highest national concern in the country. According to the Ministry of Water and Energy, the Ethiopian governments formulate a policy in 2009. According to the policy, blending of locally produced ethanol 10% and 90% of benzene for the market to overcome problems related to shortage of supply fuel but it is now stopped due to unknown reason. In addition, government policies not support Ethanol-Diesel blending in Ethiopia still now. Researches are conducting worldwide on this area, but there is no common conclusion and standard of using ethanol percentages for diesel fuel still now. The statement of the problem in the thesis is to investigate the characteristics of a compression ignition engine fueled with ethanol-diesel blends using a computer C++ language. The study aims to identify the optimum blend ratios that offer engine characteristics close to baseline diesel fuel. The research also aims to promote the use of renewable energy resources and reduce air pollution. The study is significant as it can be used as input for researchers, makes ethanol production companies profitable, increases the use of renewable energy as a source of automotive fuel, benefits the country economically due to the reduction of imported diesel fuel, and promotes renewable energy resources and reduction of air pollution.

1.3 Objectives

1.3.1 General objective

The general objective of this thesis is to study the characteristics of compression ignition fueled with ethanol-diesel blends using a computer C++ language.

1.3.2 Specific objectives

The specific objectives are:

- To develop CI engine parameters mathematical model.
- To compute the characteristics of CI engine running with ethanol-diesel blends.
- To identify the optimum blend ratios this offered engine characteristics close to baseline diesel fuel.

1.4 Significance of the thesis

The significance's of the thesis work are the following

- The results of this work can be used as input for researchers.
- Makes ethanol production companies profitable.
- Increase the use of renewable energy as a source of automotive fuel.
- Country gets benefitted economically due to reduction of imported diesel fuel.
- Promote renewable energy resources and reduction of air pollution.

1.5 Beneficiaries

The beneficiaries of the thesis are ethanol production companies, vehicle owners, researchers, society, and the country as a whole. Ethanol production companies can benefit from the study as it can make their production profitable. Vehicle owners can benefit from the study as it promotes the use of renewable energy resources and reduces air pollution. Researchers can benefit from the study as it can be used as input for further research. Society and the country as a whole can benefit from the study as it promotes the use of renewable energy as a source of automotive fuel, benefits the country economically due to the reduction of imported diesel fuel, and promotes renewable energy resources and reduction of air pollution.

1.6 Delimitation (Scope)

Detailed experimental study of the effect of diesel-ethanol blend on CI engine performance is very much time consuming and expensive. This thesis work includes the development of software programming using C++ based on mathematical modeling of CI engine performance which reduces time and expenses to study of the effect of diesel-ethanol blends in CI engine.

1.7 Limitations

The limitations of this paper are that not include the detailed experimental study of the effect of diesel-ethanol blend on CI engine performance is very much time consuming and expensive. The study is also limited to the effect of diesel-ethanol blends on CI engine performance and does not consider other alternative fuels or engine types.

1.8 Organization of the thesis

The thesis is organized in to five chapters

Chapter One: **Introduction** is an introduction to the present thesis. Statement of the problem to carry out the present thesis is given. Further, the objectives of the present research are presented also significance of the thesis, scope of the thesis and limitations are presented. At the end of the chapter, the thesis organization is briefly described.

Chapter Two: **Literature Review**, presents the literature review. The review discusses alternative fuel, ethanol and diesel fuel. The chapter also discusses importance using blended fuels and various alternative fuels as automotive fuel also diesel and ethanol fuel blend as automotive fuel discussed. Research gaps from the literature review are presented at the end of the chapter.

Chapter Three: **Materials and Methods**, presents the methods used to carry out this thesis and the selected engine specifications are presented. Further, mathematical model for actual engine along with computer code using “C++” described.

Chapter Four: **Result and discussion**, Deals the thermodynamics analysis of CI engine using a computer programming (C++) and discussion on the results obtained. Further, the thermodynamics analysis of ethanol-diesel fuel (E5, E10 and E15) on the selected CI engine discussed in terms of engine performance, combustion and emissions characteristics and compared with pure diesel also validation of the study using the work done on the same engine experimentally.

Chapter Five: **Conclusions and Recommendations**. It discusses the conclusions of the present research work. It also presents the outcomes of the present research. Finally, the future scope of research is presented.

CHAPTER TWO

LITERATURE REVIEW

2.1 Alternative fuel

Fuels are the materials that store potential heat energy in features that can be practicably released and used for work or as heat. This day, one of the most important problems in the use of internal combustion engines fuels is the production of harmful emission gases. For this reason, many studies have been carried out to reduce the emissions while maintaining of engine performances. One of the strategies for reducing the dependency of fossil fuels is to produce use of alternatives to gasoline and diesel fuel (Yahuza et al. 2015). These alternatives fuels include biofuels, gaseous fuels, renewable fuels, electricity, and fuels derived from coal. Some of the commonly used and blooming alternative fuels are biofuels. Biofuels are a renewable, biodegradable, combustible liquid or gaseous fuel derived from biomass or other renewable resource that can be used as transportation fuel and combustion fuel. Biofuels includes biodiesel or renewable diesel, renewable gasoline and alcohols, which include Ethanol, methanol and biodiesel (Brent et al. 2004).

2.2 Ethanol

Ethanol with the molecular formula C_2H_6O is a clear, colorless, volatile liquid with a pleasant smell made by fermentation of sugar. Ethanol is one of many kinds of alcohol and is the only type of alcohol that can be consumed. Apart from consumption, ethanol is used for several other purposes such as fuel to power engines. Ethanol is a clear liquid that made by the fermentation of different biological materials (Onyekwelu, 2018). Ethanol (C_2H_6O) is an alcohol, a group of chemical compounds whose molecules contain a hydroxyl group, –OH, bonded to a carbon atom. the most suited renewable, bio based and eco-friendly fuel for internal combustion engine, the most attractive properties of ethanol as an internal combustion engine fuel is that it can be produced from renewable energy sources such as sugarcane, cassava, many types of waste bio mass materials, corn and barley. In addition, ethanol has higher heat of vaporization, oxygen content and flammability temperature and therefore has a positive influence on engine performance and emission characteristics of CI engine. Ethanol has higher miscibility in diesel than methanol (Pirouzfar and Fayyazbakhsh, 2015). Ethanol as a fuel primary benefit is that it burns much cleaner than petroleum-based fuels. Ethanol (ethyl alcohol, grain alcohol) is a pure substance, colorless liquid with a characteristic, agreeable odor. Ethanol contains 35 percent oxygen, Ethanol burns well because it is an oxygenate alcohol, meaning that ethanol molecules contain oxygen. Oxygen atoms inside ethanol join forces with oxygen molecules in the air to help ethanol burn more completely. This extra amount of oxygen also helps gasoline or diesel burn better when it is blended with ethanol (Yahuza et al. 2015). Ethanol is already being used extensively as a fuel additive. Furthermore, the use of ethanol fuel alone or as part of a mix with gasoline is increasing. Compared to methanol, ethanol fuel's primary advantage is that the fuel is non-toxic, although the fuel will produce some amount of toxic exhaust emissions (Faizal et al. 2009). Different methods for adding ethanol to diesel fuel have been developed, for example: ethanol suspension (making an emulsifying dilution), spray the ethanol to diesel fuel and bilateral injections. The use of ethanol blended with diesel is receiving more attention by many researchers in recent times. It was shown that ethanol-diesel blends were technically acceptable for existing diesel engines. Ethanol as an attractive alternative fuel is a renewable bio-based resource and it is oxygenated, thereby providing the potential to reduce particulate emissions in compression-ignition engines (Yahuza et al. 2015).

2.3 Diesel Fuel

Diesel fuel is a mixture of hydrocarbons obtained by distillation of crude oil. Diesel fuel is commonly described as the middle distillate fractions of crude oil, which are separated from the lighter and heavier fractions during the refining process. Chemically, the liquid is composed of organic molecules (straight chain and aromatic) within a certain boiling range; these molecules consist primarily of carbon and hydrogen atoms. Certain straight chain molecules may require modification to provide the specified cold flow properties. If sulphur is present in crude oil molecules (as it may be naturally), it will need to be reduced or removed during the refining process to yield diesel fuel that will not damage engine or vehicle technologies, especially emission control equipment. Diesel fuel may include certain additives to improve its quality or increase the use of renewable fuel like ethanol (Huitema et al. 2019).

Diesel fuel, which is widely used in compression ignition engines, operates in diesel cycle. Due to the high compression ratio of diesel engine, typically between 12 and 24, thermal efficiency of diesel engines range is from about 35 to 40 percent. One of the most important properties of a diesel fuel is its readiness to auto-ignite at the temperatures and pressures present in the cylinder while the fuel is injected. Cetane is a measure of the compression ignition behavior of a diesel fuel; higher cetane level enable quicker ignition (Faizal et al. 2009). Cetane influences cold start ability, exhaust emissions and combustion noise, with the magnitude of the impact depending on the overall combustion and emission control strategies and their application. In general, higher cetane enables improved control of ignition delay and combustion stability, especially with modern diesels, which use high amounts of exhaust gas recirculation (EGR). Higher cetane provides room for engine calibrators to tailor combustion for the best calibration compromise among combustion noise, emissions and fuel consumption goals across the engine operating range. Cetane number is a very important property of diesel fuel. Cetane is measured or derived in various ways. The cetane number is produced by testing the fuel in a test engine. When the fuel does not contain any cetane improver, the cetane number is the same as the fuel's natural cetane. The derived cetane number, which is produced using a combustion tester, is an indirect measure of combustion ignition behavior that is equated to the cetane number. The cetane index is calculated from certain measured fuel properties (fuel density and distillation temperatures); it is designed to approximate the natural cetane. Use of cetane improver additives to increase cetane more than 10 numbers over the natural cetane is not recommended. Blending with high cetane fuels, such as renewable hydrocarbon diesel, is the preferred approach for increasing cetane (Huitema et al. 2019). One of the environmental problems that occurs with the combustion of diesel fuel is the sulfur contained in the fuel and exhaust gas emissions from diesel engines these are nitrogen oxides and particulates, of which nitrogen oxides (NOx), are usually the most critical (Faizal et al. 2009). Diesel may contain ethanol as an alternative fuel, for economic and environmental reasons. Diesel fuel is a mixture of a few hundred hydrocarbons derived from refining of crude petroleum. When petroleum prices were low, the diesel fuels were produced mostly by blending various refinery streams from the atmospheric distillation unit of petroleum refineries. To meet the increasing demand of the diesel fuels, products of secondary refinery processes like thermal and catalytic cracking, hydro-cracking, vis-breaking etc., also are used as blending components of the current diesel fuels.

The important diesel fuel quality parameters are discussed below.

2.3.1 Ignition quality

Ignition quality is a measure of ease of self-ignition of diesel fuel when the fuel is injected in hot compressed air in the engine cylinder. Cetane number (CN) is the most widely accepted measure of ignition quality as it is measured by a test on the engine. The cetane number scale is defined in terms of blends of two pure hydrocarbons used as reference fuels.

2.3.2 Density

The density of diesel fuel varies generally in the range 810 to 880 kg/m^3. The density, volatility, cetane number, viscosity and heat of combustion of petroleum fuels are interrelated. An increase of 10 percent in density increases the volumetric energy density ($MJ/\ m^3$) of the fuel approximately by 6 percent. The balance 4 percent is accounted for by the decrease in heat of combustion (MJ/kg) for the 10% heavier (higher molecular weight) fuels are metered volumetrically by injection pumps. The fuel density affects engine calibration and power as the fuel mass injected/stroke varies with fuel density. High-density fuels also have a higher viscosity thus, influence injection characteristics. Increase in the fuel density advances the dynamic injection timing by up to 1º CA (1º CA stands for 1-degree crank angle. It is a unit of measurement used in internal combustion engines to describe the timing of fuel injection. The crank angle is the angle of rotation of the engine’s crankshaft, which is a key component in the engine’s operation. By advancing the dynamic injection timing by up to 1º CA, it means that the fuel injection is timed to occur slightly earlier in the engine’s combustion cycle, which can have an impact on the engine’s performance and emissions.). Thus, the fuel density affects engine combustion and emissions. PM emissions generally increase with increase in fuel density. As the fuel injection system is calibrated for a particular fuel density, the current fuel specifications set narrow limits acceptable fuel specific gravity from 0.82 to 0.85.

2.3.3 Viscosity

The viscosity of diesel has a strong influence on fuel atomization, a high viscosity fuel resulting in larger fuel droplets. An increase in viscosity reduces spray cone angle and increases spray penetration. Low viscosity on the other hand, results in an increase in leakage of fuel past the pumping elements and loss of fuel system calibration. High viscosity of fuel is necessary for lubrication and protection of the injection equipment from wear.

2.3.4 Sulphur Content

Typically, increase of 500 ppm in sulphur content contributes to about 0.01 g/kWh increase in diesel PM emissions. Sulphur on combustion produces sulphur dioxide (SO_2), of which about 1 to 3% is oxidized to sulphur trioxide (SO_3) and forms sulphates found in particulate emissions. The balance of SO_2 is exhausted as gas. The sulphur trioxide on combining with water forms sulphuric acid that causes wear of engine cylinder liner and piston rings. Sulphur increases deposit formation in the combustion chamber and the deposits become harder in presence of sulphur. In most countries during early 1990s, sulphur content of diesel fuel was in the range from 0.2 to 0.5% (2000 to 5000 ppm) by mass. After the year 2000, a number of European countries made available the diesel fuels with less than 0.005% (50 ppm) sulphur. A large number of countries around the world have diesel fuels with sulphur below 0.05% for road vehicle application.

2.3.5 Cetane number

Diesel fuels are compared using an ignition delay metric and classified by cetane number (CN). The cetane number characterizes the ability of the fuel to auto ignition. The higher the cetane number, the shorter the ignition delay, as the ignition delay decreases approximately linearly with cetane number. If the cetane number is too low, the fuel will not ignite until late in the injection process. Delay period is the one of the most important roles plays in diesel engine, in which the delay period varies with chemical and physical properties of the diesel or biodiesel fuels. To differ the delay period cetane number improvers are the main important parameter. To minimize the ignition delay during the combustion cetane number improvers help to get better working in diesel engine ignition (Ferguson et al. 2001). During the combustion process, shorter ignition delay period leads to more complete combustion of the fuel there in the internal combustion engine, reduces the noise of the engine and also reduces the unwanted harmful pollutants in the environment. Cetane numbers for vehicular diesel fuels range from about 40 to 55. Additives such as nitrate esters can be used to increase the cetane number (Kumar et al. 2018).

2.4 Chemical and Physical Characteristics of Ethanol and Diesel Fuels

2.4.1 Characteristics of Diesel (A Hydrocarbon)

Diesel and gasoline fuels, which are derivatives of crude oil, are generally used in internal combustion engines. The approximate elemental structure of an average crude oil consists of 84% carbon, 14% hydrogen, 1–3% sulfur, and less than 1% nitrogen, oxygen atoms, metals, and salts. Crude oil consists of a wide range of hydrocarbon compounds consisting of alkenes, alkenes, naphthene, and aromatics. These are very small molecular structures such as propane (C_3H_8) and butane (C_4H_{10}) but can also be composed of mixtures of various structures with very large molecules such as heavy oils and asphalt. Therefore, crude oil needs to be distilled to be used in internal combustion engines. Because of heat distillation of crude oil, petroleum derivatives such as diesel, petroleum gases, jet fuel, kerosene, gasoline, heavy fuels, machine oils, and asphalt are obtained. In general, the distillation of crude oil resulted in an average of 30% gasoline, 20–40% diesel, 20% of heavy fuel oil and heavy oils from 10 to 20% are obtained. During the distillation of crude oil, gasoline is obtained between 40 and 200°C and diesel fuel is obtained between 200 and 425°C. In order to use these fuels in engines, some of the important physical and chemical properties such as specific gravity of the fuel, structural component, thermal value, flash point and combustion temperature, self-ignition temperature, vapor pressure, viscosity of the fuel, surface tension, freezing temperature, and cold flow properties are required.

The American Petroleum Institute (API) number is an international measurement system that classifies crude oil according to its viscosity. According to the American standards, the specific gravity can be defined as the ratio of the weight of a given volume of a given substance at 15.15°C (60°F) to the weight of the water at the same volume and temperature. According to the API number, crude oil is divided into three groups as heavy, medium, and light, and as the number of the API increases, crude oil becomes thinner. The API degree of diesel fuels varies between about 25 and 45. The viscosity, color, main component, and definition of crude oil according to the API grade are shown below in Table 2.1.

Table 2. 1. Classification of crude oil according to API grade (Emily Geary, 2017)

API grade	Definition	Viscosity	Color	Composition
0–22.3°	Heavy	Too viscous	Dark	Asphalt
22.3–31.3°	Medium	Medium	Brown	Diesel + gasoline
22.3–31.3°	Light	Fluid	Light yellow	Gasoline

The higher the diesel index value, the fuel is more alkane (in paraffinic structure), and it has the higher ignition tendency. Increasing volatility in diesel fuels causes acceleration of fuel evaporation and decrease in viscosity. This is generally undesirable since the fuel causes a reduction in the cetane number. Some fuels commonly used in engines are presented in Table 2.2 shown below. Some of the important properties of fuels such as the closed formulas, molar weight, lower heating value and higher heating value, stoichiometric air/fuel and fuel/air ratios, evaporation temperature. The evaporation temperature refers to the temperature at which a liquid change into a gas through the process of evaporation. When a liquid is exposed to an environment with a temperature equal to or higher than its evaporation temperature, the liquid molecules gain enough energy to overcome the intermolecular forces holding them together. As a result, they escape from the liquid surface and enter the surrounding air as gas molecules. motor octane number (MON), and cetane number are given.

Table 2. 2. Common fuels and their properties (Emily Geary, 2017)

Fuel	Closed formula	Mole weigh	Heating value		Stoichiometric		MON	CN	Evaporation T^0(kJ/kg)
			HHV (kJ/kg)	LHV (kJ/kg)	(A/F)s	(F/A)s			
Light diesel	$C_{12.3}H_{22.2}$	170	44,800	42,500	14.5	0.069	-	40-50	270
Heavy diesel	$C_{14.6}H_{24.8}$	200	43,800	41,400	14.5	0.069	-	35	230

High cetane means the fuel will ignite quickly at the conditions in the engine (does not mean the fuel is highly flammable or explosive). The low cetane number of diesel engines leads to an increase in ignition delay ID time, which is the time interval between the start of injection and the start of combustion, which in turn reduces the time required for combustion. An increased ID time leads to accumulate more fuel in the combustion chamber than required. Thus, this excess fuel causes sudden and high-pressure increases during the onset of combustion. These sudden pressures increase cause mechanical stresses and hard engine operation, which is known as diesel knocking. Fuel additives can improve Diesel and gasoline fuel properties such as cetane number, octane number, viscosity, and density. One of the most promising fuel additives are alternative fuels in the future. High octane number and low-density propensities of the alcohols lead to be improved the fuel properties such as increases the octane number of gasoline and decreased the viscosity, density properties of the diesel fuel. Besides, the diesel fuel cetane number can be improved by biodiesel, which has a high cetane number (Sarıkoç et al. 2020).

2.4.2 Characteristics of ethanol

The chemical compound ethanol, also known as ethyl alcohol or grain alcohol, is the bio-alcohol found in alcohol lic beverages. When non-chemists refer to "alcohol", they usually mean ethanol. It is also increasingly being used as a fuel (usually replacing or complementing gasoline). Ethanol's chemical formula is C_2H_5OH. Pure ethanol is a flammable, colorless liquid with a boiling point of 78.5° C. Its low melting point of -114.5° C allows it to be used in antifreeze products. It has a pleasant odor reminiscent of whiskey. Its density is 789 g/l about 20% less than that of water. It is easily soluble in water and is itself a good solvent, used in perfumes, paints and tinctures. Alcoholic drinks have a large variety of tastes, since various flavor compounds are dissolved during brewing. Ethanol is a polar molecule due to its hydroxyl (OH) group, which forms hydrogen bonds with other molecules. Ethanol appears as a clear colorless liquid with a characteristic vinous odor and pungent taste. Flash point is 12.7°C. Density is 778.87 g/l. Vapors are heavier than air. Ethanol is a clear, colorless, very mobile liquid. It is used in alcoholic beverages in suitable dilutions, and as a reagent in synthetic organic chemistry and chromatography, as well as industrial and laboratory organic solvent. Ethanol is flammable and burns more cleanly than many other fuels. When fully combusted its combustion, products are only carbon dioxide and water. For this reason, it is favored for environmentally conscious transport schemes and use as vehicle fuel. Ethanol's specific gravity is 0.79, which indicates it is lighter than water but since it is water-soluble, it will thoroughly mix with water. Ethanol has an auto-ignition temperature of 422°C and a boiling point of 78°C. Ethanol is less toxic than gasoline or methanol. Ethanol is a polar solvent that is water-soluble and has a 12.7°C flash point. Ethanol has a vapor density of 1.59, which indicates that it is heavier than air. Ethanol can be used as fuel for internal combustion engines either directly or in blends. Ethanol is having higher octane number that is why it allows for more compression ratio and it emits very less toxic in the environment as compared to petroleum-based fuel (Rajnish et al. 2019).

2.5 Characteristics of ethanol-diesel blended fuel

Numerous techniques have been evaluated to allow for the concurrent use of diesel and ethanol in compression ignition engines. Some of these techniques include alcohol fumigation, dual injection, alcohol-diesel fuel emulsions, and alcohol-diesel fuel blends. Among these approaches, only alcohol-diesel emulsions and blends are compatible with most commercial diesel engines. Since emulsions are difficult to achieve and tend to be unstable, blends—either as micro-emulsions or using co-solvents—are the most common approach as they are stable and can be used in engines with relatively no modifications. Blends of ethanol with diesel fuel are often referred to as "E-Diesel" or "eDiesel". Sometimes, ethanol-diesel blends are also called "oxygenated diesel" a term that is not particularly precise, as diesel blends containing methyl ester biodiesel or any other additive that includes oxygen can be also described as oxygenated diesel. In e-diesel blends, standard diesel fuel (such as US No. 2) is typically blended with up to 15% (by volume) of ethanol using an additive package that helps maintain blend stability and other properties most importantly cetane number and lubricity. The additive package may comprise from 0.2 to 5.0% of the blend. There is currently no specification for e-diesel in the USA. The use of e-diesel can bring some reductions in diesel PM emissions, while contradictory reports exist on its effect on NOx, CO, and HC emissions. Perhaps the biggest advantage of e-diesel is its renewable character, if renewable ethanol is used as the blending stock. Considering its potentially significant operational and safety issues the latter including very low flash point e-diesel will likely remain a niche market fuel of limited applicability. Ethanol fumigation into the engine intake port is an alternative way to use ethanol in the diesel engine. This method requires modifications to the engine; its application remains even more limited. The idea of Diesel-ethanol blends is not new. Research studies dating since the 1980s have shown that these blends are suitable for use in compression ignition (CI) engines. Pollutant emissions improvements associated with the use of DE blends in CI engines (without any modifications) are strongly dependent on the operating conditions of the respective engine. The combination of various alcohols, including ethanol with diesel fuel, in addition to creating a clean fuel, increase safety of the fuel in the past, there was no a stable and single-phase diesel. But today this problem solved by adding ethyl ester octyl nitrate methyl esters Nitro methane, Nitro ethane and 2-methoxy ethyl ether as a third additive or stabilizer (Şahin et al. 2007). As discussed earlier in this section there are different methods to add ethanol to diesel fuel one of these methods is direct injection which ethanol add to diesel fuel directly. When this method is used, there is no problem in combination of diesel with alcohol (ethanol) but when in temperature degree lower than 10 °C, this mixing is not done well and creates a two-phase dilution. As noted earlier, ethanol could be added to diesel fuel without produce any two-phase dilution in another method we should blend the ethanol droplets as a kind of emulsion with a novel emulsifier, so fuel efficiency and combination stability increase. On micro emulsion aqueous ethanol with 5 % water and diesel fuel and reported that this compound is very stable and it burns a little later to diesel fuel base (Brent et al. 2004).

2.6 Compression-ignition engine

The compression-ignition engine is one type of engine based on the injection of the fuel into a cylinder with air, and on the self-ignition of the fuel due to the temperature of the compressed air. In the compression-ignition engine, combustion occurs in various locations throughout the cylinder with no organized flame propagation. To satisfy this type of ignition and combustion process, the compression ignition engine must utilize fuels that can readily self-ignite. This fuel is typically a diesel fuel, but jet fuel and other oils can be used. The compression-ignition engine generally requires a relatively high compression ratio to generate sufficient high temperature air for the auto-ignition process. These engines typically must operate with excess air (fuel lean) to ensure all the fuel is burned. In many applications, the compression-ignition engine uses intake air compression (turbochargers and superchargers) to increase its power density (Jerald, 2016).

2.7 Thermodynamic engine cycle simulations

Thermodynamic engine cycle simulations have been developed to help understand, correlate, and analyze the operation of engine cycles. These include combustion models, models of physical properties, and models of flow into, though, and out of the cylinders. Even though models often cannot represent processes and properties to the finest detail, they are a powerful tool in the understanding and development of engines and engine cycles. Models range from simple and easy to use, to very complex and requiring major computer usage. In general, the more useful and accurate models are quite complex. Models to be used in engine analysis are developed using empirical relationships and approximations, and often treat cycles as quasi-steady-state processes.

2.7.1 Combustion in CI engine

Combustion in CI engines differ from SI engine due to the basic fact that CI engine combustion is unassisted combustion occurring on its' own. CI engine the fuel is injected into combustion space after the compression of air is completed. Due to excessively high temperature and pressure of air the fuel when injected in atomized form gets burnt on its' own and burning of fuel is continued till the fuel is injected. Theoretically this injection of fuel and its' burning should occur simultaneously up to the cut-off point, but this does not occur in actual CI engine. Different significant phases of combustion are explained as under. Stages of combustion in CI engine are discussed below.

A. Ignition delay period

Injection of fuel in atomized form is initiated into the combustion space containing compressed air. Fuel upon injection does not get burnt immediately instead some time is required for preparation before start of combustion. Fuel droplet injected into high temperature air first gets transformed into vapor (gaseous form). Subsequently, if temperature inside is greater than self-ignition temperature at respective pressure then ignition gets set. Thus, the delay in start of ignition may be said to occur due to 'physical delay' i.e., time consumed in transformation from liquid droplet into gaseous form, and ‘chemical delay’ i.e., time consumed in preparation for setting up of chemical reaction (combustion). The duration of ignition delay depends upon fuel characteristic, compression ratio (i.e., pressure and temperature after compression), fuel injection, ambient, air temperature, speed of engine, and geometry of combustion chamber etc. Ignition delay is inevitable stage and in order to accommodate it, the fuel injection is advanced by about 20º before TDC. Ignition delay is shown by 1 – 2 in Fig 2.1, showing pressure rise during combustion. Fuel injection begins at ‘1’ and ignition begins at ‘2’. Theoretically, this ignition delay should be as small as possible.

B. Uncontrolled combustion

During the ignition delay period also the injection of fuel is continued as it has begun at point ‘1’ and shall continue up to the point of cut-off. For the duration in which preparation for ignition is made, the continuous fuel injection results in accumulation of fuel in combustion space. The moment when ignition just begins, if the sustainable flame front is established then this accumulated fuel also gets burnt rapidly. This burning of accumulated fuel occurs in such a manner that combustion process becomes uncontrolled resulting into steep pressure rise as shown from ‘2’ to ‘3’. The uncontrolled burning continues till the collected fuel gets burnt. During this ‘uncontrolled combustion’ phase if the pressure rise is very abrupt then combustion is termed as ‘abnormal combustion’ and may even lead to damage of engine parts in extreme conditions. Thus, it is obvious that ‘uncontrolled combustion’ depends upon the ‘ignition delay’ period as during ignition delay itself the accumulation of unburnt fuel occurs and its’ burning results in steep pressure rise. Hence in order to have minimum uncontrolled combustion the ignition delay should be as small as possible. During this uncontrolled combustion phase about one-third of total fuel heat is released.

C. Controlled combustion

After the ‘uncontrolled combustion’ is over then the rate of burning matches with rate of fuel injection and the combustion is termed as ‘controlled combustion’. Controlled combustion is shown between ‘3’ to ‘4’ and during this phase maximum of heat gets evolved in controlled manner. In controlled combustion phase rate of combustion can be directly regulated by the rate of fuel injection i.e., through fuel injector. Controlled combustion phase has smooth pressure variation and maximum temperature is attained during this period. It is seen that about two-third of total fuel heat is released during this phase.

D. After burning

After controlled combustion, the residual if any gets burnt and the combustion is termed as 'after burning' point '4'.

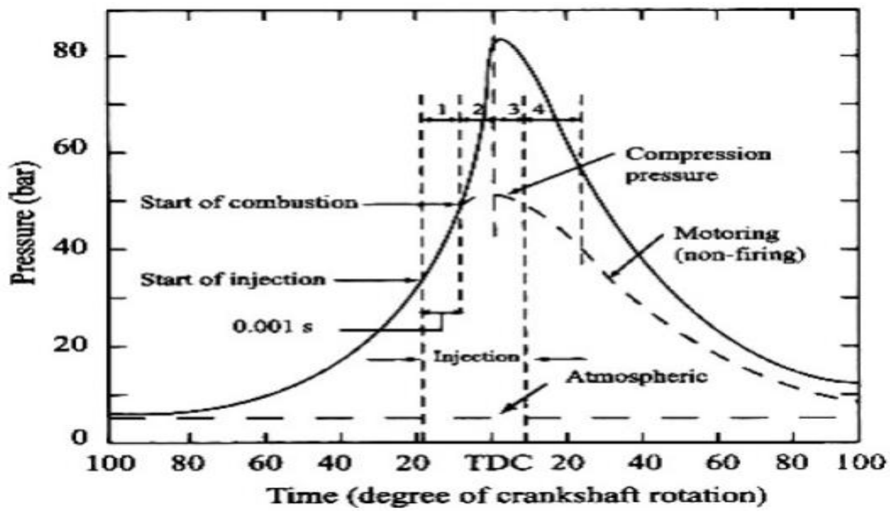

Figure 2. 1. Combustion stages of CI engine. (Ferguson et al. 2001).

This after burning may be there due to fuel particles residing in remote position in combustion space where flame front could not reach. 'After burning' is spread over 60 – 70º of crank angle rotation and occurs even during expansion stroke (Ferguson et al. 2001).

2.7.2 Engine combustion modeling

Modeling, in science and engineering, may be generally regarded as the process of describing the physical phenomena in a particular system with the help of mathematical equations (subject to "reasonable" assumptions) and solving the same to understand more about the nature of such phenomena. Usually, engineering models help in designing better devices by understanding more about the fundamental physical processes occurring therein. Engine modeling activities, at least in recent decades, have largely been concentrated in the direction of designing better performing engines with lower emissions. In this regard, modeling of engine combustion processes assumes importance. The thermodynamic or fluid dynamic model was developed based on energy conservation or complete analysis of fluid motion in the cylinder predicting the performance and combustion characteristics of the engine. Zero-dimensional model predicts and analyses the characteristics of thermodynamic properties of the engine solving energy equations whereas multi-dimensional momentum conservation equations allow visualizing the gas flow and combustion products. Therefore, zero-dimensional model is suitable for observing the effects of variation in heat release rate and in-cylinder pressure parameters of engine operation. The zero-dimensional model used in the study was based on the first law of thermodynamics applied to a closed cycle for duration when both the inlet and exhaust valves are closed (i.e. for the period of combustion). The energy equation for above process is written as $\frac{dU}{dt} = Q - W$

where U is the internal energy of the system, Q is the heat supplied to the system, and W is the work done by the system (Hariram et al., 2016).

Some of the engine combustion models that are mainly used are discussed below.

2.7.3 Zero dimensional models

Zero dimensional models are the simplest and most suitable to observe the effects of empirical variations in the engine operating parameters. These models are zero dimensional in the sense that they do not involve any consideration of the flow field dimensions. Zero dimensional models are further sub-divided into single zone, two zone and multi zone.

In single zone models, the working fluid in the engine is assumed to be a thermodynamic system, which undergoes energy and/or mass exchange with the surroundings and the energy released during the combustion process is obtained by applying the first law of thermodynamics to the system.

In two zone models, the working fluid is imagined to consist of two zones, an unburned zone and a burned zone. These zones are actually two distinct thermodynamic systems with energy and mass interactions between themselves and their common surroundings, the cylinder walls. The mass-burning rate (or the cylinder pressure) as a function of crank angle is numerically computed by solving the simplified equations resulting from applying the first law to the two zones (Hariram et al., 2016).

These models have been traditionally used in two different directions as shown in Figure below.

1. In one way, both these models have been used to predict the in-cylinder pressure as a function of crank angle from an assumed energy release or mass burned profile (as a function of crank angle)

2. Another use of these models lies in determining the energy release/mass burning rate as a function of crank angle from experimentally obtained in-cylinder pressure data.

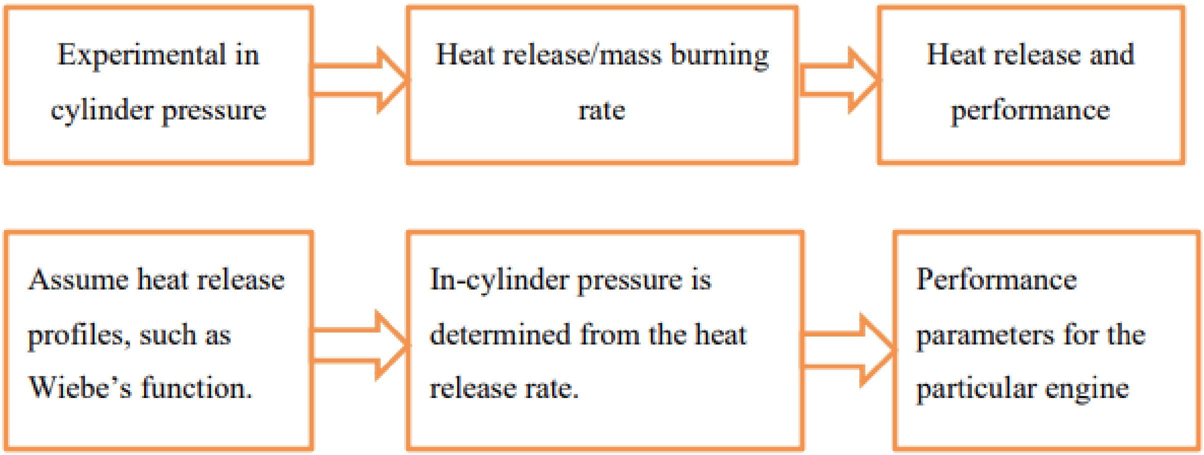

Figure2.2. the directions followed by different variations of the single and two zone models for different purposes. (Pavel et al. 2004)

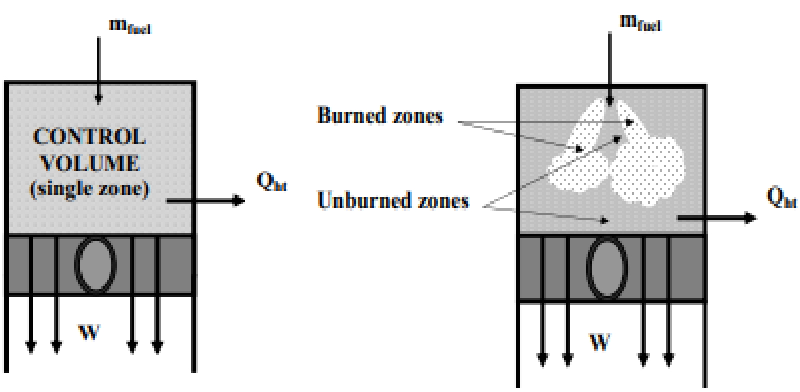

Figure 2. 2. Schematic diagram of the single zone and two zone combustion models depicting energy and mass transfers (Pavel et al. 2004)

The overall goal of such thermodynamic engine simulations is to mathematically simulate the significant processes, and to predict engine performance and engine operation details including certain emission parameters (Pavel et al. 2004).

2.7.4 Governing equations in Thermodynamic Engine Cycle Simulations

The governing equation for calculating variation of in-cylinder pressure, temperature and volume with respect to crank angle is presented in Equation 2.1. The zero-dimensional model used in the study was based on the first law of thermodynamics applied to a closed cycle for duration when both the inlet and exhaust valves are closed (i.e. for the period of combustion). (Pulkrabek, 2006). The energy equation is written as equation 2.1.

$$\frac{dU}{d\theta} = \frac{\partial Q}{d\theta} - \frac{\partial W}{d\theta} \qquad 2.1$$

Where, $\frac{dU}{d\theta}$, is the rate of change of internal energy of the system, $\frac{\partial Q}{d\theta}$, is the net apparent rate of heat release which is defined as difference between $\frac{\partial Q_{in}}{d\theta}$, heat generated by the fuel due to combustion and $\frac{\partial Q_{loss}}{d\theta}$, heat that is convicted to the cylinder walls due to in cylinder gasses i.e. $\frac{\partial Q}{d\theta} = \frac{\partial Q_{in}}{d\theta} - \frac{\partial Q_{loss}}{d\theta}$ and $\frac{\partial W}{d\theta}$, denotes rate of work transfer out of the system.

The rate of change of internal energy of the system and the rate of work transfer out of the system can be expressed in terms of M, C_v, T, p and V which are in-cylinder instantaneous mass, specific heat of gases at constant volume, instantaneous temperature, in-cylinder pressure at the given instant of crank angle and instantaneous volume respectively. The above equation is rearranged as equation 2.2.

$$\frac{\partial Q_{in}}{d\theta} - \frac{\partial Q_{loss}}{d\theta} = MC_v \frac{dT}{d\theta} + P \frac{dV}{d\theta} \qquad 2.2$$

2.8 Literature reviews of previous work

Li et al. (2004): studied experimentally the effects of different ethanol–diesel blended fuels on the performance and emissions of diesel engine. The experiments were conducted on a water-cooled single-cylinder Direct Injection (DI) diesel engine using 0% (neat diesel fuel), 5% (E5–D), 10% (E10–D), 15% (E15–D), and 20% (E20– D) ethanol–diesel blended fuels. With the same rated power for different blended fuels and pure diesel fuel, the engine performance parameters including power, torque, fuel consumption, and exhaust temperature and exhaust emissions [CO, NOx, total hydrocarbon (THC)] were measured. The results indicate that, the brake specific fuel consumption and brake thermal efficiency increased with an increase of ethanol contents in the blended fuel at overall operating conditions; smoke emissions decreased with ethanol–diesel blended fuel, especially with E10–D and E15–D. CO and NO x emissions reduced for ethanol–diesel blends

Tarek et al. (2019): In this research fuel properties of ethanol blended with diesel were experimentally determined to find their stability and to increase their properties and efficiency in the diesel engines. Four blends of diesel with ethanol and the fifth sample was pure diesel. The samples were 0% ethanol and 100 % diesel, the second sample was 5% ethanol and 95 % diesel, the third sample was 10 % ethanol and 90% diesel, the fourth sample was 15 % ethanol and 85 % diesel and the fifth and last sample was 20 % ethanol and 80 % diesel. Many tests and experiments done during this research project and the obtained results were similar to the EURO 5 standard emissions regulation and an optimum blend of ethanol diesel blend which was 10% ethanol by volume to meet the EURO 5 regulations and standards for reducing emissions

Achmad et al. (2014): This paper addresses the possibility study of ethanol percentage to changing of performance and exhaust emissions for Diesel Engine using Virtual Engine Simulation Tool AVL Boost. In this study the blend formulation between Ethanol and Diesel Fuel were E0, E2.5, E5, E7.5 and E 10. The direct blending of ethanol and diesel fuel has advantages reducing exhaust emissions CO, Soot and NOx percentages. The engine power break of pure diesel is slightly lower than those of E2.5-E10, especially for speed above than 1400 rpm.

Patil et al. (2013): The simulation model by MATLAB program for numerical solution was used to analyze the engine parameters of a single cylinder 3.5 kW rated power diesel engine fuelled with diesel, Palm Oil Methyl Ester and POME-diesel blends. The results reported that, the simulated results on the brake thermal efficiency and in-cylinder pressure were closer by about 2–3% to the experimental results. A single-zone thermodynamic model was developed for a diesel engine fuelled with biodiesel from waste. The single zone model coupled with a triple-Wiebe function was performed to simulate heat release and cylinder pressure. It was reported that, the heat release rate and cylinder pressure predicted were 2.5% and 2.2% closer to the experimental results of the engine.

Iliev et al. (2019): In this study, the effects of ethanol-diesel (E5, E10 and E15) and methanol–diesel (M5, M10 and M15) fuel blends on the emissions of a diesel engine were investigated by using single-cylinder, four-stroke, direct injection compression ignition engine. The results obtained from the use of ethanol-diesel and methanol–diesel fuel blends were compared to those of pure diesel fuel (E0 and M0). One-dimensional simulation of a single-cylinder diesel engine with the help of the GT-Power software has been done. This software is capable of predicting the exhaust emissions. The simulation is designed to test the characteristics of the diesel engine working with methanol-diesel blends at full load. From the results obtained, it becomes clear that with the increase of the load the temperature of the exhausted gas increases. With an increase in the ethanol and methanol content (E10 - E15) and (M10 - M15), a slight increase in the exhaust gas temperature is noticeable and is the highest at E15 and M15.

Şahin et al. (2007): the effects of the using of ethanol-diesel fuel blends on the engine performance characteristics such as brake specific fuel consumption (BSFC), brake effective power, brake effective efficiency and exhaust emissions have been investigated theoretically in different direct-injection (DI) diesel engines by using a multi-zone thermodynamic model. The results indicate that as ethanol percentage in the mixture increases, BSFC reduces and brake effective efficiency improves significantly and brake effective power increases slightly. On the other hand, equivalence ratio decreases and ignition delay increases for ethanol blends and combustion duration indicates generally a decreasing tendency. The concentrations of nitric oxide (NO), carbon monoxide (CO) and hydrogen (H_2) increase at low ethanol ratios because of increment of cylinder temperatures. But at high ethanol percentage ratios they decrease because of decrement of cylinder temperatures.

Midhat et al. (2019): Experimental tests of diesel engine using Ethanol–Diesel Fuels blends have been done on a four stroke, one cylinder; water cooled C.I. Engine (Lister Engine) with a pump injection fuel system. Performance tests were conducted for fuel consumption rate, brake power, engine torque, brake thermal efficiency, indicated thermal efficiency, mechanical efficiency, brake mean effective pressure, indicated mean effective pressure, and brake specific fuel consumption, using different load and pure unleaded diesel at different load. From the experimental and the results which was obtained during testing four different blends. It can be concluded that, the optimum alternative fuel for CI engine that has been investigated by this experimental study is the pure unleaded diesel fuel.

Gnanamoorthi et al. (2009): Studied about engine performance, combustion and exhaust emission characteristics of a single cylinder four stroke diesel engine using ethanol blended in different ratios with diesel fuel. The test was performed with five different blends of ethanol (E0-neat diesel, E10, E20, E30, E40 and E50). To satisfy homogeneity and prevent phase separation, 3% of Ethyl acetate has been added to ethanol diesel blend. The result of engine test showed that the addition of ethanol caused an increase in the performance which includes improved Brake thermal efficiency by 8% for 10% ethanol addition and only 2% increase on further ethanol addition with an added advantage of reduced specific fuel consumption and the increase of ethanol, ignition\retards and combustion duration shorten, which results in rapid combustion. Ethanol heat release rate increases with increasing ethanol concentration. Considering emissions, CO, HC and smoke are increased at lower loads but reduced at higher loads as ethanol which is an oxygenated fuel provides free oxygen to assist combustion. The NOx is reduced by 8% with 10% of ethanol blend.

2.9 Summary of literature review

The previous related work studied experimentally and determined the fuel properties of ethanol blended with diesel. There are number of papers available which study the effect of ethanol-diesel blend fuel in compression ignition engine and there are more studies carried out in this area of alternative fuel but no common understanding is taken still now. The methodologies used by different authors were experimental analysis and using software's like using AVL Boost purchased from software companies. From the literature review of pervious works, that there is no study carried out that specifically mentions the standard and optimum amount of ethanol for compression ignition engines, besides there are studies that discuss the use of ethanol in CI engines in different aspect.

Literature survey of the existing researches on Diesel fuel, Ethanol fuel and various previous works done related to the effect of Diesel-Ethanol blend fuel in compression ignition engine were referred in order to have comprehensive understanding of the subject under research.

2.10 Research gap

Most of the pervious works related to performance of CI engine with ethanol-diesel blend were focused on experimental research. Very less work reported on simulation of CI engine with ethanol-diesel blend using commercial software's like AVL BOOST which are expensive. It is observed that the results obtained by different researchers were varying and standardization was not achieved. There is a need of further investigation using less expensive means.

In this thesis an effort is made to develop software using C++ which can help in investigation of CI engine performance in a quickly and reduces expenses of experimentation.

CHAPTER THREE

MATERIALS AND METHODS

3.1 Introduction

In this chapter, the various steps involved in methodology followed and the materials are given.

3.2 Methods

The methodology followed in achieving the set objectives, includes various stages as briefly described below. The stages include literature review, mathematical modeling of engine performance, and simulation of engine performance using the developed software programming using C++. The simulation results were analyzed to evaluate the effects of using diesel-ethanol blended fuel on engine performance and emissions. It is shown in the form of a flowchart in Figure 3.1.

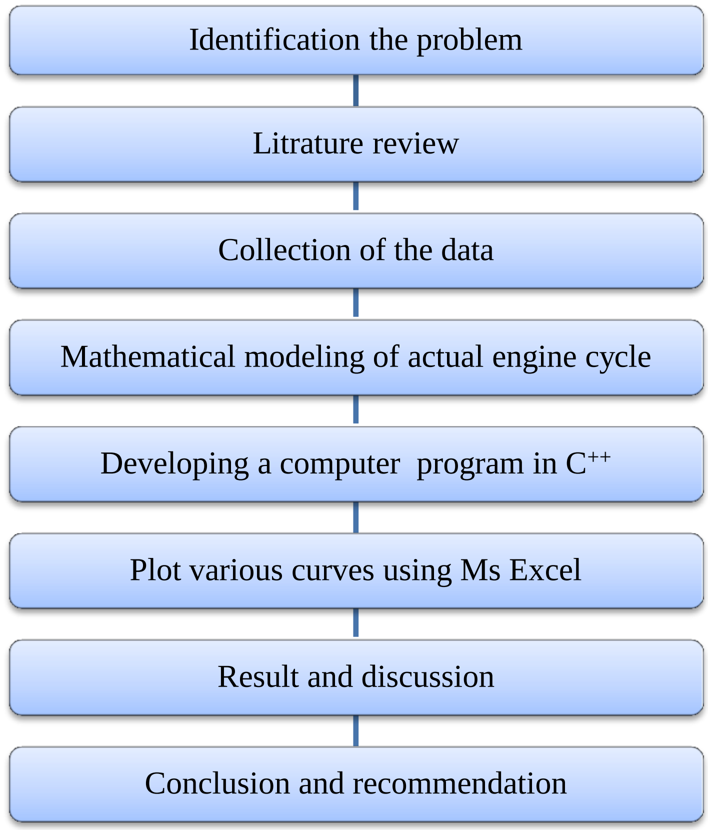

Figure 3. 1. Methods

3.4 Methods of data collection

The data pertaining to the specifications of the engine were from the manual. The physical and chemical characteristics of Diesel and Ethanol were collected from the literature.

3.5 Materials

IC engines obtain their energy from the combustion of a hydrocarbon fuel with air, which converts chemical energy of the fuel to internal energy in the fuel within the engine. There are many thousands of different hydrocarbon fuel components, which consist mainly of hydrogen and carbon but may also contain oxygen (alcohols), nitrogen, and/or sulfur. Among different hydrocarbon fuel types and alcohols light diesel and ethanol specifications are chosen for this particular work.

3.5.1 Properties of diesel fuel and ethanol

Among different hydrocarbon fuel types and alcohols, E5 (contains 5% ethanol and 95% diesel fuel by mass), E10 (contains 10% ethanol and 90% diesel fuel by mass) and E15 (contains 15% ethanol and 85% diesel fuel by mass) were chosen for the selected CI engine in this particular work. The diesel fuel and ethanol properties are shown in the Table 3.1 (Heywood, 1988).

Table 3. 1. Diesel fuel and Ethanol characteristics

Characteristics	Ethanol	Light Diesel
Chemical formula	C_2H_5OH	$C_{10}H_{22}$
Molecular weight	46	147.66
Density at 15℃ [kg/m^3]	790	820
LHV[MJ/kg]	27	42.5

3.5.2 Engine specifications

Generally, diesel engine vehicles are categorized into three different groups as light duty vehicles including minibus, cargo trucks and bus. The total number of light duty diesel vehicles available in Addis Ababa that were manufactured after 2000 EC indicates 34.1 % and light duty diesel vehicles (Addis Ababa institute of technology, 2012). For this particular work 4-cylinder direct injection engine were selected for simulation purpose and to validate the program experimentally investigated engine from the literature (Gvidonas et al. 2014) is taken. Engine specifications of the selected engine shown in Table 3.2.

Table 3. 2. Engine specifications (Gvidonas et al. 2014)

Engine Specification	
Engine Code/Name	In-line, 4 stroke, water-cooled
Fuel type	Diesel
Engine capacity	2986 cc
Number of cylinders	4
Injection pump	Mechanical
Displacement	3.0 L, 2,986 cm^3
Compression	16:1
Bore	110 mm
Stroke	125 mm
Fuel system	direct injection (DI)
Maximum Power	66 kW @ 4000rpm
Maximum Torque	192Nm @ 2400rpm
IVOBTDC	10°
IVCABDC	35°
EVOBBDC	35°
EVCATDC	10°

3.6 Mathematical model for actual engine cycle

3.6.1 Engine geometry

The cylinder volume at each crank angle position or the instantaneous total volume of gases in the cylinder can be obtained by knowing the geometrical engine parameters and using the equation of crank-slider mechanism as shown in the Figure 3.2.

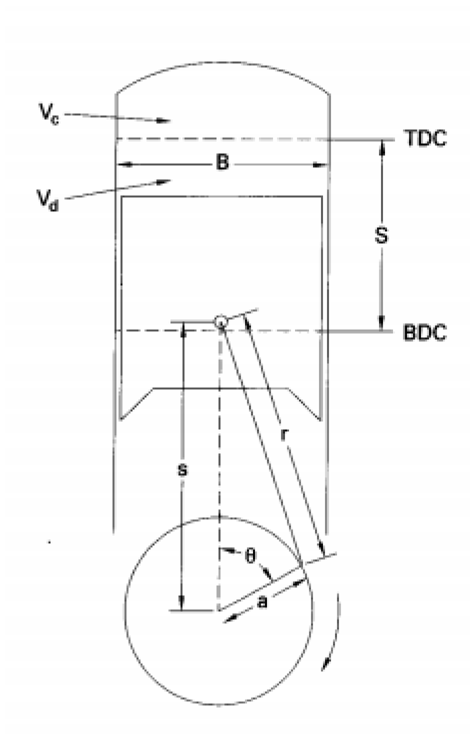

Figure 3. 2. Crank-slider mechanism of reciprocating engine (Ferguson, 2001)

Where bore B (mm) is the diameter of each cylinder, stroke S (mm) is the distance travelled by the piston from top dead center (TDC) to bottom dead center (BDC), l (mm) is connecting rod length; a (mm) is crank offset or crank radius (half stroke), s (mm) is position of piston pin from crank center, θ is crank angle (from TDC), V_c (mm) is clearance volume and V_s (mm) is the swept volume.

The following parameters define the basic geometry of a reciprocating engine. Swept volume (V_s) is the volume displaced by the piston as it travels from BDC and TDC. Some books call this displacement volume (V_d) as it is the difference between the maximum and minimum volume, swept volume is given as equation 3.1

$$V_s = \frac{E_c}{Z_c} \qquad 3.1$$

Where, E_c is engine capacity, Z_c is number of cylinders

Clearance Volume V_c is minimum cylinder volume occurs when the piston is at TDC as equation 3.2.

$$V_c = \frac{V_s}{r_c - 1} \qquad 3.2$$

Where r_C is compression ratio usually compression ignition (Cl) engines have compression ratios in the range 12 to 24, in this case the value of r_C is 22.

For the simulation, an important item is the instantaneous cylinder volume and it is determined from simple geometry.
The cylinder volume at any crank angle position, θ, is given as equation 3.3

$$V(\theta) = V_c + \frac{\pi}{4} B^2(x) \tag{3.3}$$

where x is the instantaneous stroke distance from top dead center, given as equation 3.3.

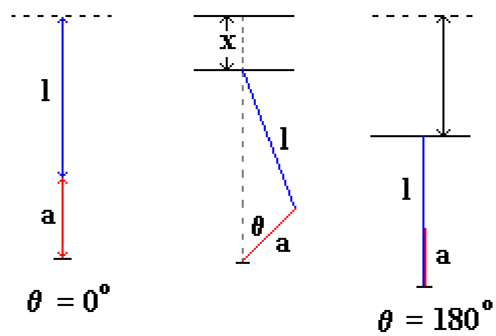

Figure 3. 3 Piston cylinder geometry

Again, using geometry, a relationship for x can be developed:

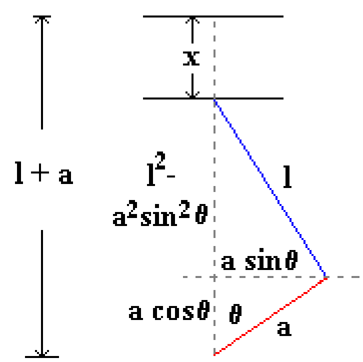

Figure 3. 4 Geometric solution for x

$$x = \alpha + l - [(l^2 - a^2 \sin^2 \theta)^{1/2} + a \cos \theta] \tag{3.4}$$

Where Crank radius (α) is the distance between the crank center and the crank pin, i.e., half of the stroke. (l) is the distance between the piston pin and the crank pin.

Substituting the maximum volume with the displacement volume yields the equation is given as equation 3.5. Githens (2020)

$$V(\theta) = V_c + \frac{\pi}{4} B^2 [\alpha + l - [(l^2 - a^2 \sin^2 \theta)^{1/2} + a \cos \theta] \qquad 3.5$$

3.6.2 Zero-dimensional combustion model

By considering the mixture inside the combustion chamber as a closed system then for small change of the process, the first law of thermodynamics is simply written as equation 3.6.

$$dU = \partial Q - \partial W \qquad 3.6$$

Therefore, by using the definition of work, the first law can be expressed as equation 3.7.

$$\partial Q_{in} - \partial Q_{loss} - (PdV) = dU \qquad 3.7$$

For an ideal gas the equation of state is expressed as equation 3.8.

$$PV = ZmRT \qquad 3.8$$

The compressibility factor (Z) is a dimensionless quantity that describes how much a real gas deviates from ideal gas behavior. It is defined as the ratio of the molar volume of a gas to the molar volume of an ideal gas at the same temperature and pressure. For an ideal gas, Z equals to one because the behavior of an ideal gas exactly follows the ideal gas law. However, real gases deviate from this behavior, especially at high pressures and low temperatures. Therefore, Z is used to correct for these deviations when using the ideal gas law for real gases and the value is found from compressibility chart. By differentiating above equation given as equation 3.9.

$$PdV + VdP = ZmRdT \qquad 3.9$$

Also, for an ideal gas with constant specific heats the change in internal energy is expressed as equation 3.10.

$$dU = m\, c_v dT \qquad 3.10$$

By substituting equation 3.10 in to above equation 3.9, it become

$$dU = \frac{C_v}{R}(PdV + VdP) \qquad 3.11$$

Then, the first law becomes,

$$\partial Q - PdV = \frac{C_v}{R}(PdV - VdP) \qquad 3.12$$

Further reducing the above equation,

$$\frac{\partial Q}{d\theta} - \left(1 - \frac{C_v}{R}\right) P \frac{dV}{d\theta} = \frac{C_v}{R} V \frac{dP}{d\theta} \qquad 3.13$$

Using $R = c_p - c_v$ and $n = c_p / c_v$, to define $\frac{C_v}{R} = \frac{1}{n-1}$ then, the energy equation after rearrangement becomes as equation 3.14.

$$\frac{\partial Q}{d\theta} = \frac{1}{n-1} V \frac{dP}{d\theta} + \frac{n}{n-1} P \frac{dV}{d\theta} \qquad 3.14$$

From the definition of work

$$\partial Q_{in} - \partial Q_{loss} - (PdV) = \frac{C_v}{R}(PdV + VdP) \qquad 3.15$$

The total amount of heat input to the cylinder by combustion of fuel in one cycle is expressed as equation 3.16.

$$Q_{in} = m_f \, Q_{LHV} \eta_c \qquad 3.16$$

In a CI engine the cylinder pressure is depends on the fuel burning rate during combustion. The high cylinder pressure ensures the better combustion and heat release. The total heat added from the fuel to the system until the crank position reaches angle θ is given as equation 3.17.

$$Q(\theta) = Q_{in} X_b \qquad 3.17$$

The total amount of heat loss from the system when the crank moves an increment of d (θ)

$$Q_{loss} = \frac{h_{cv} A_h}{\omega} (T - T_w) d(\theta) \qquad 3.18$$

Where h_{cv} is heat transfer coefficient,

By substituting Equations (3.18)– (3.17) into Equation (3.14) followed by differentiation with respect to crank angle (θ) the following equation is obtained for solving the equation for pressure as a function of crank angle given as equation 3.19.

Thus, explicitly solving the equation for pressure as a function of crank angle as given in equation

$$\frac{dP}{d\theta} = \frac{n-1}{V} \left[Q_{in} \frac{dx_b}{d\theta} - \frac{h_{cv} A_h}{\omega} (T - T_w) \frac{\pi}{180} \right] - n \frac{P}{V} \frac{dV}{d\theta} \qquad 3.19$$

Where n is the polytropic exponent

Once the pressure is calculated, the temperature of the gases in the cylinder can be calculated using the equation of state as equation given as equation 3.20.

$$T(\theta) = \frac{P(\theta)V(\theta)}{mR} \qquad 3.20$$

Where R is universal gas constant, m is mass of the fuel/air mixture, P is pressure and V is volume.

3.6.3 Heat release model

In this present study, the combustion model was developed based on equally distributed heat release phenomena (i.e. heat is liberated evenly from all the areas of the combustion chamber). Since single zone combustion model predominantly give the accurate results in case of constant pressure combustion, Wiebe's function was employed to determine the mass fraction of the fuel burnt and which in turn applied for finding the heat release rate due to combustion of fuel.

Wiebe's function was employed to determine the mass fraction of the fuel burnt and which in turn applied for finding the heat release rate due to combustion of fuel. The Wiebe heat release pattern is based on the exponential rate of the chemical reactions. x_b is the burned fraction of the fuel and is expressed as equation 3.21.(Heywood, 1988). The fraction of heat released is expressed by the non-dimensional equation.

$$x_b = 1 - \exp(-a\left(\frac{\theta-\theta_{sc}}{\theta_{dc}}\right)^{m+1}) \qquad 3.21$$

Where x_b denotes mass fraction of fuel burnt at given crank angle, θ and θ_{dc} represents the duration of combustion, θ is the crank angle, θ_{sc} is the start of heat release, constants a and m 6.9 and 3, respectively for CI engine, the value of the m Wiebe form factor and a Wiebe efficiency factor was used by Wiebe in his engine modeling calculations. (Heywood, 1988).

The Wiebe function can be used for modeling the energy release in a wide variety of combustion systems. From the governing equation described in chapter two, $\frac{\partial Q_{in}}{d\theta}$ is the heat release rate and by differentiating equation 3.21. The rate of heat released can be expressed as equation 3.22 and equation 3.23.

$$\frac{\partial Q}{\partial \theta} = Q_{in}\frac{\partial x_b}{\partial \theta} \qquad 3.22$$

$$\frac{\partial Q}{\partial \theta} = a(m+1)\left(\frac{\theta-\theta_{sc}}{\theta_{dc}}\right)^2 \exp[-a(\left(\frac{\theta-\theta_{sc}}{\theta_{dc}}\right)^{m+1} \qquad 3.23$$

Where θ is crank angle, θ_{sc} is start of energy release, θ_{dc} is duration of energy release, m is Wiebe form factor and a is Wiebe efficiency factor.

Wiebe efficiency factor is the parameter which characterizes the completeness of combustion, Wiebe assumed that maximum $x_b = 0.990$ and hence $a = 6.0908$. also, Wiebe form factor characterizing the rate of combustion, the small value for m means a high rate at the beginning of combustion, while the large value for m means a high rate by the end of combustion (Ganesan, 2000).

3.6.4 Combustion chemistry

In a combustion process of an internal combustion engine the fuel and the oxidizer air react to produce products of different compositions. The exhaust gas contains incomplete combustion products like H_2O, CO, H_2, O2, H_2, O, NOx, N_2, HC, Soot particles (which are mainly solid carbon) etc. But to calculate the molar fraction of all combustion products need a very complex mathematical model. Due to this reason, in the present case only five combustion products in the combustion process are considered.

The chemical formula of a fuel contains ethanol defined as $C_aH_bO_c$ and assume the following reaction, atom balancing yields of the following reaction is given as equation 3.24

$$C_aH_bO_c + \lambda a_s(O_2 + 3.76N_2) \rightarrow n_1CO_2 + n_2H_2O + n_3N_2 + n_4O + n_5CO \qquad 3.24$$

where a_s , the stoichiometric molar air-fuel ratio, and the moles (n_i = 1, 2, 3) that describe the product composition.

At low temperatures (T< 1000 K, such as in the product gases in the exhaust stream) the overall complete combustion reaction for any equivalence ratio can be written as equation 3.25

$$C_xH_yO_z + \lambda a_s(O_2 + 3.76N_2) \rightarrow n_1CO_2 + n_2H_2O + n_3N_2 + n_4O_2 + n_5CO \qquad 3.25$$

This equation assumes the dissociation of reactants is negligible because dissociation level of the in-cylinder combustion products is small. This chemical reaction is applicable for lean, stoichiometric, or rich mixtures. Solutions for both rich and lean cases are given in Table 3.3.

Table 3. 3 Solutions for the molar coefficients of the products (J. B Heywood, 1988).

Species	n_i	$\lambda > 1$	$\lambda < 1$
$C0_2$	n_1	a	$a - n_5$
H_2O	n_2	$b/2$	$b/2 + n_5$
N_2	n_3	$\frac{(\lambda)(3.76)(2)}{2}$	$\frac{(\lambda)(3.76)(2)}{2}$
O_2	n_4	$2\lambda ath - (2n_1) + n_2)$	0
CO	n_5	0	n_5

.

Mole fraction of a solute is the ratio of the number of moles of that solute to the total number of moles of the solute and solvent in the solution (always less than one) can be found as in equation 3.26.

$$y_i = n_i/n_{tot} \qquad 3.26$$

Here, x_i is the molar fraction of the i^{th} component, n_i is the mole of the i^{th} component in the mixture, and n_{tot} is the total mole of the mixture. also know, the total mole of a mixture is the sum of the mole of each component in the mixture.

$$n_{tot} = \sum n_i \qquad 3.27$$

The molar mass of the mixture is determined as

$$M_m = \sum_{i=1}^{n} y_i M_i \qquad 3.28$$

Where M_i is the molecular weight of mixture is written as equation 3.29.

$$M_m = y_f M_f + y_a M_a \qquad 3.29$$

3.6.5 Emission prediction

In diesel engine, maximum concentration of NO_X in engine exhaust gas shifts toward somewhat higher air/fuel ratio. The concentrations of the different pollutants are plotted as a function of the air/fuel ratio (λ) in Figure 3.5. The curve of the NO_X concentration rises continuously as (λ) drops. Also, other major pollutant concentrations in diesel engine exhaust gas with a varied air/fuel ratio based on Zeldovich reaction equilibrium is presented in handbook of diesel engine. Diesel engine emission prediction model is done by using curve fitting method considering the emission graph behavior from handbook of diesel engine (Klaus et al. 2010). Considering the overall behavior of internal combustion engine emission graph for both lean and rich mixture curve fitting equations are prepared as shown below for each emission parameters.

$$y_{CO} = y_{COmax}\, x^{-0.65} \quad 3.30$$

$$y_{NOx} = y_{NOxmax} - x^3 + x^2 + x \quad 3.31$$

$$y_{PM} = y_{PMmax}\, x^{-0.85} \quad 3.32$$

$$y_{HC} = y_{HCmax}(x^2 - 1.2x + 4.2) \quad 3.33$$

Where y_{CO}, y_{NOx}, y_{PM}, y_{HC} are emissions of CO, NOx, PM, and HC, $x = \lambda / \lambda yNOxmax$

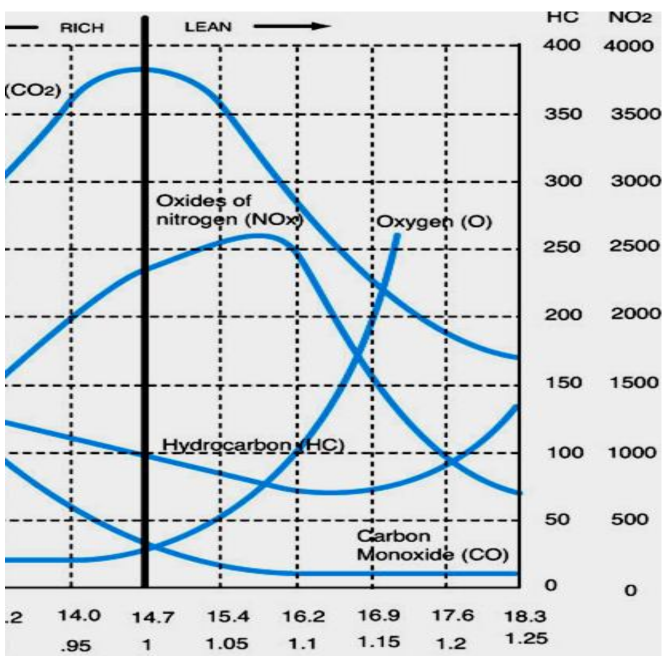

Figure 3. 5. Pollutant concentrations in diesel engine with a varied air/fuel ratio (Klaus et al. 2010).

Based on the curve fit equations each emission will be plotted. Consider the mole fraction oxygen from the chemical equilibrium reaction from Table 3.3. and divided by sum of the mole of each component in the mixture and curve fitting method as shown above, the graph can be plotted.

NOx emission simulation result from the using the above curve fitting are converted to g/kWh according to the following equation 3.34 (Duy et al. 2020).

$$NOx\left(\frac{g}{kWh}\right) = NOx\ (ppm)\frac{(m_{air}+m_{fuel})\ 1.587}{1000\ P_b\ (kW)} \qquad 3.34$$

3.6.6 Engine performance parameters

Having calculated the needed thermodynamic properties (P and T), the indicated power can be calculated by the integration of the PV diagram. Indicated power is given as equation 3.35.

$$P_i = Q_{in}\ \eta_{i,th} \qquad 3.35$$

Where Q_{in} is heat addition, $\eta_{i,th}$ is indicated thermal efficiency and P_i is indicated power.

$$Q_{in} = \ m_f\ Q_{LHV}\eta_c \qquad 3.36$$

And further explicit $m_f = (m_a)(AF)$ also $m_a = \rho\ \eta_v V_s\ n$

Where m_a mass of air is, m_f is mass of fuel, AF is air fuel ratio, ρ is density, η_v is volumetric efficiency, n is the number of revolutions per cycle.

Substituting the mass of air and mass of fuel in to equation, heat addition given as equation 3.37.

$$Q_{in} = \ \rho\ \eta_v V_s (AF) Q_{LHV}\eta_c\ n \qquad 3.37$$

Volumetric efficiency is given as equation 3.38.

$$\eta_v = \frac{m_a}{\rho\ V_s} \qquad 3.38$$

Indicated Power (P_i) is the power generated within the engine cylinder during the combustion process and used to push the piston. The indicated Power can be expressed as shown in equation 3.39.

$$P_i = Q_{in}\ \eta_{i,th}\ \text{or}\ P_i = i_{mep}\ V_s\ n \qquad 3.39$$

Where P_i is indicated power, $\eta_{i,th}$ is indicated thermal efficiency, n is the number of revolutions per cycle, so indicative mean effective pressure$i_{mep},$ can be expressed as equation 3.40.

$$i_{mep} = \eta_c \eta_{i,th} \eta_v (AF) Q_{LHV} \rho \qquad 3.40$$

The major performance parameters of IC engines are brake Power (Pb), brake Torque (Tb) and brake specific fuel consumption (bsfc).

Brake Power (Pb) is the net power available at the output shaft or at the flywheel, the net brake power for one cylinder given as equation 3.41.

$$P_b = P_i \, \eta_m \, Z \qquad 3.41$$

Where η_m is mechanical efficiency, P_b is brake power and Z is number of cylinders.

$$\eta_m = \frac{b_{mep}}{i_{mep}}(100) \qquad 3.42$$

where $b_{mep} = \frac{P_b}{E_c \, n}$ is brake mean effective pressure, $i_{mep} = b_{mep} + f_{mep}$ is indicated mean effective pressure and $fmep = 0.97 + (0.15(0.001\,N) + 0.05(0.001\,N)^2$ (Barnes, 1997).

Brake Torque (T_b) is the torque available at the output shaft of the engine or at the flywheel which is an engine performance parameter which mainly determines the effort or the traction, acceleration and uphill drive ability of the vehicle. Brake torque is given as equation 3.42.

$$T_b = \frac{P_b}{\omega} \qquad 3.42$$

Where $\omega = \frac{2\pi \times N}{60}$ which is angular velocity, T_b is brake torque and P_b brake power and N is Engine speed in rpm.

Brake specific fuel consumption (b_{sfc}) is an engine performance parameter which mainly determines the amount of fuel consumed by the engine in kilogram per hour (kg/h) to generate a brake power of one kilowatt (kW). Brake specific fuel consumption is given as equation 3.43.

$$b_{sfc} = \frac{m_f}{P_b} \qquad 3.43$$

Where b_{sfc} is brake specific fuel consumption, m_f is mass of fuel and P_b is brake power.

Combining analytical and empirical methods to model a diesel engine performance parameters such as brake effective power, brake effective efficiency, brake specific fuel consumption etc. using relationships given by (A.Kolchin, 1980). The number of engine performance parameters are determined by empirical relations obtained on the basis of processing many experimental data Since the selected engine is compression ignition direct injection engine. The computation points of the effective power curve are defined by the empirical relations equation 3.44.

$$P_b = P_{b\,max} (-x^3 + 1.4x^2 + 0.6\,x) \qquad 3.44$$

In the above formulae, $x = {}^{N}/_{N_{@P_{b\,max}}}$ $N_{@P_{b\,max}}$ is maximum speed at maximum brake power and P_b is brake power, $N\ (rpm)$ is engine speed in revolution pre minute and $P_{b\,max}$ is brake power analytical.

The specific effective fuel consumption, at the searched point of the speed characteristic for diesel engines with open combustion chambers given as equation 3.45.

$$bsfc = bsfc_{Pbmax}(x^2 - 1.55x + 1.55\,) \qquad 3.45$$

Where $bsfc$ is the specific effective fuel consumption at the rated power.

3.7 General steps used for solution

The C++ program begins with known engine inputs. The bore, stroke, connecting rod length, number of cylinders, compression ratio and operating characteristics has to be mentioned to the program as inputs for the validation of an experimental details. Based on the inputs script would calculate the area of the cylinder, clearance volume of the cylinder and surface area of the piston head. And the atmospheric conditions were chosen like, the initial inlet Temperature 300K (room temperature) and pressure as 1 atm and fuel inputs such as mass of the fuel, lower heating values and air-fuel ratio were taken as per the experiment model of the engine. The pressure and temperatures at each crank angle step in the cycle are obtained by solving the energy equation; the properties in the combustion period are calculated using Wiebe's heat release model. Pressures and temperatures with crank angle are plotted for the entire cycle. Emission formation is predicted using empirical data and curve fitting method. Brake power and brake specific fuel consumption is calculated by using appropriate formula based on empirical relation. Methodology of simulation process used in the thesis shown in flow chart diagram below

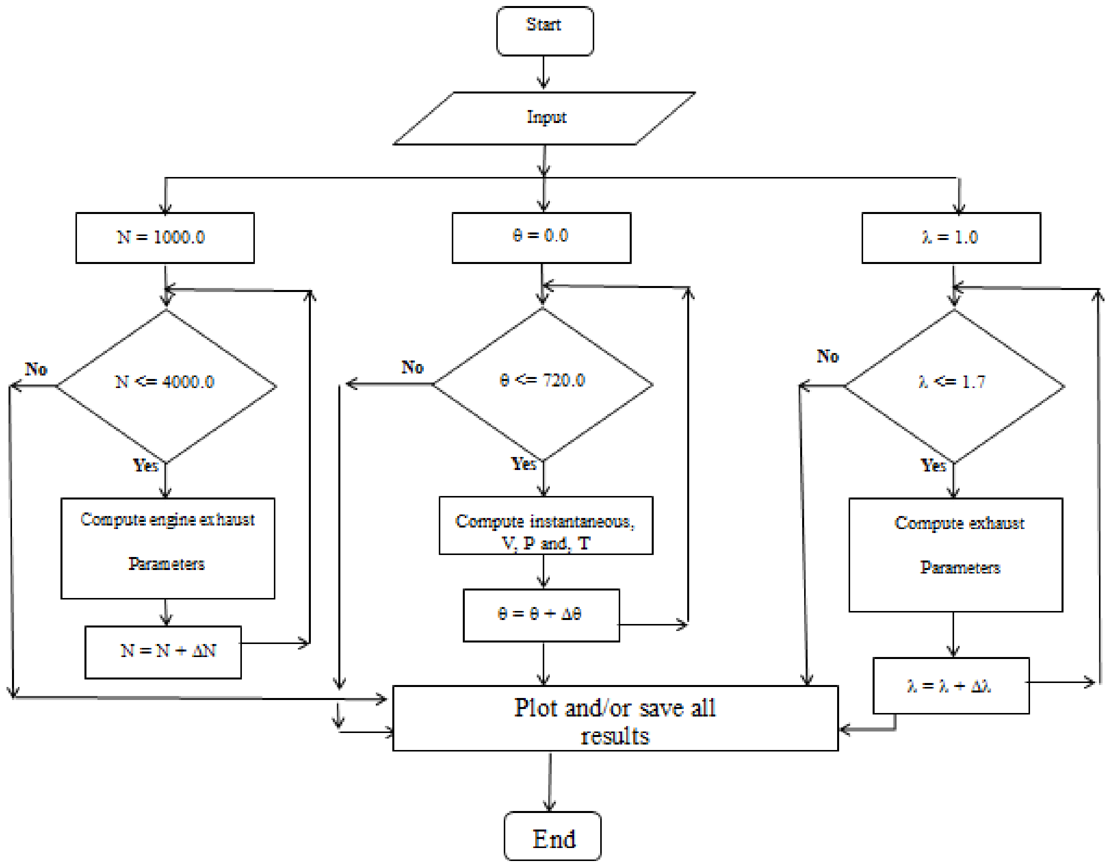

Figure 3.6 Methodology of simulation process flow chart diagram.

CHAPTER FOUR

RESULTS AND DISCUSSION

4.1 Introduction

This chapter discusses the results obtained from simulation of the selected engine. The major performance parameters (brake power, brake specific fuel consumption and brake thermal efficiency), combustion parameters (cylinder gas pressure and temperature, heat release rate and combustion efficiency) and exhaust emissions (CO, HC, NOx and PM) results are analyzed.

4.2 Combustion parameters

The major combustion parameters such as the variation of cylinder gas pressure and temperature, heat release rate and combustion efficiency are discussed with reference to the crank angle for various ethanol blends. Using a computer program in C++ together with an application software (MS-Excel) various important curves are plotted for each blend fuel (E5, E10 and E15). The results are compared with the expected experimental results obtained using commercial GT power software for the purpose of validation. Table 4.1 and Table 4.2 shown below are the output results from C++ programing for each blend and pure diesel.

Table 4. 1. Variation of pressure P with crank angle (θ) for E0, E5, E10 and e15

	p (bar)			
(deg.)	E0	E5	E10	E15
320	7.87	7.87	7.87	7.87
330	13.26	13.26	13.26	13.26
340	21.56	21.57	21.57	21.58
350	35.72	35.73	35.74	35.75
360	54.72	54.72	54.73	54.73
370	71.27	71.23	71.19	71.15
380	53.82	53.77	53.71	53.66
390	34.21	34.16	34.11	34.05
400	21.04	21.01	20.98	20.94
410	13.96	13.94	13.92	13.89
420	9.91	9.9	9.88	9.86

Table 4. 2. Variation of temperature T with crank angle (θ) E0, E5, E10 and e15

	T (K)			
(deg.)	**E0**	**E5**	**E10**	**E15**
320	651.218	651.218	651.218	651.218
330	745.575	745.575	745.575	745.575
340	782.428	782.608	782.802	783.013
350	851.973	852.21	852.467	852.747
360	1069.025	1069.058	1069.095	1069.135
370	1678.272	1677.401	1676.458	1675.433
380	1920.835	1919.083	1917.184	1915.119
390	1894.717	1892.02	1889.096	1885.918
400	1719.208	1716.761	1714.108	1711.225
410	1583.76	1581.505	1579.062	1576.405
420	1478.868	1476.763	1474.481	1472

Combustion analysis was made based on cylinder pressure, net heat release rate and mass fraction burnt at constant speed of 4000 rpm. The results were discussed as follows.

4.2.1 Cylinder Pressure

Figure 4.1 shows the variation of cylinder pressure with crank angle for various types of fuels. When the temperature of combustion chamber reaches near to the self-ignition condition of ethanol-diesel blend fuel, combustion takes place and hence causes rise in temperature and pressure. Figure 4.1 shows the variation of cylinder pressure. There are no such observable differences in the peak pressure due to blending a small percentage of ethanol with pure diesel. The main reason is that the presence of oxygen in ethanol which tends to promote combustion the of oxygen in ethanol can improve the combustion process in diesel engines by increasing the oxygen-fuel ratio which enhances the flame propagation and reduces the ignition delay also the higher oxygen content of ethanol which enhances the oxidation and combustion of the fuel.

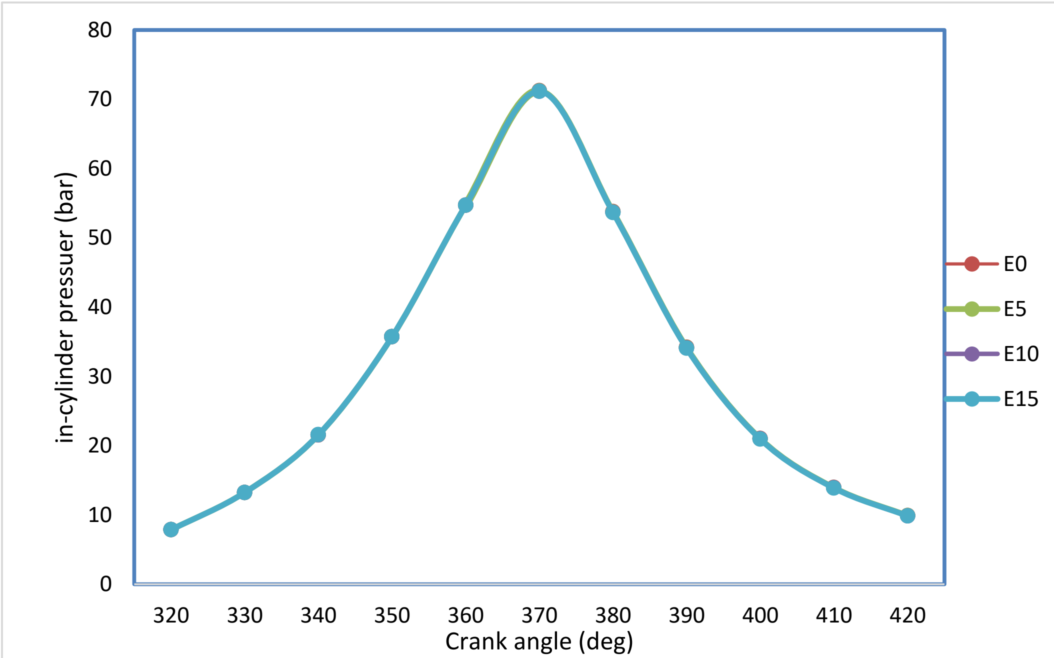

Figure 4. 1 In-cylinder pressure vs Crank angle

As shown in Figure 4.1, the maximum combustion pressure of E15, E10, E5 and E0 are 71.15 bar, 71.19 bar,71.23 bar and 71.27 bar respectively. The peak pressures in the cylinder are also indicated in Figure 4.2. In compression ignition engine, pressure is mainly dependent on the amount of fuel burnt during premixed combustion phase. If more fuel is burnt in this phase, then the peak cylinder pressure would be high. Also, the amount of fuel that will be burnt in premixed phase depends on ignition delay period. Ignition delay is the time between fuel injection and fuel ignition. If ignition delay is more, then premixed combustion phase is reduced and less amount of fuel will burn. This in turn reduces the peak pressure inside the engine cylinder. Since ethanol has a lower cetane number than diesel and it has a higher latent heat of vaporization than diesel, which means it absorbs more heat from the mixture during evaporation. Ethanol blend fuels have relatively higher ignition delay and lower combustion peak pressure than pure diesel fuel. This may affect the combustion noise, combustion efficiency.

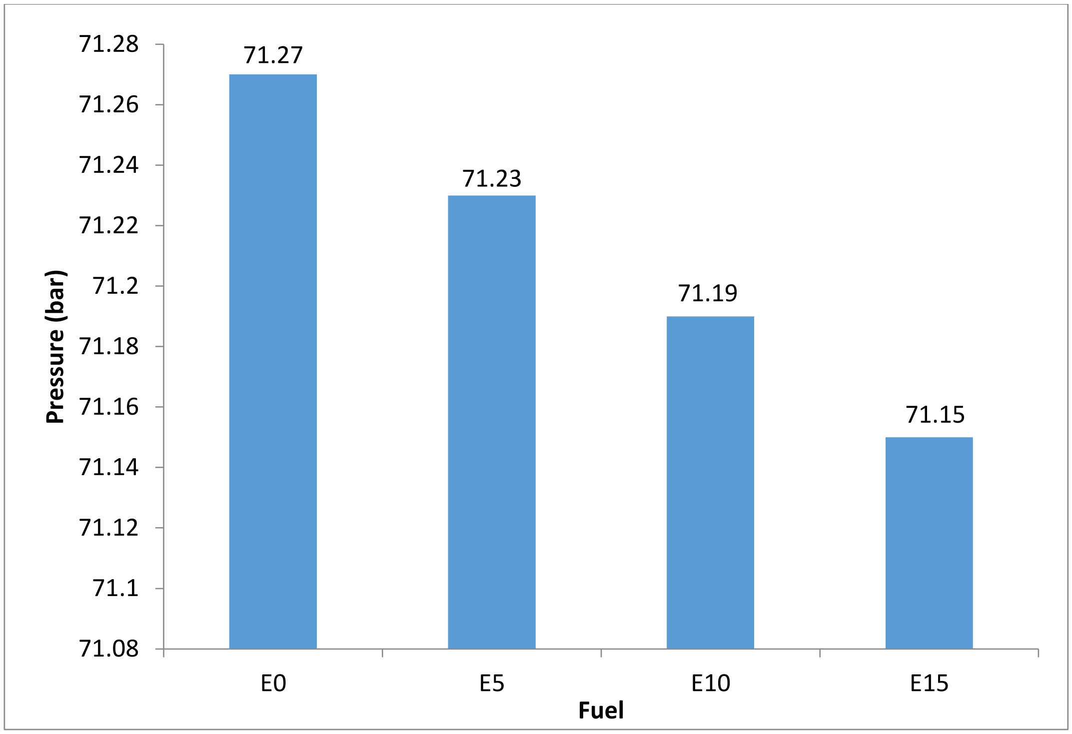

Figure 4. 2 Peak pressure developed in cylinder

Ignition delay is also dependent on the cetane number of the fuel. Here as shown in the above figure 4.2 the peak pressure developed is in pure diesel fuel (E0), this is due to due the pure diesel fuel has a higher cetane number than ethanol-diesel blend fuel, which means it has a higher reactivity and a shorter ignition delay under compression, Pure diesel fuel has a lower latent heat of vaporization than ethanol-diesel blend fuel and maintains a higher temperature and pressure in addition pure diesel fuel has a higher lower heating value than ethanol-diesel blend fuel, which means it releases more heat per unit mass during combustion and increases the peak pressure. These factors make pure diesel fuel more efficient and enhances the combustion process and increase peak pressure However, there are no such observable differences in the peak pressure due to blending a small percentage of ethanol with pure diesel.

4.2.2 Cylinder temperature

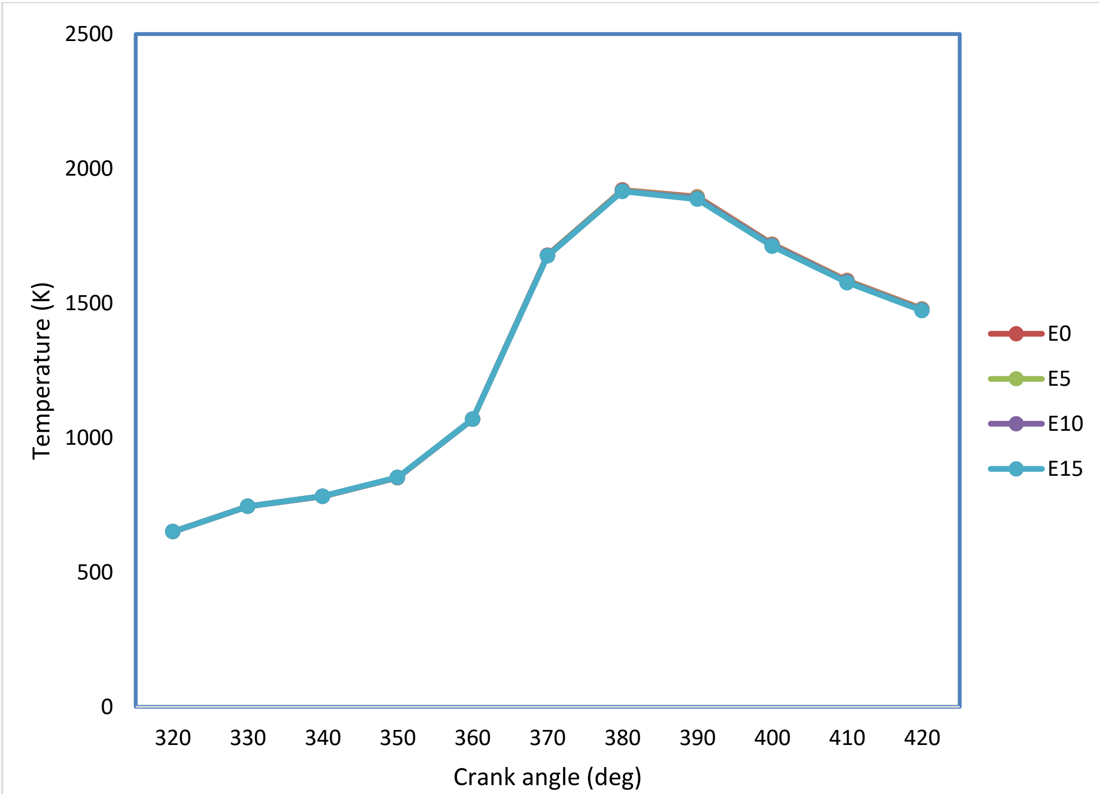

Figure 4. 3 Temperature vs crank angle

The obtained data related to cylinder temperature is given in Table 4.2. The variation of temperature with respect to crank angle is shown in Figure 4.3. The peak temperature obtained with fuels is given in Figure 4.4. ethanol-diesel fuel has a lower cetane number and calorific value than pure diesel, which means it takes longer to ignite and has less energy content. This affects the combustion process, which depends on the temperature and pressure of the air-fuel mixture. Higher temperature and pressure lead to faster and more complete combustion, while lower temperature result in slower and less efficient combustion. That is why, ethanol-diesel blending fuel has lower cylinder temperature than pure diesel. It is observed that the cylinder temperature obtained in pure diesel is more than ethanol-diesel blended fuels. The other reason is that the lower heating value of ethanol is less than that of pure diesel fuel. Lower QLHV means less heat energy and lower cylinder temperature but there are no such observable differences in the peak temperature due to blending a small percentage of ethanol with pure diesel.

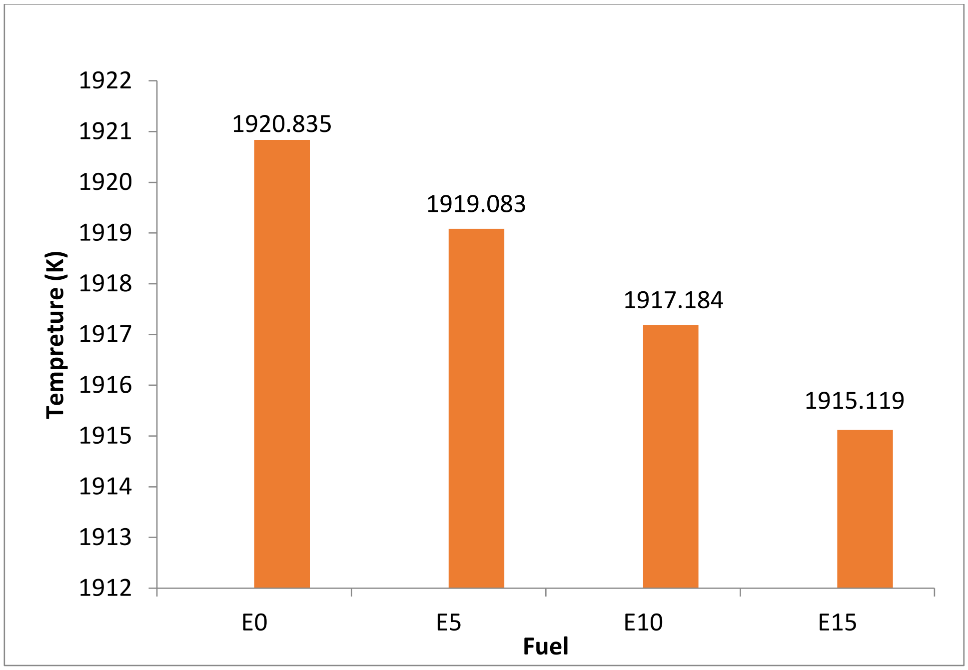

Figure 4. 4 Maximum temperature

The cetane number of a fuel is a measure of its ignition quality in a CI engine. It indicates how easily and quickly the fuel can self-ignite under high pressure and temperature conditions. A higher cetane number means a shorter ignition delay, which is the time interval between the start of injection and the start of combustion. A shorter ignition delay means less time for fuel-air mixing, less accumulation of unburned fuel, and less tendency to knock. A lower cetane number means a longer ignition delay, which leads to poor combustion efficiency. The cetane number of pure diesel is relatively higher than that of ethanol alcohol. Therefore, adding ethanol to diesel lowers the cetane number and increases the ignition delay. Because of this effect peak cylinder temperature is observed in pure diesel which is 1920.83 K. The effect of ethanol-diesel blends on peak temperature are depend on the percentage of ethanol that added on but there is no observable difference seen in the peak temperature.

4.2.3 Mass fraction burned

The Mass Fraction Burned (x_b) reflects the amount of fuel burned and the rate of burning during the combustion process in an internal combustion engine. The mass fraction graph indicates the start and end of combustion. Analytically in a compression ignition (CI) engine, these parameters are often modeled with the Wiebe function, mass fraction burn formulation, which is a function of “a” (efficiency parameter), “m” (form factor), crank angle, and the duration of combustion. The Wiebe function curve has a characteristic S-shaped curve and is commonly used to characterize the combustion process. The mass fraction burned profile grows from zero, where zero mass fraction burn indicates the start of combustion, and then tends exponentially to one indicating the end of combustion. The Figure 4.5 shows mass fraction vs crank angle diagram.

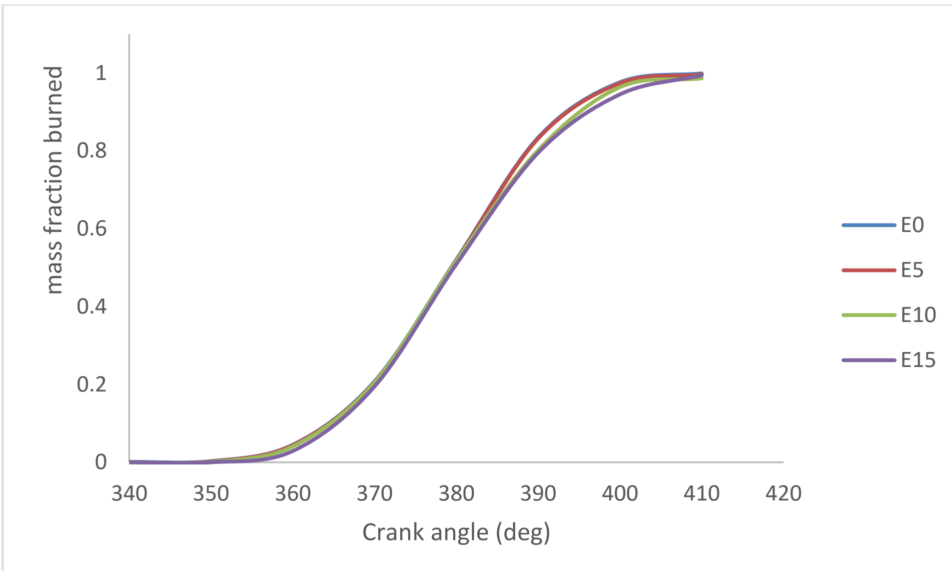

Figure 4. 5 Mass fraction burned vs crank angle

The above Figure 4.5 shows the rate of mass fraction burned with respect to crank angle of engine for all blends and pure diesel. The length of the burning process was same for all fuel blends but burning rates are slightly different. When ethanol blended percentage increases, the ignition delay increases due to the high evaporation latent heat and this may be due to the low boiling temperature of ethanol and low cetane number of ethanol. It is observed from figure 4.5 that mass fraction burned curve for ethanol-diesel fuels is similar to that of pure diesel, but slightly shifted to the right, indicating a later start of combustion and a longer duration of combustion process. This is because ethanol-diesel fuel has a lower cetane number than pure diesel, which means it is more resistant to auto-ignition.

4.2.4 Heat release rate (HRR)

The total heat released is the sum of heat released during the premixed and the diffusion (after burning) phase. The premixed phase is when the fuel and air are mixed before ignition, and the diffusion phase is when the fuel and air are mixed during combustion. The premixed phase usually has a higher heat release rate than the diffusion phase. Weibe functions were used to describe the heat released in both the premixed and the diffusion phases. Weibe functions is a mathematical function that describe the rate of heat release as a function of crank angle and it is widely used in engine simulation model. Heat Release is dimensionless curve. It is observed that due to relatively higher peak pressure and peak temperature of pure diesel fuel leads to relatively higher heat release rate of pure diesel fuel that means faster combustion. It is also seen that the volume of ethanol is directly having impact in this parameter. Increase in the volume of ethanol decreases the heat release rate. The maximum heat release rate for ethanol–diesel fuel blends is slightly lower as compared to that of pure diesel fuel as shown in Figure 4.6. but there is no observable difference seen in the peak heat release rate.

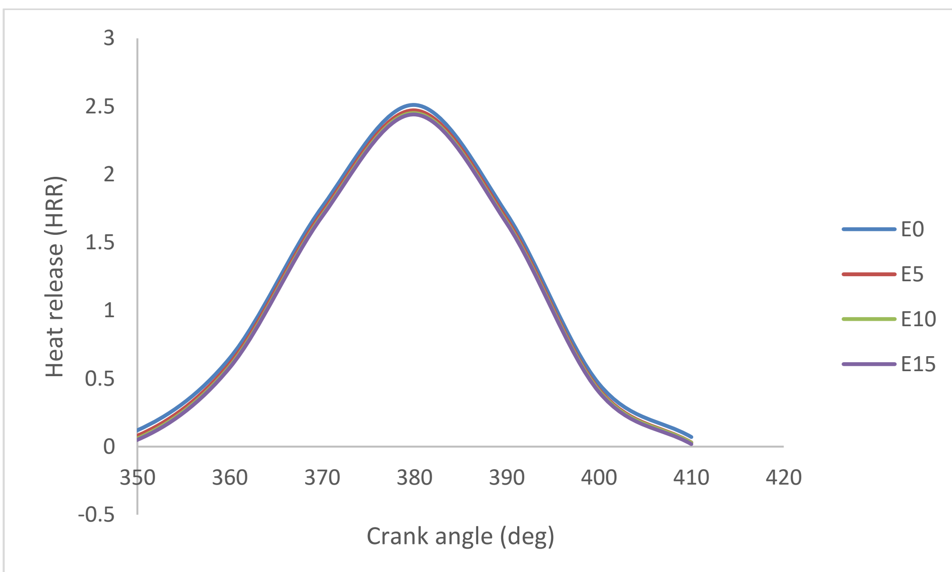

Figure 4. 6 Heat release vs crank angle

4.3 Performance analysis

The major engine performance parameters considered in this work are: brake power (P_b), brake specific fuel consumption (bsfc) and brake thermal efficiency (η_{bth}). To validate the result of computer simulation (in C++) an experimental investigation which is done on same engine specification was used from literature.

4.3.1 Brake power

Effect of using different quantities of ethanol at various engine load on engine brake power is shown in Figure 4.7. The engine load percentage is a measure of how much air and fuel the engine is using compared to its maximum capacity. A higher engine load percentage means that the engine is working harder and consuming more fuel. A lower engine load percentage means that the engine is running more efficiently and using less fuel. Engine load is the ratio of engine brake power (P_b) and the maximum brake power of the engine which 66 kW at 4000 rpm. The engine load percentage is dimensionless and is expressed as a percentage (%). The general behavior of the graph comparing brake power and engine load for pure diesel and an ethanol-diesel blend is remains the same, in that as engine load increases, brake power also increases for the pure diesel fuel (E0) and a slight drop is observed for blend fuels E5, E10 and E15 as shown in Figure 4.7. The major reason why power tends to drop is that due to reduced heating value (Q_{LHV}) of the blend fuels and ethanol has a longer ignition delay than diesel, which can affect the timing of the combustion process and potentially reduce brake power. The graphs show that the brake power increases with the load for all fuels, but the ethanol-diesel blends have lower brake power than pure diesel at the same load. This is because ethanol has lower calorific value (The calorific value of ethanol is the amount of heat released when one unit of ethanol is completely burned) and density than diesel, so it reduces the energy content and mass flow rate of the fuel. This means that less heat and work are produced by the combustion of ethanol-diesel blends than pure diesel.

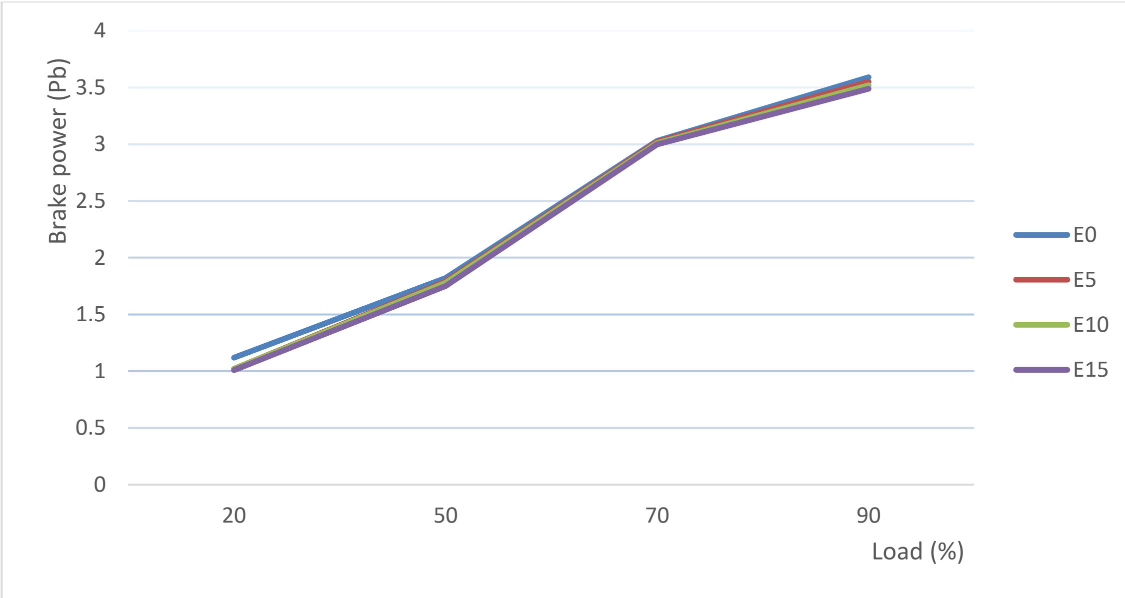

Figure 4. 7 Brake power vs load

Effect of using different quantities of ethanol at various engine speed on engine brake power is shown in Figure 4.8. It can be seen that diesel fuel alone still producing the highest brake power compared to blend fuels. The blend fuels tend to reduce engine power output as the portion of oxygenated compounds (ethanol) in the blends increases. This is due to the lower cetane number and calorific value which increases the ignition delay also found out a slight reduction in maximum power output by using diesel-ethanol blends compared to pure diesel fuel. The shape of the graph is similar for pure diesel and ethanol-diesel blend fuel, but the brake power of ethanol-diesel blend fuel become lower than pure diesel at any speed. This is because ethanol has a lower heating value and a lower density than diesel, which means it produces less energy and less mass of fuel per unit volume. The difference in brake power can increase as the percentage of ethanol in the blend increases. A graph of brake power and speed for pure diesel and ethanol-diesel blend fuel show the effect of using different fuels on the output of the engine. Generally, the graph shows that pure diesel has a higher brake power than ethanol-diesel blend fuel at the same speed, because diesel has a higher heating value and a better combustion efficiency than ethanol. However, the graph follows the same pattern under the same conditions and the results are slightly different.

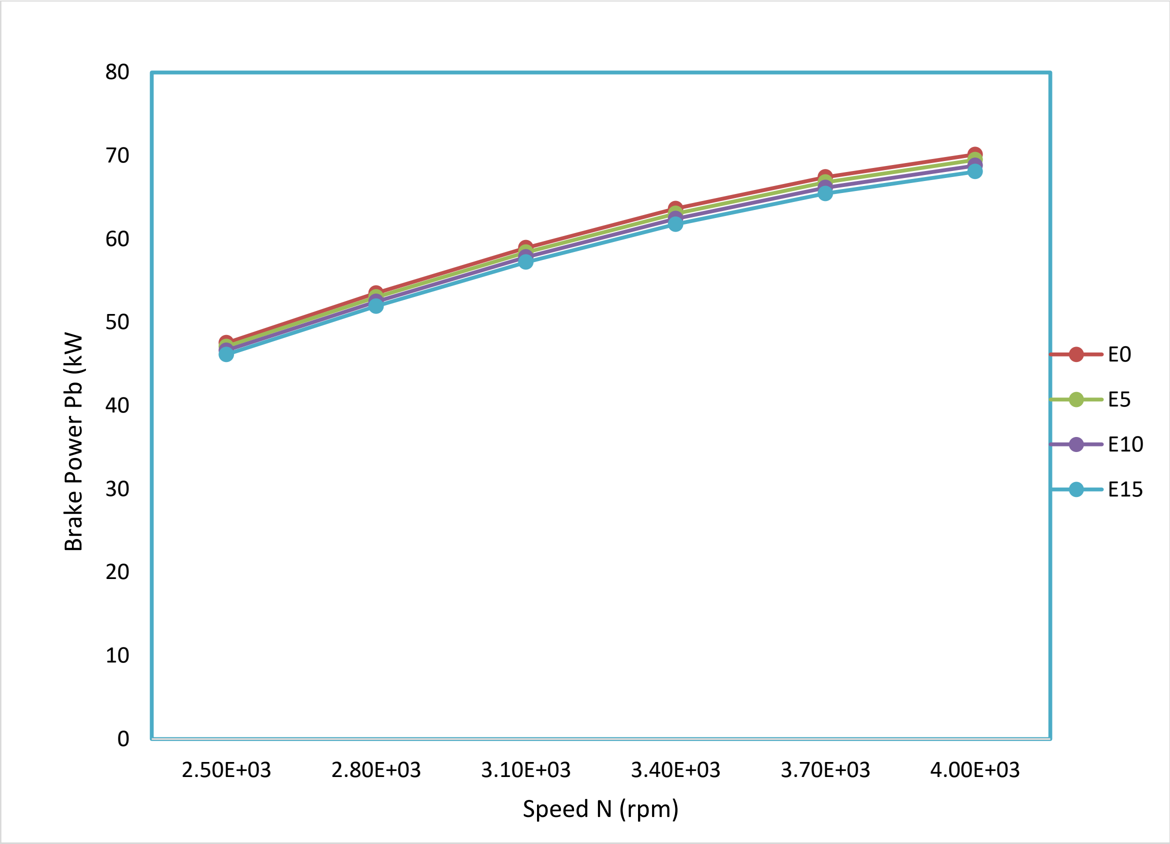

Figure 4. 8 Brake Power vs Engine Speed

The following factors are the cause the curve for an ethanol-diesel blend to be slightly lower than that for pure diesel. Lower energy content: Ethanol has a lower energy content than diesel, so an engine running on an ethanol-diesel blend might produce less brake power compared to one running on pure diesel. Higher latent Heat of vaporization: Ethanol absorbs more heat when changing from liquid to gas, which can reduce the temperature of the air-fuel mixture and cause a decrease in brake power. lower cetane number: Ethanol has a longer ignition delay than diesel, which can affect the timing of the combustion process and potentially reduce brake power.

4.3.2 Brake specific fuel consumption

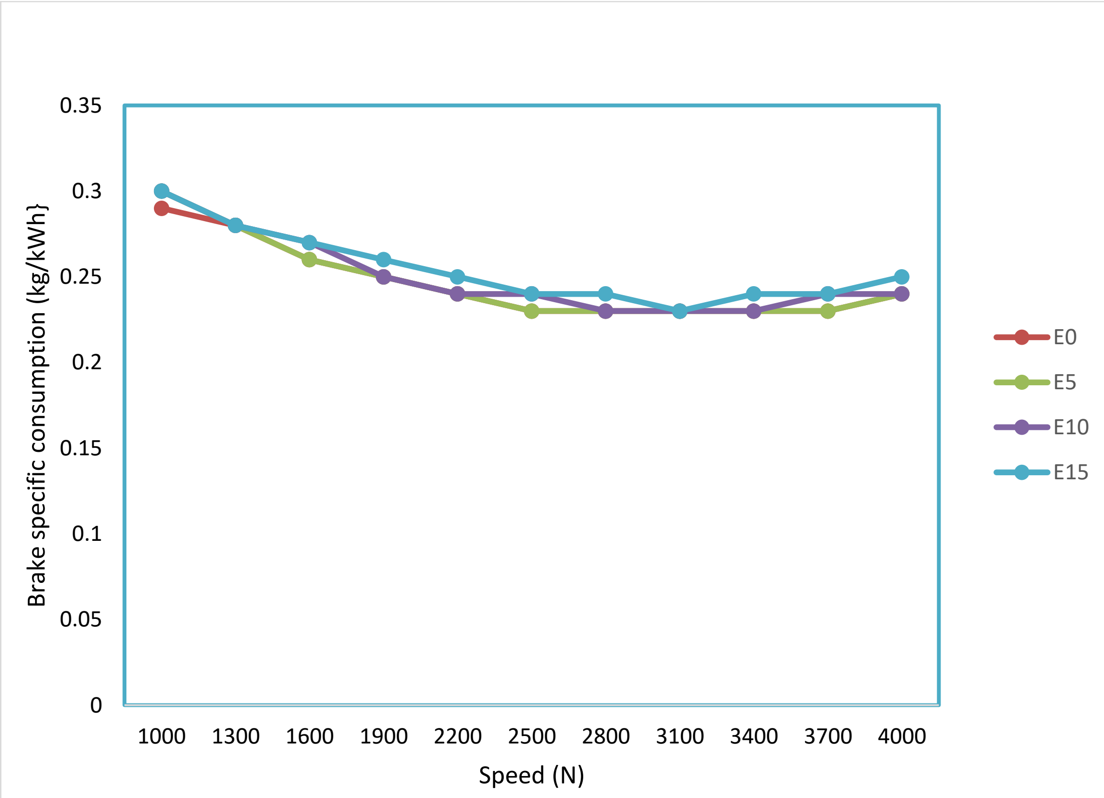

Figure 4. 9 Brake Specific Fuel Consumption vs Engine Speed

The variation of bsfc with engine speed for various ethanol-diesel blend fuels is shown in Figure 4.9. This is a measure of the fuel efficiency of the engine at a given speed. It is defined as the amount of fuel consumed (in kg) per unit of power produced (in kW). Lower values of BSFC indicate higher fuel efficiency. The above graph allows us to compare the performance of pure diesel and ethanol-blend fuels by comparing the BSFC values at different speeds, we can determine which fuel provides better fuel efficiency. As shown in the graph pure diesel has lower BSFC (i.e., higher fuel efficiency) compared to ethanol-blend fuels due to its higher energy content. The graph also shows how the engine speed affects the BSFC for each type of fuel. This provide insights into the optimal operating conditions for the engine i.e. identifying the optimal speed range. The graph shows the range of the optimal speed BSFC is at its lowest, indicating the most efficient use of fuel. In addition, the above graph shows which fuel provides better efficiency at different speeds. As seen in the graph the pure diesel provide better efficiency at lower speeds, while an ethanol-blend perform better at higher speeds. As the speed increases, the BSFC for both pure diesel and ethanol-diesel fuel tends to decrease because the engine operates more efficiently at higher speeds. However, the BSFC for ethanol-diesel fuel be slightly lower than that of pure diesel at the same speed due to the higher oxygen content of ethanol.

The effect of ethanol on brake specific fuel consumption ($bsfc$) is as shown in Figure 4.9. In the range of engine speed 1000 rpm to 4000 rpm, the minimum value of $bsfc$ is 0.361 kg/kWh at 2900 rpm for the pure diesel fuel (E0). Whereas, the $bsfc$ of blend fuels show a slight increment due to power drop and lower heating value of the blend fuels. This is may be more fuel was consumed due to the lower heating value of ethanol. In the other hand, fueled with diesel and the ethanol blends with the variation in engine load shown in Figure 4.10. It is observed that reduction of brake specific fuel consumption at high load for all fuel. At higher loads, in cylinder temperature becomes higher and this helps in proper mixing and atomization which results higher combustion efficiency. The diesel-ethanol blends show a slight reduction in brake specific fuel consumption as compared to that of pure diesel, this is due to adequate oxygen is present in the cylinder that oxidizes the fuel completely resulting in reduction of fuel consumption for generation of similar brake power. Brake specific fuel consumption vs load is shown in Figure 4.10.

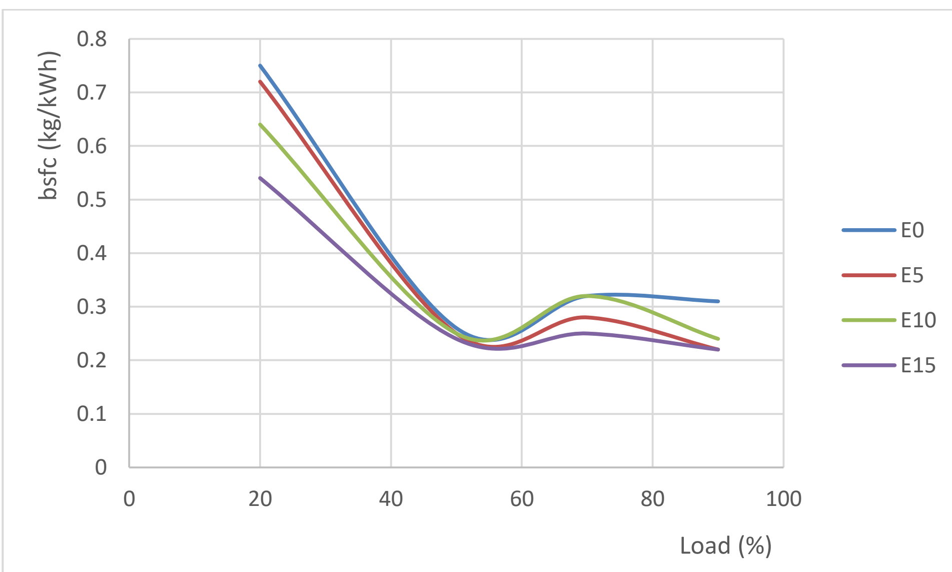

Figure 4. 10 Brake specific fuel consumption vs load

As shown in the graph that as the load increases, the BSFC for both pure diesel and ethanol-diesel fuel tends to decrease because the engine operates more efficiently at higher loads and consumes less fuel per unit of power output. However, the BSFC for ethanol-diesel fuel might be slightly lower than that of pure diesel at the same load due to the higher oxygen content of ethanol, which can lead leading to more complete combustion and lower BSFC. The graph also shows how the fuel consumption varies with the load applied. For pure diesel, at full load, the maximum BSFC is observed to be 0.28 Kg/kw-hr and 15% ethanol-diesel blend, the BSFC at full load is slightly lower at 0.25 Kg/kw-hr. This indicates a decrease in BSFC compared to pure diesel.

4.3.3 Brake thermal efficiency

Brake thermal efficiency for all fuels shown in the Figure 4.11. From the simulation results, brake thermal efficiency for E5, E10 and E15 ethanol-diesel blended fuel calculated results are plotted in the graph. As shown below brake thermal efficiency for E0, E5, E10 and E15 ethanol-diesel blended fuel generally low at low speed. Due to the lower heating value of blended fuel the highest brake thermal efficiency obtained with E0, as shown in Figure 4.11. this may be due to lower power output of the blended fuel result of lower thermal efficiency. shows the brake thermal efficiency (BTE) result. The main reason for decrease of thermal efficiency with increase in blend ratio is shorter ignition delay which results in earlier start of combustion than for diesel. This increases the compression work as well as heat loss and thus reduces the efficiency of the engine.

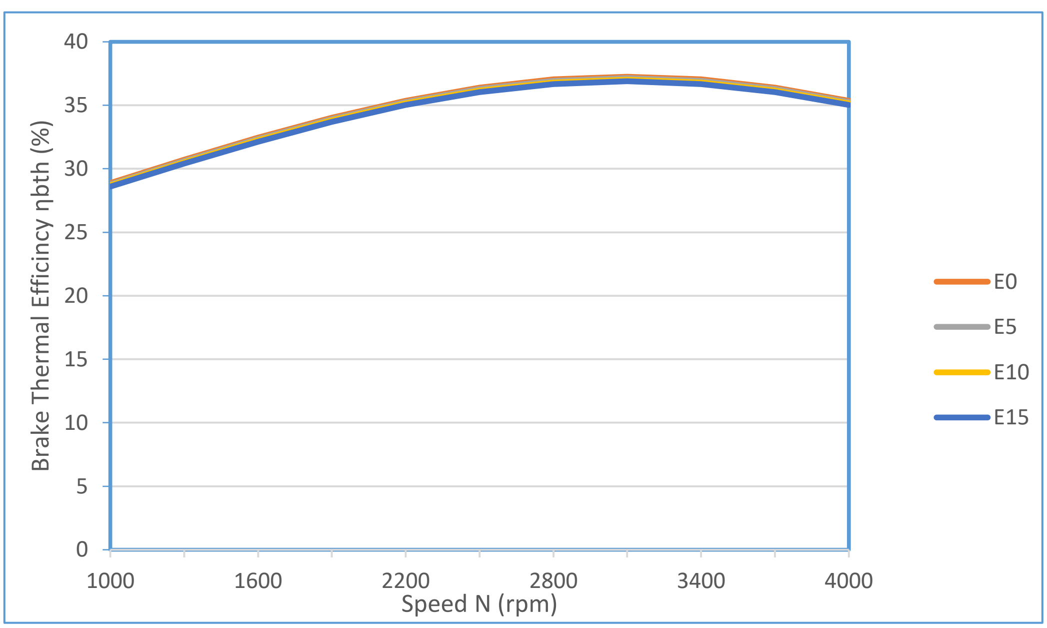

Figure 4. 11 Brake Thermal Efficiency vs Engine Speed for E0, E5, E10 and E15.

4.4 Emission parameters

Ethanol is an oxygenated fuel. If generally oxygenated fuels are used the lower emission levels are observed. Pure diesel fuel can produce a number of harmful emissions, including particulate matter (PM), nitrogen oxides (NOx), and carbon monoxide (CO). When ethanol is blended with diesel, it can help to reduce certain types of emissions. Ethanol is an oxygenated fuel, which means it contains oxygen in its molecular structure. This can lead to more complete combustion in the engine, reducing the emissions of pollutants.

4.4.1 NO emission (NOx)

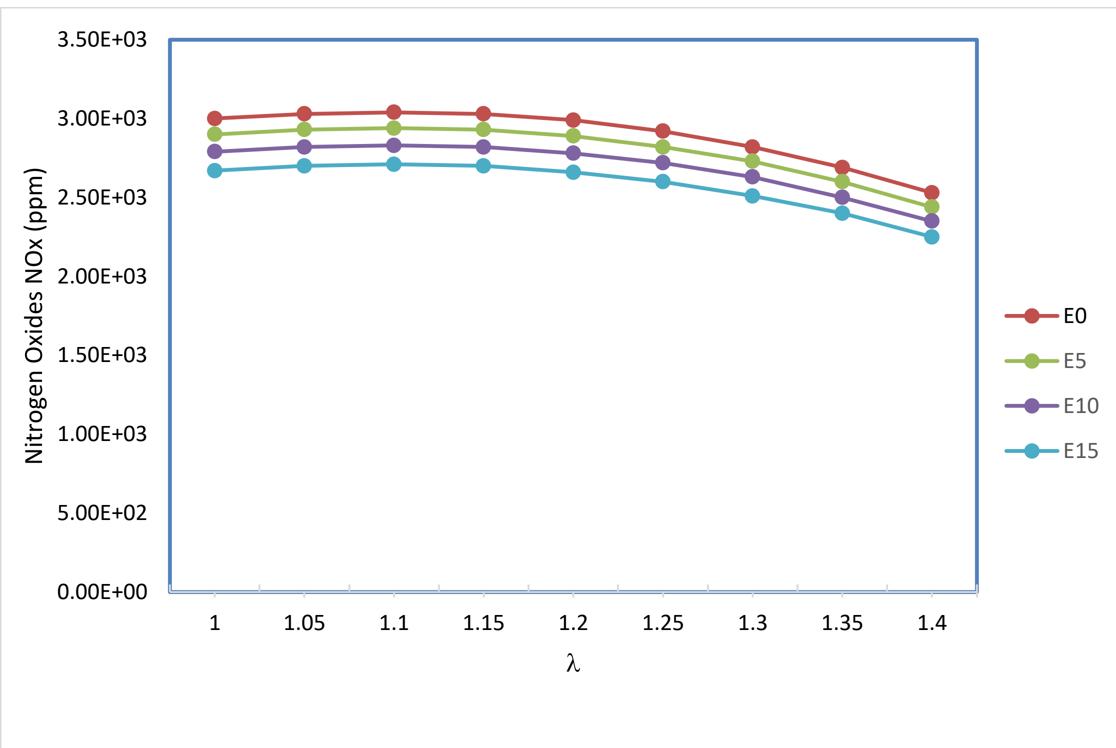

Figure 4. 12 NOx vs λ.

Nitrogen emission of the blends are presented in Figure 4.12. It is observed that there was a slight reduction of Oxide of Nitrogen for all blends in comparison with diesel. Nitrogen Oxides (NOx) emissions are a significant concern in the combustion of fuels.

When comparing pure diesel and ethanol-diesel fuel, there are noticeable differences in NOx emissions. Pure diesel fuel has a certain level of NOx emissions. However, when ethanol is blended with diesel, the NOx emissions reduce to 93.5% of the level of pure diesel fuel. This reduction in NOx emissions is a significant advantage of using ethanol-diesel blended fuel. The lower NOx emission from ethanol added fuel may be its cooling effect of evaporation, which leads to a reduced flame temperature as the NOx emission highly dependent on cylinder temperature. The NOx emission of the 15% ethanol addition is lower than that of diesel fuels.

4.4.2 PM emission

Particulate matter emission for all blends presented in Figure 4.13. The particulate emissions were consistently reduced with increasing quantity of oxygenated fuel. Oxygenated fuel is nothing more than fuel that has a chemical compound containing oxygen. It is used to help fuel burn more efficiently and cut down on some types of atmospheric pollution. It was observed that there is a slight reduction of PM for all blends in comparison with neat diesel fuel. The particulate matter level was decreased with the addition of ethanol for higher air-fuel equivalence ratio. The addition of ethanol to diesel fuel naturally reduces the amount of PM in the fuel. Therefore, lower particulate matter emissions are expected with ethanol–diesel fuels.

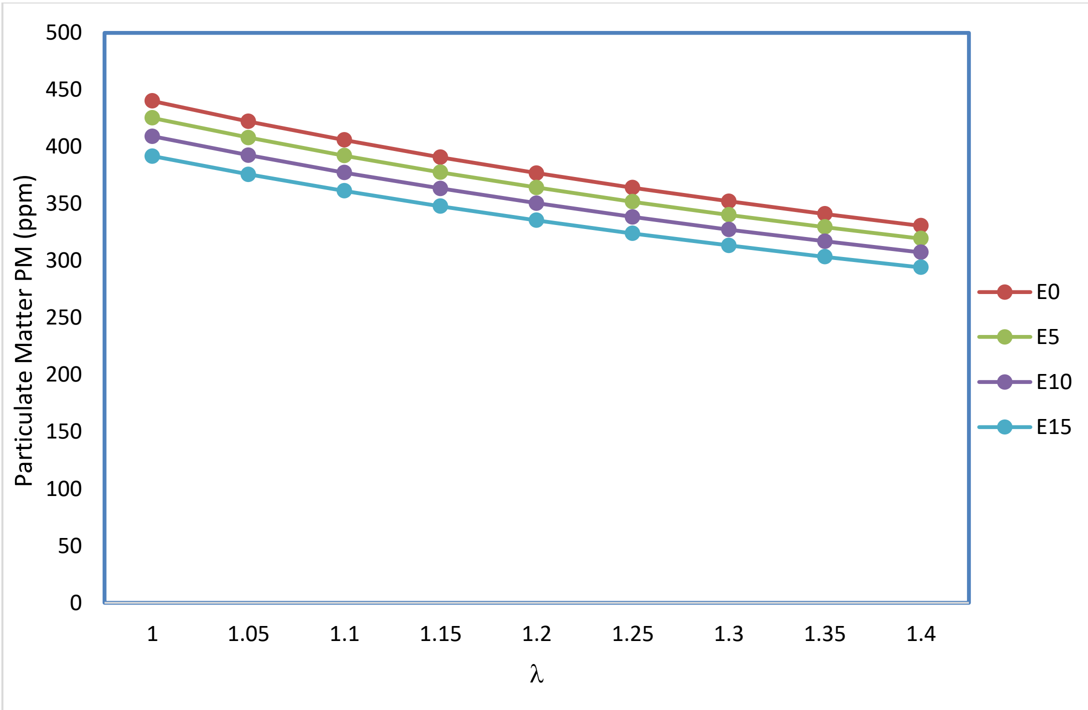

Figure 4. 13 PM vs λ.

4.4.3. Carbon Monoxide (CO)

CO emission is a toxic, colorless and odorless gas. The fuel-rich combustion is the reason behind the formation of CO. The addition of ethanol in diesel fuel tend to increase the oxygen content of the blends which in return helps to increase the oxygen-to-fuel ratio in the fuel rich regions. Thus, complete combustion and hence reduced CO emission. Carbon Monoxide emission for all the blends is presented in Figure 4.14. It is observed that there is a slight decrease of carbon monoxide emission for all blends at full load condition in comparison with neat diesel fuel. The decrease of carbon monoxide of the blends is may be due to addition of ethanol which enhances oxygen proportion and helps to promote the better combustion. Since ethanol has less carbon than diesel fuel and as ethanol is oxygenated alcohol, the CO emissions are generally reduced because of the increased air-fuel ratio and more complete combustion.

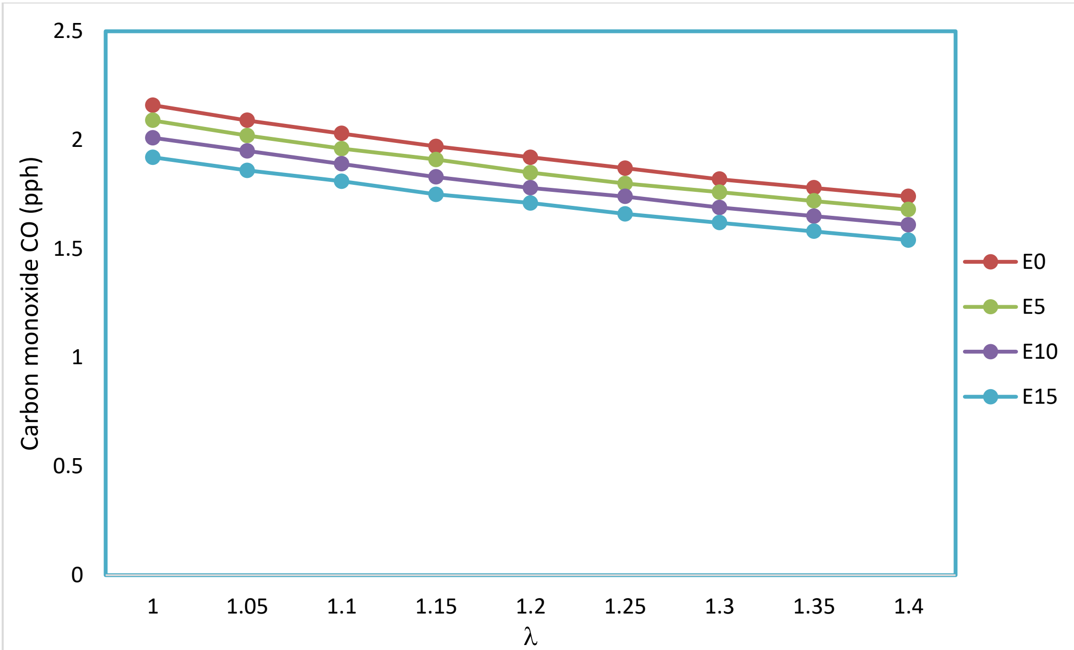

Figure 4. 14 CO vs λ.

4.4.4 Hydrocarbon emission (HC)

In internal combustion engine the hydrocarbon emission is produced by unburned fuels. HC emission reduced with higher oxygen quantity and higher cetane number.

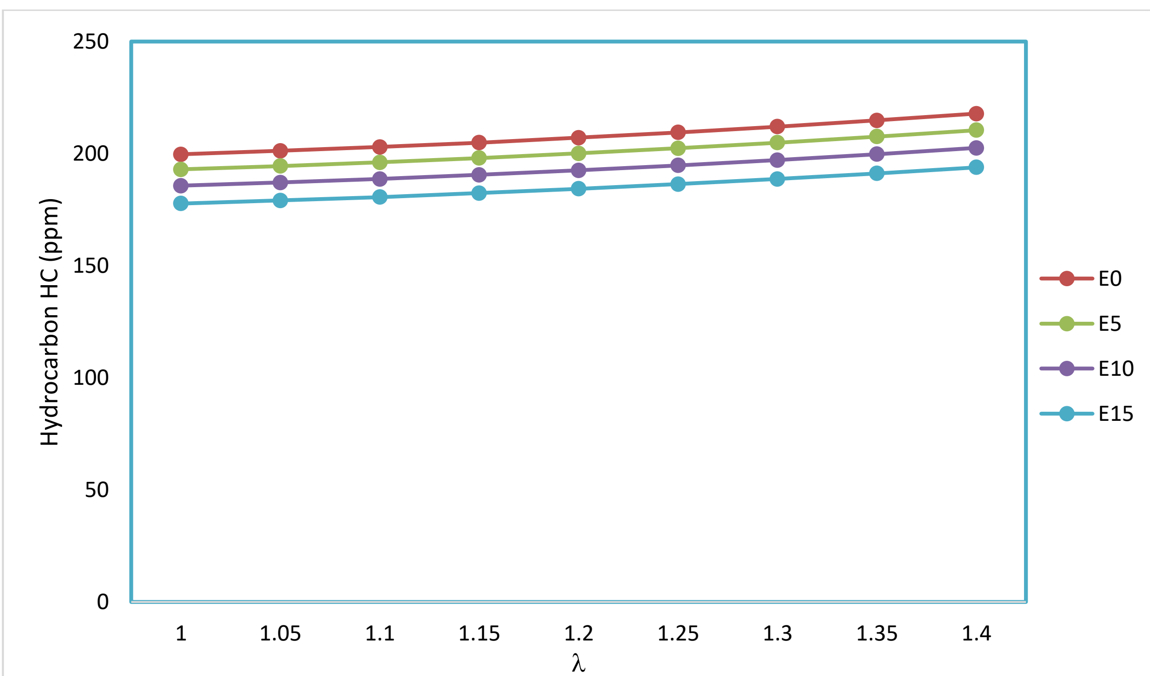

Figure 4. 15 HC vs λ.

The quantity of the oxygen present in the ethanol leads to complete combustion and lower ignition delay decreases the unburned hydrocarbon. Hydrocarbon Emission for all blends are presented in Figure 4.15. It is observed from the graph that HC emission is less for the blends compared to neat diesel fuel. The decrease of HC hydrocarbon emission for all blends in comparison with pure diesel fuel is due to better combustion occurs and more amount of oxygen content present in ethanol. The HC emission of E5, E10 and E15 are 198.87, 182.24 and 154.35 respectively are less than pure diesel fuel by 2.6%,10.1% and 24% respectively.

4.5 Validation of the program

To validate the output of C++ computer program, engine specification was taken from previously investigated engine which study the effect of ethanol-diesel (E5, E10 and E15) in CI engine. In addition to engine specifications both diesel and ethanol physical and chemical characteristics was collected from the literature (Simeon et al., 2019).

Table 4. 3 Table engine specification

Fuel	Diesel
Engine type	Single cylinder
Piston diameter (mm)	87.5
Piston stroke (mm)	110
Volume (cc)	661
Power (kW)	3.5
Maximum rotation speed (rpm)	1500
Compression ratio	17.5
Opening suction valve before TDC (°)	4.5
Closing suction valve after BDC (°)	35.5
Opening exhaust valve before TDC (°)	35.5
Closing exhaust valve after BDC (°)	4.5
Diesel fuel	$C_{10}H_{15}$
Ethanol	C_2H_5OH
Density at 20 ^oC Diesel (kg/m^3)	0.83
Density at 20 ^oC Ethanol (kg/m^3)	0.78

4.5.1 Cylinder pressure

From the results obtained from literature, it can be seen that the maximum pressure for pure diesel fuel is 7.96 MPa, E5 is 7.8 MPa, E10 is 7.56 MPa and f E15 is 7.15 MPa. The result from the simulation for pure diesel fuel is 8.15MPa, E5 is 7.98MPa, E10 is 7.52 MPa and f E15 is 7.48 MPa. The simulation results showed a bit higher value than the actual output from the engine from literature. This may be due to assumptions that considered duiring the simulation process.

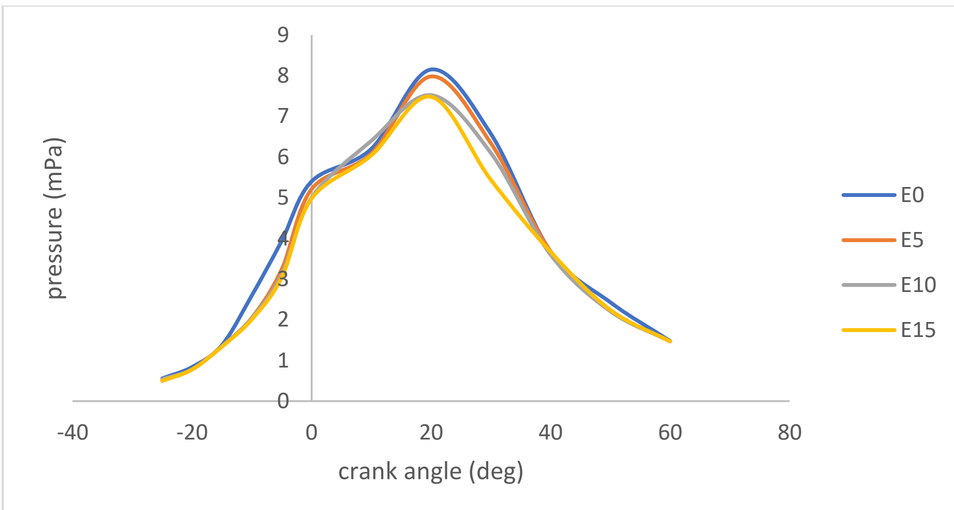

Figure 4. 16 Simulated pressure vs crank angle

4.5.2 NO_x emission

Figure shows the variation of NOx emissions depending on the ethanol percentage in the diesel fuel. As the ethanol percentage increases, NOx decreases due to the low latent heat of evaporation of the mixture. Another possible cause is the formation of a lean mixture which is ethanol as oxygenated fuel improved the combustion process, which leads to a decrease in the combustion temperature. The result from simulation and experimental had the same nature of curve as shown in Figure 4.17 and Figure 4.18.

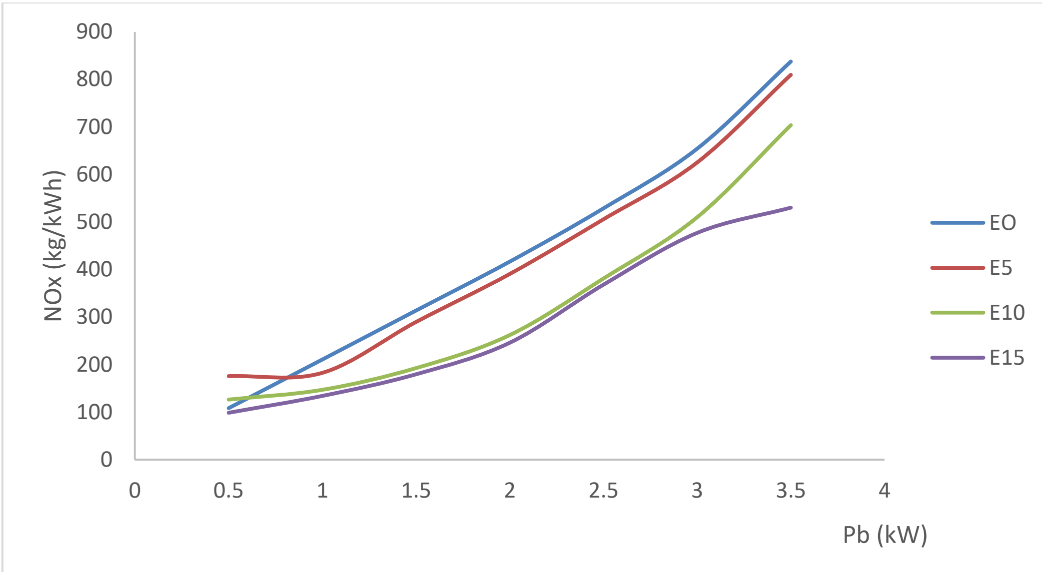

Figure 4. 17 NOx vs brake power simulated

Figure 4.18 shows NOx emission result from the literature, investigated engine which study the effect of ethanol-diesel (E5, E10 and E15) in CI engine (Simeon et al., 2019).

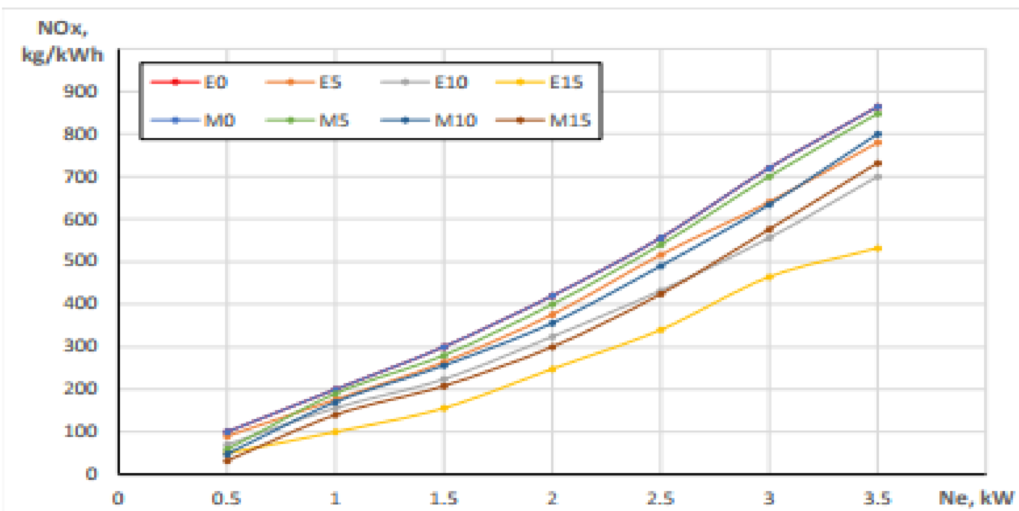

Figure 4. 18 NOx vs brake power GT power

4.5.3 Brake specific fuel consumption

The simulated and the experimental results for the Brake specific fuel consumption for the pure diesel and blended fuel follow a similar trend. As shown below the results gets from the simulation are similar with that of result get from GT power software. Some deviations are expected because of assumption taken for simulation.

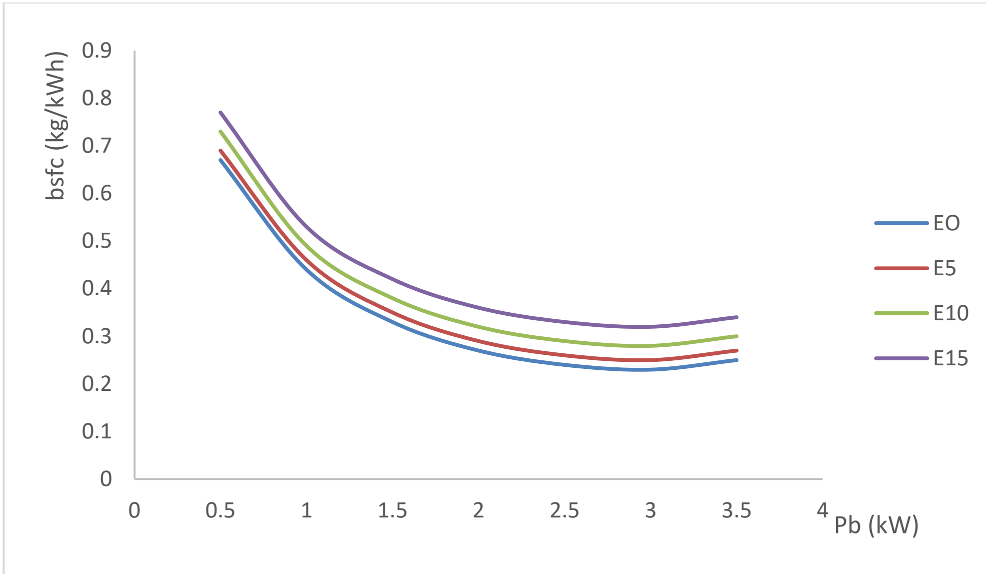

Figure 4. 19 bsfc vs brake power simulated

Figure 4.20 shows the result of bsfc from GT power software, as per the result its observed that bsfc curve shows the slight reduction for pure diesel fuel (Simeon et al., 2019).

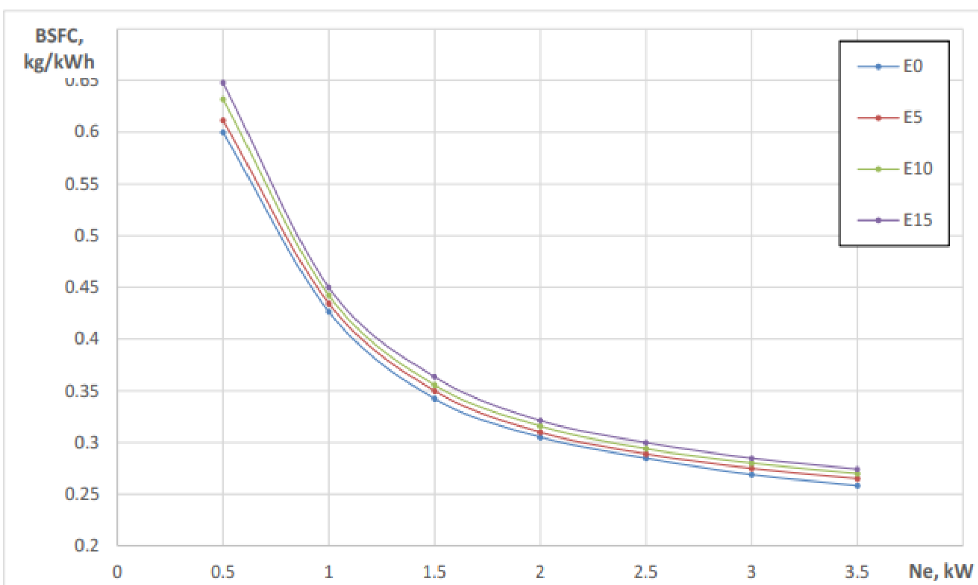

Figure 4. 20 bsfc vs brake power GT software

CHAPTER FIVE

CONCULUSIONS AND RECOMMENDATIONS

5.1 Summary

The goal of this thesis was simulation of the performance of CI engine fueled with diesel-ethanol blends using zero-dimensional combustion model. Primarily, mathematical thermodynamic analysis was carried out using energy equation based on the first law of thermodynamics. A mathematical model is prepared using thermodynamic analysis of actual diesel engine cycle and a program in C++ was developed to plot various important curves. The major performance parameters (brake power, brake specific fuel consumption); combustion parameters (in-cylinder gas pressure, combustion efficiency) and exhaust emission (CO, HC, NOx, PM) analyzed using MS Excel.

At last validation of C++ program done using previously investigated engine specification from the literature using GT power software and the result showed that combustion and emission parameter gained by C++ program is proper and in line with the one which is investigated using GT power software. According to the simulation investigation the emissions of ethanol-diesel fuel blends are lower as compare to that of pure diesel also, Engine power slightly decreased with increasing volumetric ethanol ratio in diesel fuel.

5.2 Conclusions

The properties of ethanol–diesel blends have an important effect on engine performance and emissions. Based on the computational studies in predicting the performance of diesel engine, the following conclusions are drawn:

- ✓ Thermodynamic study was conducted with a four cylinder, naturally aspirated, DI diesel engine to evaluate the effects of using 5 vol% (E5), 10 vol% (E10), 15 vol% (E15) blends of diesel-ethanol blended fuel.
- ✓ The properties of ethanol-diesel blends have a significant effect on engine performance and emissions.

- ✓ Ethanol-diesel blends decreased CO, HC and smoke emissions due to its oxygen content. Blends of up to 10% of ethanol in diesel fuel, ethanol-diesel blend fuel produce certain reductions in regulated diesel emissions.
- ✓ The peak combustion pressure decreased to some extent with the increase of the ethanol oxygen content in the fuel blends up to 10 % with the biggest 3.12 % reduction measured for the fuel blend E5.
- ✓ In this paper conclusion, the simulations performed that the engine break power of pure diesel (E0) is slightly higher than those of E5, E10 and E15. This phenomenon is also clearly presented the rate of heat release. In other hand increasing the rate of brake specific consumption is depending to the lower heating value of ethanol where the ethanol lower heating value lower than diesel fuel
- ✓ It is observed that the slight decrease of HC, NOx, CO and PM emissions shown in the ethanol-diesel fuel blends in comparison with pure diesel.
- ✓ The simulated results showed higher value than the actual output from the engine from literature. This may be due to assumptions taken for simulation.
- ✓ Ethanol is a renewable, domestically produced transportation fuel. This contributes to energy security as it reduces dependence on petroleum.
- ✓ Ethanol production creates jobs in different areas where employment opportunities are needed.
- ✓ The use of ethanol-diesel blend fuel can bring reductions in emission and one of the biggest advantages of ethanol in diesel fuel is that its renewable energy source also, it's economic importance for country like Ethiopia in which fully import diesel fuel. Considering the result from simulation that using ethanol up 10% is used to obtain a homogeneous mixture. Verified ethanol-diesel blends are fully compatible with at least E10, which is a blend of 10% ethanol and 90% pure diesel.

5.3 Recommendations

Now a day's increasing petroleum fuels consumption and increasing engine emissions created an interest to researchers for alternative fuels which are renewable. The idea of alternative fuels in internal combustion engines arises from the need of using fuels which are environmentally safe, energy efficient, and sustainable energy. Therefore, it is recommended to fix the standards of blending concentration in diesel fuel. The country has to consider the usage of bio ethanol in diesel engine to reduce emission and consumption of diesel fuel.

5.4 Future scope of the work

The study of diesel-ethanol blend fuel has become a large discipline over the course of time and expense. There is a need to further study to improve the performance and emission characteristics of diesel-ethanol blend fuel using various types of additives. Experimental methods should also be used to compare and validate the simulation result. The thesis can be extended by using a complex mathematical modeling that consider emission products of all compositions. Further research can be done on performance of the engine by varying the injection timing, compression ratio, EGR etc.

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APPENDIX A: Engine - Actual Engine Cycle (CI-PEC)

```
// Engine - Actual Engine Cyle (CI-PEC).
#include<iostream.h>
#include<conio.h>
#include<iomanip.h>
#include<math.h>
#include<fstream.h>
 void main()
 {
 clrscr();
 char ch;
 int Fuel;
// Engine Size and Basic Dimensions of CGM:
 float
Ec_cc,Zs,Zc,Rsd,Rrl,pim,a,b,QLHV,RHO_Fkgplit,RHO_Alkgplit,LAMBDA;
 float
ETAvmax_per,p1i,T1i,Rc,SIT,ETAcmax_per,ETAm_per,Pbmax,N_Pbmax,xres;
 float
N,ETAbth_per,pcc,hcv,TW,Tim,xres_per,Z_rpc,n_cps,ETAvmax,EFin_Pb;
 float
ETAcmax,ETAc,ETAm,ETAbth,ETAc_per,RHOF,T_STP,pA,TA,ZA,cpA,cvA;
 float
RA,nA,RHOA,vA,Ec_liter,Ec,Vs,Vcl,Vt,Vs_cc,Vcl_cc,Vt_cc,powd,d;
 float
d_mm,s,s_mm,lp_mm,cd_mm,lcl_mm,dp_mm,Ap,lcyl_mm,tcyl_mm,tcylH_mm;
 float
r,r_mm,lcon,lcon_mm,tcon_mm,Hcon_mm,H1con_mm,H2con_mm,dpp_mm,dcp_mm;
 float
dmj_mm,Ru,g,nFuel,nAir,mm1,ETAith_per,Pi1,yE_per,QLHV_D,QLHV_Al;
 float theta_sc,theta_scbTDC,theta_ec,theta_dc,theta_4,theta_6;
// Engine Performance:
 float cpim,cvim,Rim;
 float
Tima,Zim,RHOim,x,OMEGA,Pb,Tb,EFin_kW,EFin_kJ,EFin1,EFin_Pbmax;
 float
mF_Pbmaxkgps,bsfc_Pbmax,bsfc,bmep,imep,ETAv,ETAv_per,ETAv_Mmax;
 float ETAv_Mmin,ZMmin,ZMmax,mF_kgSph,mA_kgph,mED,mm1_kg;
 float Cpm,ETAv_Mmin2;
 float ZMmin1,ZMmin2,nres1;
 float mF_kgps,mA_kgps,mF_kg,mA_kg,mF1_kg,mA1_kg,xA,xF,ZM;
 float div,dp_iv,liv_max,dev,dp_ev,lev_max,RETAv,ZMdiff;
 float Kfl_ev,mF1;
 float
Qin1_kW,Pb1,Qout1_kW,Tmin,Tmax,ETAcarn,fmepa,fmepb,fmep,MmAth,MmFth;
 float
ZC,ZH,ZO,nE,nD,MmD,MmE,yAl,yD,xAl,mE,xD,mG,nED,MmFED,AFthED,AFED,yED
;
 float theta_ivobTDC,theta_ivcaBDC,theta_evobBDC,theta_evcaTDC,RHOa;
```

```
 float
va,cva,theta_ivc,theta_evo,theta_ivo,theta_evc,theta_b,theta_c;
 float theta_d,theta_e,theta_f,ETAv_Mmin1,Pi1_kW;
 float Wnet1,ETAith1,ETAith1_per,mF11,Qin_t,Pb_t,Pb1_kW;
 float
EFin1_kW,EFin_t,Pi_t,Qout_t,ETAith,bsfc_t,mFbb_kgps,mFbb_kgph,nDAl;
 float yAl_per,RHO_Dkgplit,RHO_D,RHO_Al,MmAl,nAl,nDAlmDAl,mAl,mDAl;
 float nF_pkmol,QLHV_DAl,RHO_DAL,Civ_Pbmax,Kv_iv,Kv_ivPbmax,mF_kgph;
 float sumCiv11,sumi_Civ11,Kl_IS,Cg_ivPbmax,Tcd_m,cvcd_m;
 float ETAith_carn,ETAith_carnper,bmep_t,OPD;
// mixture composition:
 float yO2,yN2,MmO2,MmN2,MmA,MmF,n1th,n2th,mAth,mFth,AFth,AF,nF,n2;
 float ath,n3,n1,n4,n5,nO2,nN2,nr,np,yF,yA,y1,y2,y3,y4,y5,Mmr,athED;
 float FAth,FA,n1ED,n2ED;
 float Rmr,MmO,MmN,MmC,MmH,yres,Q_LHVD,Q_LHVE,Q_LHVED,xE_per,n3ED;
// Thermodynamic Analysis:
 float d1,s1,l,p1,T1,theta,rad,Vsc,Vc,V,p2,nc,t2,rr,eff,t3,p3;
 float p4,t4,ne,p5,t5;
// process a-1 (polytropic intake process):
 float theta_a,paa,Taa,maa,thetaa,delpa1,dpa1;
 float deltheta,pa1i,Ta1i,pa1,dxa1a,dxpa1,liv_m,cp_m,civ_m;
 float thetaa1,rada1,xpa1,Vsa1,Va1;
 float dxpa1a,Cpa1,dVa1,delpa1rad,dUa1;
 float
theta_cama1,theta_cama1i,rad_cama1,a1O2r,a2O2r,a3O2r,a4O2r,cpO2,cvaa
;
 float a1N2r,a2N2r,a3N2r,a4N2r,cpN2,cpAa1,cvAa1,cpa1,cva1,Ua1;
 float Ra1,na1,RHOa1,va1,Za1,pra1,Tra1,Acva1,Aiv_m,Aiv_max,pa1_pa;
 float mia1,mia1_kgps,dma1,Cia1,Cia1sqr,cpia1,Tia1,hia1,Raa;
 float
dmhiCia1,cvTdma1,fodpa1ia,pmcvTa1,pmdma1,pVdVa1,fodpa1i,V1,Ta1res;
 float m1,theta1,m11,m22,mdd,m66,Rpa1,na1Tp,mAF1,xres1,pres1,Tres1;
 float
rad6b2,Qa1,Wa1,Q11,W11,Qin1a1,dQa1_W,Ua1i,Qa1i,Wa1i,dQa11,dQa12,Zaa;
 float
delpaa1,Rna1Tp,Civ,CivcpTim,Rpa1a,vaa,na1inv,Civsqr,Cv_iv,EFina1;
 float
sumCiv,sumi_Civ,Civi,i_Civ,dQa1i,dWa1i,dWa11,dWa1_other,dWa12;
 float Cg_iv,Cg_ivsqr,delpa1a,RTa1,Tres,Tresa1,Kv_ivinv,Cg_miv;
 float Cg_mivsqr;
// process 0-1 (isobaric intake process):
 float p0,T0,theta01,rad01,x01,Vs01,Vt01,p01,T01,Vt1_cc,t1;
// process 1-b (polytropic intake-compression process):
 float p11,T11,mi1b,m1b,t1b,RV1b;
 float mi1b_kgps,dx1ba,delp1b,delp1brad,p1b,V1b,T1b,Vb,pb;
 float mb,tb,pbb,Tbb,mbb,Q1b,W1b,Qin11b,mFbb,mFb,dQ1b_W,Rn1bTV;
 float
V1bi,p1bi,T1bi,m1bi,theta1b,rad1b,V11,U1bi,Q1bi,W1bi,U1b,p1b_pa;
 float xp1b,Vs1b,dV1b,dQ1b1,dQ1b2;
 float cpA1b,cvA1b,cp1b,cv1b,n1b;
 float pr1b,Tr1b,Acv1b,Ci1b,dQ1b,Ci1bsqr,dW1b,EFin11;
 float cpi1b,Ti1b,dmhiCi1b,cvTdm1b,fodp1bia,pmcvT1b,pmdm1b,pVdV1b;
```

```
 float fodp1bi,thetab,Rn1bTp,Rp1b,T1bres,EFin1b,EFinbb,Tres1b;
 float dQ1bi,dW1bi,dW1b1,dW1b_other,dW1b2,cv11,R11,Z11,delp1ba,RT1b;
// process b-c (polytropic compression process).
 float xpbc,Vsbc,Vbc,VbVbc,nbc,pbc,mbc,Rbc,Tbc,nbcT;
 float pc,Tc,mc,Wbc,Wcc,Qbc_m,Wbc_m,Zibc;
// process c-2 (polytropic ignition-combustion process):
 float Vc2i,pc2i,Tc2i,mc2i,pc2,Tc2,xpc2,Vsc2,Vc2,mc2,cvc2;
 float Qinc2,Q1c2,Qc2,Wc2_d,Wc2,delUc2,QWc2,nrc2,npc2;
 float m2,W22,Qin1c2;
// process 2-d (polytropic combustion process):
 float V2di,p2di,T2di,m2di,p2d,T2d,xp2d,Vs2d,V2d,m2d,cv2d;
 float Qin2d,Q12d,Q2d,W2d_d,W2d,delU2d,QW2d,Vd;
 float pd,Td,md,nr2d,np2d,Wdd,Qin12d;
// process d-4 (polytropic expansion process):
 float Vd4i,pd4i,Td4i,md4i,Td4,xpd4,Vsd4,Vd4,cpd4,cvd4,Rd4,nd4,RVd4;
 float pd4,md4,nd4T,V4,T4,m4,Wd4,W44,Qd4_m,Wd4_m;
// process 4-5 (polytropic exhaust-blowdown process):
 float p45i,T45i,me45i,cv45i,R45i,T45,m45,dQ45,dW45,theta45;
 float rad45,xp45,Vs45,V45,beta_cam45;
 float t45,R45,n45,RHO45,pr45,Tr45;
 float Acv45,me45,dme45,nV45,npV45,nm45,Aexv;
 float
me45_kgps,dp45,p45,dT45a,dT45,dQ45_W,dQ451,dQ452,dW451,p45_pa,ZMa;
 float
dW45_other,dW452,Q45,W45,V5,T5,m5,theta5,Q55,W55,ZMainv,dp45_pa;
 float
Aev_m,AevOMG45,RT45,RTsqrt45,pRT45,Rn45,Rpem45,Rpem45powa,m44i;
 float Rpem45powb,Cexg_m,Cg_evPbmax,Kv_ev,V44,p44,T44,Cg_mev;
 float Cg_mevsqr,delp45a,delp45,Rp45,Rn45Tp;
// process 5-6 (polytropic exhaust-displacement process):
 float p55,T55,theta56,rad56;
 float xp56,Vs56,V56,dV56,dxp56a;
 float dxp56,theta_cam56,theta_cam,rad_cam56;
 float
cpA56,cvA56,cp56,cv56,R56,n56,RHO56,v56,Z56,pr56,Tr56,Acv56,Cp56;
 float me56,delp56a,delp56,p56,Rp56,Rn56Tp,he56;
 float dU56,Q56,W56,V6,p6,T6;
 float theta6,dVs56,dme56,y55,yHC_max,LAMBDA2;
// emission
 float
yNOxmax,yCO_max,LAMBDA_yNOxmax,yNOx,yCO,DELLAM,yCO1,y5_pph,yNO_MF;
 float yPM_MF,yHC_MF,ac2,mC,mpow,expc2,a2d,exp2d,x2,yPM,yHC,yPM_max;
// process c-d (polytropic combustion process).
 float Vcdi,pcdi,Tcdi,pcd,Tcd,Vcd,mcd,mFcd,cvcd,Rcd,ncd,Zcd;
 float thetascdc,xb,Qincd,Q1,As,Q2,Qcd,Wcd_other,Wcd_d,Wcd;
 float QWcd,td,T23,To,V2,p22,T22,T2,Z2,Z23,cpO2o2,cpN2o2,cpFo2;
 float cpr2,cvr2,Mmr2,cp2,cv2,R2,RHO2,v2,hfoF,hfoO2,hfoN2,delhFo2;
 float
delhO2o2,delhN2o2,UrFo2,UrO2o2,UrN2o2,Ur2,Ur,a1CO2p,a2CO2p,a3CO2p;
 float
a4CO2p,cpCO2,cvCO2,a1H2Op,a2H2Op,a3H2Op,a4H2Op,cpH2O,cvH2O,cp223;
```

```
 float
a1N2p,a2N2p,a3N2p,a4N2p,cp323,a1O2p,a2O2p,a3O2p,a4O2p,cp123,Qin1;
 float cp423,a1COp,a2COp,a3COp,a4COp,cpCO,cvCO,cp523,cp23_kmol;
 float cv23_kmol,Mm23,n23,cp23,cv23,W23,delh1o23,delh2o23,delh3o23;
 float
delh5o23,hfo1,hfo2,hfo3,hfo4,hfo5,U123,U223,U323,U423,U523,U23,Up;
 float Qin23,mF23,nF23,delUpr,error23,T3,mm23,T23m,cvm23;
 float Qmcv23,T33,pmax,p33,Tcc,Qr;
// process c-2 (polytropic ignition process).
// process 2-d (polytropic compression process).
// process d-e (polytropic expansion process):
 float pdd,Tdd,mde,Vde,nde,pde,Tde,pe,Te,me;
// process e-5 (polytropic exhaust-blowdown process):
 float pee,Tee,mee,Ve5,delthetae5,delrade5,thetae52,rade52,mee5,te5;
 float mee5_kgps,dxe5a,dxpe5,lev_m,cev_m,delpe5,Re5,Ze5,delpe5rad;
 float pe5,Te5,RHOe5,ne5,ne5pow;
// process 5-f (polytropic exhaust-displacement process):
 float V5f,deltheta5f,delrad5f,theta5f2,rad5f2,me5f,m5f,t5f,Z5f;
 float
me5f_kgps,dx5fa,dxp5f,delp5f,R5f,delp5frad,p5f,T5f,pf,Tf,mf,RHO5f;
 float n5f,n5fpow;
// process 4-5 (isochoric 'exhaust-blowdown' process).
 float p00,T00,m00;
// process 5-6 (isobaric 'exhaust-displacement' process).
 float theta60,Vt60,p60,T60,Vt60_cc,p66,T66;
 float N_min,N_max,delN;
 float PI = 3.14; Ru = 8.314; g = 9.81;
 ofstream outfile ("CI-PCE.dat");
 cout << "\n\n\t\t Adama Science and Technology University ";
 outfile << "\n\n\n\n\n\t\t\t Adama Science and Technology
University (ASTU) ";
 cout << " \n\t Department of Mechanical and Vehicle Engineering ";
 outfile << "\n\t\t\t Department of Mechanical and Vehicle
Engineering ";
 cout << " \n\t\t Habtamu Belay (MSc Student) ";
 outfile << " \n\t\t\t Habtamu Belay (MSc Student) ";
 cout << "\n ........................................... \n";
 outfile <<
"\n\t\t\t…............................................. \n";
 cout << "\n\t Effects of Diesel-Ethanol (E5,E10,E15) Blends ";
 cout << "\n\t              on ";
 cout << "\n\t Performance,Combustion and Exhaust Emission of Diesel
Engine ";
 outfile << "\n Effects of Diesel-Ethanol (E5,E10,E15) Blends on ";
 outfile << "\n Performance,Combustion and Exhaust Emission of
Diesel Engine ";
 cout << "\n ----------------------------------------------------------
---------------------- ";
 outfile << "\n ------------------------------------------------------
------------------------- ";
 cout <<"\n\n Enter the 'number of Carbon atoms in the Diesel (a)' =
";
```

```
 cin >> a;
 cout <<"\n Enter the 'number of Hydrogen atoms in the Diesel (b)' =
";
 cin >> b;
 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 'Lower Heating Value of Diesel (QLHV_D)' in
'MJ/kg' = ";
 cin >> QLHV_G;
 cout <<"\n Enter the 'Lower Heating Value of Alcohol (QLHV_AL)' in
'MJ/kg' = ";
 cin >> QLHV_Al;
 cout <<"\n Enter the 'density of Diesel (RHO_D)' in 'kg/liter' = ";
 cin >> RHO_Gkgplit;
 cout <<"\n Enter the 'density of Alcohol (RHO_Al)' in 'kg/liter' =
";
 cin >> RHO_Alkgplit;
 cout <<"\n Enter the 'percentage of Alcohol by volume' = ";
 cin >> yAl_per;
 cout <<"\n Enter the 'Air-Factor (LAMBDA)' = ";
 cin >> LAMBDA;
 cout <<"\n Enter the 'mass fraction of exhaust residual (xres)' =
";
 cin >> xres1;
 cout <<"\n Enter the 'Engine capacity (Ec)' (100 to 3000 cc) = ";
 cin >> Ec_cc;
 cout <<"\n Enter the 'Number of cylinders (Zc)' = ";
 cin >> Zc;
 cout <<"\n Enter the 'stroke-to-bore Ratio (Rsd)' (0.8 to 1.2) = ";
 cin >> Rsd;
 cout <<"\n Enter the 'crank-to-conrod length Ratio (Rrl)' (0.2 to
0.3) = ";
 cin >> Rrl;
 cout <<"\n Enter the 'engine Load (pim)' in 'bar' (0.55 to 0.95
bar) = ";
 cin >> pim;
 cout <<"\n Enter the 'compression Ratio (Rc)' = ";
 cin >> Rc;
 cout <<"\n Enter the 'maximum combustion Efficiency (ETAc)' (80 t0
95 %) = ";
 cin >> ETAcmax_per;
 cout <<"\n Enter the 'maximum brake Power (Pbmax) in 'kW' = ";
 cin >> Pbmax;
 cout <<"\n Enter the 'engine Speed at Pbmax (N_Pbmax)' in 'rpm' =
";
 cin >> N_Pbmax;
 cout <<"\n Enter the 'required engine Speed (N)' in 'rpm' = ";
```

```
 cin >> N;
 cout <<"\n Enter the 'bsfc at Pbmax (bsfc_Pbmax)' (0.2 to 0.4
kg/kWh) = ";
 cin >> bsfc_Pbmax;
 cout <<"\n Enter 'theta_ivobTDC' in 'deg.' = ";
 cin >> theta_ivobTDC;
 cout <<"\n Enter 'theta_ivcaBDC' in 'deg.' = ";
 cin >> theta_ivcaBDC;
 cout <<"\n Enter 'theta_evobBDC' in 'deg.' = ";
 cin >> theta_evobBDC;
 cout <<"\n Enter 'theta_evcaTDC' in 'deg.' = ";
 cin >> theta_evcaTDC;
 cout <<"\n Enter 'theta_scbTDC' in 'deg.' = ";
 cin >> theta_scbTDC;
 cout <<"\n Enter 'theta_dc' in 'deg.' = ";
 cin >> theta_dc;
 outfile << "\n\n\n Input: ";
 outfile << "\n ---------------- \n";
 outfile <<"\n a = "<<a;
 outfile <<"\n b = "<<b;
 outfile <<"\n ZC = "<<ZC;
 outfile <<"\n ZH = "<<ZH;
 outfile <<"\n ZO = "<<ZO;
 outfile <<"\n QLHV_D = "<<QLHV_D;
 outfile <<"\n QLHV_AL = "<<QLHV_Al;
 outfile <<"\n RHO_D = "<<RHO_Dkgplit;
 outfile <<"\n RHO_Al = "<<RHO_Alkgplit;
 outfile <<"\n yAl_per = "<<yAl_per;
 outfile <<"\n LAMBDA = "<<LAMBDA;
 outfile <<"\n xres = "<<xres1;
 outfile <<"\n Ec = "<<Ec_cc;
 outfile <<"\n Zc = "<<Zc;
 outfile <<"\n Rsd = "<<Rsd;
 outfile <<"\n Rrl = "<<Rrl;
 outfile <<"\n pim = "<<pim;
 outfile <<"\n Rc = "<<Rc;
 outfile <<"\n ETAc = "<<ETAcmax_per;
 outfile <<"\n Pbmax = "<<Pbmax;
 outfile <<"\n N_Pbmax = "<<N_Pbmax;
 outfile <<"\n N = "<<N;
 outfile <<"\n bsfc_Pbmax = "<<bsfc_Pbmax;
 outfile <<"\n theta_ivobTDC = "<<theta_ivobTDC;
 outfile <<"\n theta_ivcaBDC = "<<theta_ivcaBDC;
 outfile <<"\n theta_evobBDC = "<<theta_evobBDC;
 outfile <<"\n theta_evcaTDC = "<<theta_evcaTDC;
 outfile <<"\n theta_scbTDC = "<<theta_scbTDC;
 outfile <<"\n theta_dc = "<<theta_dc;
 Zs = 4.0; pcc = 1.15; hcv = 350.0; TW = 400.0; xres = xres1;
 xres_per = xres*100.0; Z_rpc = Zs/2.0; n_cps = N/(Z_rpc*60.0);
 ETAcmax = ETAcmax_per/100.0; Cg_ivPbmax = 100.0; Cg_evPbmax =
200.0;
 ETAc = ETAcmax*(-1.6+(4.65*LAMBDA)-(2.07*LAMBDA*LAMBDA));
```

```
 ETAc_per = ETAc*100.0; T_STP = 298.0;
 RHO_D = RHO_Dkgplit*1.0e3; RHO_Al = RHO_Alkgplit*1.0e3;
 theta_evc = theta_evcaTDC; theta_a = theta_evc;
 theta_ivc = 180.0+theta_ivcaBDC; theta_b = theta_ivc;
 theta_sc = 360.0-theta_scbTDC; theta_c = theta_sc;
 theta_ec = 360.0+(theta_dc-theta_scbTDC); theta_d = theta_ec;
 theta_evo = 540.0-theta_evobBDC; theta_4 = theta_evo;
 theta_ivo = 720.0-theta_ivobTDC; theta_6 = theta_ivo;
 pA = 1.0; TA = 298.0; ZA = 1.0;
 cpA = 1.005; cvA = 0.718; RA = cpA-cvA; nA = cpA/cvA;
 RHOA = (pA*1.0e5)/(ZA*RA*1.0e3*TA); vA = 1.0/RHOA;
 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; Ap = (PI*d*d)/4.0;
 lp_mm = 0.9*s_mm; cd_mm = 0.1; lcl_mm = 8.0;
 dp_mm = d_mm-cd_mm; lcyl_mm = lcl_mm+s_mm+lp_mm;
 tcyl_mm = 10.0; tcylH_mm = 1.2*tcyl_mm; r = s/2.0; r_mm = r*1.0e3;
 lcon = r/Rrl; lcon_mm = lcon*1.0e3; tcon_mm = 3.0; Hcon_mm =
5.0*tcon_mm;
 H1con_mm = 1.1*Hcon_mm; H2con_mm = 0.9*Hcon_mm; dpp_mm = 0.2*d_mm;
 dcp_mm = 0.4*d_mm; dmj_mm = 0.5*d_mm;
 cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
 getch();
 getch();
 clrscr();
 cout << "\n\n Engine Size: ";
 outfile << "\n\n Engine Size: ";
 cout << "\n ---------------- \n";
 outfile << "\n ---------------- \n";
 cout <<setprecision(3)<<" Ec = "<<Ec_cc<<" cc ";
 cout <<setprecision(2)<<"\n Vs = "<<Vs_cc<<" cc ";
 cout <<setprecision(2)<<"\n Vcl = "<<Vcl_cc<<" cc ";
 cout <<setprecision(2)<<"\n Vt = "<<Vt_cc<<" cc ";
 outfile <<setprecision(2)<<" Ec = "<<Ec_cc<<" cc ";
 outfile <<setprecision(2)<<"\n Vs = "<<Vs_cc<<" cc ";
 outfile <<setprecision(2)<<"\n Vcl = "<<Vcl_cc<<" cc ";
 outfile <<setprecision(2)<<"\n Vt = "<<Vt_cc<<" cc ";
 cout << "\n\n Basic Dimensions of CGM: ";
 outfile << "\n\n Basic Dimensions of CGM: ";
 cout << "\n -------------------------- \n";
 outfile << "\n -------------------------- \n";
 cout <<setprecision(2)<<" d = "<<d_mm<<" mm ";
 cout <<setprecision(2)<<"\n s = "<<s_mm<<" mm ";
 cout <<setprecision(2)<<"\n r = "<<r_mm<<" mm ";
 cout <<setprecision(2)<<"\n l = "<<lcon_mm<<" mm ";
 outfile <<setprecision(2)<<" d = "<<d_mm<<" mm ";
 outfile <<setprecision(2)<<"\n s = "<<s_mm<<" mm ";
 outfile <<setprecision(2)<<"\n r = "<<r_mm<<" mm ";
 outfile <<setprecision(2)<<"\n l = "<<lcon_mm<<" mm ";
 cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
```

```
 getch();
 getch();
 clrscr();
 cout << "\n\n Engine Performance: ";
 outfile << "\n\n Engine Performance: ";
 cout << "\n ------------------------- \n";
 outfile << "\n ------------------------- \n";
 N_min =700.0; N_max = N_Pbmax; delN = (N_max-N_min)/10.0;
 for (N = N_min; N <= N_max; N = N+delN)
 {
 x = N/N_Pbmax; OMEGA = (2.0*PI*N)/60.0;
 Pb = Pbmax*(-(x*x*x)+(1.4*x*x)+(0.6*x));
// Pb = Pbmax*(-(x*x*x)+(1.13*x*x)+(0.87*x));
 Tb = (Pb*1.0e3)/OMEGA;
 bsfc = bsfc_Pbmax*((x*x)-(1.55*x)+1.55);
 cout << setprecision(2)<<"\n\t "<<Pb;
 cout << setprecision(2)<<"\t "<<Tb;
// cout << setprecision(2)<<"\t "<<bmep;
 cout << setprecision(2)<<"\t "<<bsfc;
 cout << setprecision(2)<<"\t "<<N;
 outfile << setprecision(2)<<"\n\t "<<Pb;
 outfile << setprecision(2)<<"\t "<<Tb;
// outfile << setprecision(2)<<"\t "<<bmep;
 outfile << setprecision(2)<<"\t "<<bsfc;
 outfile << setprecision(2)<<"\t "<<N;
 getch();
 }
 MmO = 16.0; MmN = 14.0; MmC = 12.0; MmH = 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) - n1th*CO2+n2th*H2O+n3th*N2
 yO2 = 0.21; yN2 = 0.79; MmO2 = 2.0*MmO; MmN2 = 2.0*MmN;
 yAl = yAl_per/100.0; yD = 1.0-yAl; yF = yAl+yD;
 MmG = (a*MmC)+(b*MmH); MmAl = (ZC*MmC)+(ZH*MmH)+(ZO*MmO);
 MmA = (yO2*MmO2)+(yN2*MmN2); MmF = (yD*MmD)+(yAl*MmAl);
 n1th = (ZC*yAl)+(a*yD); n2th = ((ZH*yAl)+(b*yG))/2.0;
 ath = ((2.0*n1th)+n2th)/2.0;
 MmAth = MmA; MmFth = MmF; nO2 = LAMBDA*ath;
 nN2 = LAMBDA*ath*3.76; nA = nO2+nN2;
 AFth = (nA*MmAth)/MmFth; AF = LAMBDA*AFth;
 cpim = 1.005; cvim = 0.718; Rim = cpim-cvim;
 Zim = 1.0;
 Tim = 323.0; // TA*pow(RpimpA,npTAim);
 RHOim = (pim*1.0e5)/(Zim*Rim*1.0e3*Tim);
 p1i = 1.0; T1i = 323.0;
 x = N/N_Pbmax; n_cps = N/(60.0*Z_rpc); OMEGA = (2.0*PI*N)/60.0;
 Pb = Pbmax*(-(x*x*x)+(1.4*x*x)+(0.6*x));
// Pb = Pbmax*(-(x*x*x)+(1.13*x*x)+(0.87*x));
 Tb = (Pb*1.0e3)/OMEGA;
 bsfc = bsfc_Pbmax*((x*x)-(1.55*x)+1.55);
 bmep = ((Pb*1.0e3)/(Ec*n_cps))*1.0e-5;
```

```
 fmepa = (0.001*N); fmepb = (0.001*N)*(0.001*N);
 fmep = 0.97+(0.15*fmepa)+(0.05*fmepb); imep = bmep+fmep;
 ETAm = bmep/imep; ETAm_per = ETAm*100.0;
 mF_kgph = bsfc*Pb; mA_kgph = mF_kgph*AF;
 mF_kgps = mF_kgph/3600.0; mA_kgps = mA_kgph/3600.0;
 mF_kg = mF_kgps/n_cps; mA_kg = mA_kgps/n_cps;
 mF1_kg = mF_kg/Zc; mA1_kg = mA_kg/Zc;
 mm1_kg = (mA1_kg+mF1_kg)/(1.0-xres);
 Cpm = (2.0*s*N)/60.0;
 ETAv = mA1_kg/(RHOim*Vs); // (bsfc*bmep*AF)/(RHOim*36.0);
 ETAv_per = ETAv*100.0;
 ETAv_Mmin1 = 0.85; ETAv_Mmax = 0.895; ETAv_Mmin2 = 0.65;
 ZMmin1 = 0.4; ZMmax = 0.55; ZMmin2 = 0.95;
 if (ETAv >= 0.85 && ETAv <= 0.875)
 {
 RETAv = (ETAv-ETAv_Mmin1)/(ETAv_Mmax-ETAv_Mmin1);
 ZMdiff = ZMmax-ZMmin1;
 ZM = (RETAv*ZMdiff)+ZMmin1;
 }
 else if (ETAv >= 0.65 && ETAv <= 0.875)
 {
 RETAv = (ETAv-ETAv_Mmin2)/(ETAv_Mmax-ETAv_Mmin2);
 ZMdiff = ZMmax-ZMmin2;
 ZM = (RETAv*ZMdiff)+ZMmin2;
 }
 div = 0.4*d; dp_iv = 0.8*div; liv_max = 0.25*div;
 dev = 0.8*div; dp_ev = 0.8*dev; lev_max = liv_max;
 Aiv_m = PI*dp_iv*(liv_max/2.0); Kv_iv =
(Ap/Aiv_m)*(Cpm/Cg_ivPbmax);
 Aev_m = PI*dp_ev*(lev_max/2.0); Kv_ev =
(Ap/Aev_m)*(Cpm/Cg_evPbmax);
 cout <<setprecision(2)<<"\n\n\t bmep = "<<bmep<<" bar ; ";
 cout <<setprecision(2)<<" ETAv_per = "<<ETAv_per<<" % ";
 cout << "\n\n\n\t\t Press 'enter' twice to proceed ... ";
 getch();
 getch();
 clrscr();
 LAMBDA2 = 0.95;
 MmO = 16.0; MmN = 14.0; MmC = 12.0; MmH = 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) - n1th*CO2+n2th*H2O+n3th*N2
 yO2 = 0.21; yN2 = 0.79; MmO2 = 2.0*MmO; MmN2 = 2.0*MmN;
 yAl = yAl_per/100.0; yD = 1.0-yAl;
 nAl = yAl; nD = yD; nDAl = nD+nAl;
 MmG = (a*MmC)+(b*MmH); MmAl = (ZC*MmC)+(ZH*MmH)+(ZO*MmO);
 MmA = (yO2*MmO2)+(yN2*MmN2); MmF = (yD*MmD)+(yAl*MmAl);
 mAl = nAl*MmAl; mD = nD*MmD; mDAl = mD+mAl;
 xAl = mAl/mDAl; xD = mD/mDAl; nF_pkmol = nDAl;
 n2 = ((ZH*nAl)+(b*nD))/2.0; n4 = 0.0;
 n1 = (LAMBDA2*ath*2.0)+(ZO*nAl)-n2-(2.0*n4)-((ZC*nAl)+(a*nD));
```

```
 n5 = (ZC*yAl)+(a*nD)-n1; n3 = (LAMBDA2*ath*3.76*2.0)/2.0;
 np = n1+n2+n3+n4+n5;
 y1 = n1/np; y2 = n2/np; y3 = n3/np; y4 = n4/np; y5 = n5/np;
 y55 = y5;
 cout << "\n\n Emission Prediction: ";
 outfile << "\n\n Emission Prediction: ";
 cout << "\n --------------------- \n";
 outfile << "\n -------------------- \n";
 cout <<"\n\t LAMBDA\t NOx\t\t PM\t CO\t HC";
 cout <<"\n\t ()\t (ppm)\t\t (ppm)\t (pph)\t (ppm)";
 y5_pph = y55*35.0; yNO_MF = 1400.0; yPM_MF = 200.0;
 yHC_MF = 25.0; yNOxmax = y5_pph*yNO_MF; yPM_max = y5_pph*yPM_MF;
 yCO_max = y5_pph; yHC_max = y5_pph*yHC_MF;
 LAMBDA_yNOxmax = 1.1; DELLAM = 0.05;
 for (LAMBDA = 1.0; LAMBDA <= 2.0; LAMBDA = LAMBDA+DELLAM)
 {
 x = LAMBDA/LAMBDA_yNOxmax;
 yNOx = yNOxmax*(-(x*x*x)+(x*x)+x);
 yPM = yPM_max*pow(x,-0.85);
 yCO = yCO_max*pow(x,-0.65);
 yHC = yHC_max*((x*x)-(1.2*x)+4.2); //((x*x)-(5.0*x)+7.5);
// yCO1 = -(x*0.35); yCO = yCO_max*exp(yCO1);
       cout << setprecision(2)<<"\n\t "<<LAMBDA;
       outfile << setprecision(2)<<"\n\t "<<LAMBDA;
       cout << setprecision(2)<<"\t"<<yNOx;
       outfile << setprecision(2)<<"\t"<<yNOx;
       cout <<setprecision(2)<<"\t"<<yPM;
       outfile <<setprecision(2)<<"\t"<<yPM;
       cout <<setprecision(2)<<"\t"<<yCO;
       outfile <<setprecision(2)<<"\t"<<yCO;
       cout <<setprecision(2)<<"\t"<<yHC;
       outfile <<setprecision(2)<<"\t"<<yHC;
       getch();
       }
//     }
 cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
 getch();
 getch();
 clrscr();
 if (LAMBDA > 1.0)
 {
 cout << "\n\n The type of fuel used is: 'Diesel+Ethanol' and the
mixture is 'Lean'. ";
 cout << "\n Assuming that, there is no 'dissociation' and
considering only 'Five Products': ";
 cout << "\n n1*CO2 ; n2*H2O ; n3*N2 ; n4*O2 ; n5*CO , ";
 cout << "\n\n the balaced chemical equation for '1 kmol' of fuel
is: ";
 cout << "\n\n (Diesel+Ethanol)+LAMBDA*ath*(02+3.76*N2) -
n1*CO2+n2*H2O+n3*N2+n4*O2+n5*CO ";
 outfile << "\n The type of fuel used is: 'Diesel+Ethanol' and the
mixture is 'Lean'. ";
```

```
 outfile << "\n Assuming that, there is no 'dissociation' and
considering only 'Five Products': ";
 outfile << "\n n1*CO2 ; n2*H2O ; n3*N2 ; n4*O2 ; n5*CO , ";
 outfile << "\n\n the balaced chemical equation for '1 kmol' of fuel
is: ";
 outfile << "\n\n (Diesel+Ethanol)+LAMBDA*ath*(O2+3.76*N2) -
n1*CO2+n2*H2O+n3*N2+n4*O2+n5*CO ";
 // nwv = 1.25;
 MmO = 16.0; MmN = 14.0; MmC = 12.0; MmH = 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) - n1th*CO2+n2th*H2O+n3th*N2
 yO2 = 0.21; yN2 = 0.79; MmO2 = 2.0*MmO; MmN2 = 2.0*MmN;
 yAl = yAl_per/100.0; yD = 1.0-yAl;
 nAl = yAl; nD = yD; nDAl = nD+nAl;
 MmD = (a*MmC)+(b*MmH); MmAl = (ZC*MmC)+(ZH*MmH)+(ZO*MmO);
 MmA = (yO2*MmO2)+(yN2*MmN2); MmF = (yD*MmD)+(yAl*MmAl);
 mAl = nAl*MmAl; mD = nD*MmD; mDAl = mD+mAl;
 xAl = mAl/mDAl; xD = mD/mDAl; nF_pkmol = nDAl;
 n1 = a; n2 = b/2.0; ath = ((2.0*a)+(b/2.0))/2.0; // ath = Zcc =
C+(H/4.0);
 n3 = (LAMBDA*ath*3.76*2.0)/2.0;
 n4 = ((LAMBDA*ath*2.0)-((2.0*n1)+n2))/2.0; n5 = 0.0;
 np = n1+n2+n3+n4+n5;
 y1 = n1/np; y2 = n2/np; y3 = n3/np; y4 = n4/np; y5 = n5/np;
 nO2 = LAMBDA*ath; nN2 = LAMBDA*ath*3.76; yres = xres/0.98;
 nA = nO2+nN2; nF = nF_pkmol; nr = (nA+nF)/(1.0-yres);
 yO2 = nO2/nr; yN2 = nN2/nr; yF = nF/nr; yA = nA/nr; AF =
LAMBDA*AFth;
 QLHV_DAl = (xG*QLHV_D)+(xAl*QLHV_Al); QLHV = QLHV_DAl;
 RHO_DAL = (yG*RHO_D)+(yAl*RHO_Al); RHOF = RHO_DAL;
 xF = 1.0/(AF+1.0); xA = 1.0-(xF+xres);
 cout << "\n\n\n mixture composition: ";
 outfile << "\n\n\n mixture composition: ";
 cout << "\n --------------------- \n";
 outfile << "\n -------------------- \n";
 cout <<setprecision(5)<<" yAL = "<<yAl<<" ; yD = "<<yD;
 cout <<setprecision(5)<<"\n xAL = "<<xAl<<" ; xD = "<<xD;
 cout <<setprecision(5)<<"\n yF = "<<yF<<" ; yA = "<<yA<<" ; yres =
"<<yres;
 cout <<setprecision(5)<<"\n xF = "<<xF<<" ; xA = "<<xA<<" ; xres =
"<<xres;
 cout <<setprecision(5)<<"\n\n y1 = "<<y1<<" ; y2 = "<<y2<<" ; y3 =
"<<y3<<" ; y4 = "<<y4<<" ; y5 = "<<y5;
 outfile <<setprecision(5)<<"\n yF = "<<yF<<" ; yA = "<<yA<<" ; yres
= "<<yres;
 outfile <<setprecision(5)<<"\n xF = "<<xF<<" ; xA = "<<xA<<" ; xres
= "<<xres;
      }
 cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
 getch();
```

```
 getch();
 clrscr();
 cout << "\n\n Thermidynamic Analysis - Cycle Processes: ";
 outfile << "\n Thermidynamic Analysis - Cycle Processes: ";
 cout << "\n ------------------------------------------ \n";
 outfile << "\n ------------------------------------------ \n";
 cout <<"\n theta"; outfile <<"\n theta";
 cout <<"\tvolume" ;   outfile <<"\tvolume" ;
 cout <<"\tpress." ;       outfile <<"\tpress. " ;
 cout <<"\ttemp." ; outfile <<"\ttemp." ;
// cout <<"\t\t\t m"; outfile <<"\t\t\t m";
// cout <<"\teff" ; outfile <<"\teff" ;
 cout <<"\n (deg.)";  outfile <<"\n (deg.)";
 cout <<"\t (cc)" ;    outfile <<"\t (cc)" ;
 cout <<"\t (bar)" ;       outfile <<"\t (bar)" ;
 cout <<"\t (K)" ; outfile <<"\t (K)" ;
 cout <<"\t\t\n " ; outfile <<"\t\t\n " ;
// cout <<"\t (kg)\n" ; outfile <<"\t (kg)\n" ;
     xA = AF/(AF+1.0); xF = 1.0/(AF+1.0);
     xres = xres1; Tres = 1200.0;
     deltheta = 10.0;
 for (theta = 0.0; theta <= 720.0; theta = theta+deltheta)
 {
// process a-1 (polytropic intake process):
     paa = pim; Taa = Tim; maa = mm1_kg;
 if (theta > theta_a && theta <= 180.0)
 {
 thetaa1 = theta; rada1 = thetaa1*(PI/180.0);
 xpa1 = r*((1.0-cos(rada1))+((Rrl*sin(rada1)*sin(rada1))/2.0));
 Vsa1 = Ap*xpa1; Va1 = Vcl+Vsa1;
 V1 = Va1*1.0e6; p1 = paa; T1 = Taa; m1 = maa; theta1 = thetaa1;
      cout << setprecision(2)<<"\n "<<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(3)<<"\t"<<T1;
      outfile <<setprecision(3)<<"\t"<<T1;
//    cout <<setprecision(6)<<"\t\t\t"<<m1;
//    outfile <<setprecision(6)<<"\t\t\t"<<m1;
//    cout <<setprecision(2)<<"\t "<<Cg_miv;
      getch();
      }
// process 1-b (polytropic intake-compression process):
     V11 = V1*1.0e-6; p11 = pim; T11 = 333.0; m11 = m1; mFbb =
mF1_kg;
     T1b = T1; m1b = m1; dQ1b = 0.0; dW1b = 0.0;
 if (theta > 180.0 && theta <= theta_b)
 {
 theta1b = theta; rad1b = theta1b*(PI/180.0); t1b = rad1b/OMEGA;
 xp1b = r*((1.0-cos(rad1b))+((Rrl*sin(rad1b)*sin(rad1b))/2.0));
```

```
 Vs1b = Ap*xp1b; V1b = Vcl+Vs1b; dV1b = V1b-V11;
// for 300 K < T < 1000 K:
 a1O2r = 3.21; a2O2r = 0.11e-2; a3O2r = -0.06e-5; a4O2r = 0.13e-8;
 cpO2 = (a1O2r+(a2O2r*T1b)+(a3O2r*T1b*T1b)+(a4O2r*T1b*T1b*T1b))*Ru;
 a1N2r = 3.29; a2N2r = 0.14e-2; a3N2r = -0.04e-4; a4N2r = 0.06e-7;
 cpN2 = (a1N2r+(a2N2r*T1b)+(a3N2r*T1b*T1b)+(a4N2r*T1b*T1b*T1b))*Ru;
 cpA1b = (0.21*cpO2)+(0.79*cpN2); cvA1b = cpA1b-Ru;
 cp1b = cpA1b/MmA; cv1b = cvA1b/MmA; n1b = cp1b/cv1b;
 Acv1b = (2.0*Ap)+(PI*d*xp1b);
 Aiv_m = PI*dp_iv*(liv_max/2.0);
 Cg_miv = (Ap/Aiv_m)*(Cpm/Kv_iv); Cg_mivsqr = (Cg_miv*Cg_miv)/2.0;
 delp1ba = -(Cg_mivsqr*RHOim); delp1b = delp1ba*1.0e-5;
 mi1b = ETAv*RHOim*Vs1b; m1b = m11+mi1b; p1b = p11+delp1b;
 Rp1b = p1b/paa; Rn1bTp = (n1b-1.0)/n1b;
 Tres1b = Tres*pow(Rp1b,Rn1bTp);
 T1b = (xres*Tres1b)+(1.0-xres)*Taa;
 mAF1 = mi1b; mF1 = xF*mAF1;
 dQ1b_W = hcv*Acv1b*(TW-T1b); dQ1b1 = 0.0; dQ1b2 = dQ1b_W*t1b;
 dW1b1 = ((p1b-pcc)*1.0e5)*dV1b; dW1b_other = 0.0; dW1b2 =
dW1b_other;
 dQ1b = dQ1b1+dQ1b2; dW1b = dW1b1+dW1b2; Q1b = dQ1b; W1b = dW1b;
 Vb = V1b*1.0e6; pb = p1b; Tb = T1b; mb = m1b; thetab = theta1b;
 T1b = Tb; m1b = mb; dQ1b = Q1b; dW1b = W1b;
       cout << setprecision(2)<<"\n "<<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(3)<<"\t"<<Tb;
       outfile <<setprecision(3)<<"\t"<<Tb;
//     cout <<setprecision(6)<<"\t\t\t"<<mb;
//     outfile <<setprecision(6)<<"\t\t\t"<<mb;
//     cout <<setprecision(2)<<"\t "<<Cg_miv;
       getch();
       }
// process b-c (polytropic compression process):
      pbb = pb; Tbb = Tb; mbb = mb; mFbb = mF1; mbc = mb;
 if (theta > theta_b && theta < theta_c)
 {
 rad = theta*(PI/180.0);
 xpbc = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
 Vsbc = Ap*xpbc; Vbc = Vcl+Vsbc; VbVbc = Vt/Vbc; mbc = mbb; nbc =
1.35;
 pbc = pbb*pow(VbVbc,nbc); mbc = mbb; Rbc = 0.287;
 nbcT = nbc-1.0; Tbc = Tbb*pow(VbVbc,nbcT);
 Wbc = (mbc*Rbc*1.0e3*(Tbc-Tbb))/(1.0-nbc);
 Vc = Vbc*1.0e6; pc = pbc; Tc = Tbc; mc = mbc; Zibc = Zibc+1.0;
       cout << setprecision(2)<<"\n "<<theta ;
       outfile << setprecision(2)<<"\n "<<theta ;
       cout << setprecision(2)<<"\t"<<Vc;
       outfile << setprecision(2)<<"\t"<<Vc;
```

```
       cout <<setprecision(2)<<"\t"<<pc;
       outfile <<setprecision(2)<<"\t" <<pc;
       cout <<setprecision(3)<<"\t"<<Tc;
       outfile <<setprecision(3)<<"\t"<<Tc;
//     cout <<setprecision(6)<<"\t\t\t"<<mc;
//     outfile <<setprecision(6)<<"\t\t\t"<<mc;
       getch();
       }
// process c-2 (polytropic ignition-combustion process):
       Vc2i = Vc*1.0e-6; pc2i = pc; Tc2i = Tc; mc2i = mc; Tcc = Tc;
       pc2 = pc; Tc2 = Tc; Wcc = Wbc*1.0e-3;
 if (theta > theta_c && theta <= 360.0)
 {
 rad = theta*(PI/180.0);
 xpc2 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
 Vsc2 = Ap*xpc2; Vc2 = Vcl+Vsc2; mF1 = mF1_kg;
 mc2 = mc2i; Qin1 = mFbb*(QLHV*1.0e3);
 cvc2 = 0.821;
 thetascdc = (theta-theta_sc)/theta_dc;
 a2d = 6.908; mC = 3.0; mpow = mC+1.0;
 exp2d = -a2d*pow(thetascdc,mpow); xb = 1.0-exp(exp2d);
 Qinc2 = (Qin1*1.0e3)*xb; Q1c2 = Qinc2;
 As = (2.0*Ap)+(PI*d*xpc2);
 Qc2 = Q1c2;
// Wc2_d = (pc2-pcc)*1.0e5*(Vc2-Vc2i); Wc2 = Wc2_d+Wc2_other;
// delUc2 = mc2*cvc2*1.0e3*(Tc2-Tc2i)*;
 QWc2 = Qc2/(mc2*cvc2*1.0e3);
 Tc2 = QWc2+Tc2i; // nrc2 = mc2/Mmr; npc2 = mc2/Mmp;
 pc2 = ((Tc2/Tc2i)*(Vc2i/Vc2))*pc2i;
 Qc2 = (hcv*As*(TW-Tc2))*(rad/OMEGA);
 Wc2 = (pc2-pcc)*1.0e5*(Vc2-Vc2i);
 V2 = Vc2*1.0e6; p2 = pc2; T2 = Tc2; m2 = mc2;
 pc2 = p2; Tc2 = T2;
       cout << setprecision(2)<<"\n "<<theta ;
       outfile << setprecision(2)<<"\n "<<theta ;
       cout << setprecision(2)<<"\t"<<V2;
       outfile << setprecision(2)<<"\t"<<V2;
       cout <<setprecision(2)<<"\t"<<p2;
       outfile <<setprecision(2)<<"\t"<<p2;
       cout <<setprecision(3)<<"\t"<<T2;
       outfile <<setprecision(3)<<"\t"<<T2;
//     cout <<setprecision(6)<<"\t\t\t"<<m2;
//     outfile <<setprecision(6)<<"\t\t\t"<<m2;
       getch();
       }
// process 2-d (polytropic combustion process):
       V2di = V2*1.0e-6; p2di = p2; T2di = T2; m2di = m2;
       p2d = p2; T2d = T2; W22 = Wc2*1.0e-3;
 if (theta > 360.0 && theta <= theta_d)
 {
 rad = theta*(PI/180.0);
 xp2d = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
```

```
 Vs2d = Ap*xp2d; V2d = Vcl+Vs2d;
 m2d = m2di; Qin1 = mFbb*(QLHV*1.0e3);
 cv2d = 0.821;
 thetascdc = (theta-theta_sc)/theta_dc;
 a2d = 6.908; mC = 3.0; mpow = mC+1.0;
 exp2d = -a2d*pow(thetascdc,mpow); xb = 1.0-exp(exp2d);
 Qin2d = (Qin1*1.0e3)*xb;
 Q12d = Qin2d;
 As = (2.0*Ap)+(PI*d*xp2d);
 Q2d = Q12d;
// W2d_d = (p2d-pcc)*1.0e5*(V2d-V2di); W2d = W2d_d+W2d_other;
// delU2d = m2d*cv2d*1.0e3*(T2d-T2di);
 QW2d = Q2d/(m2d*cv2d*1.0e3);
 T2d = QW2d+T2di; // nr2d = m2d/Mmr; np2d = m2d/Mmp;
 p2d = ((T2d/T2di)*(V2di/V2d))*p2di;
 Q2d = (hcv*As*(TW-T2d))*(rad/OMEGA);
 W2d = (p2d-pcc)*1.0e5*(V2d-V2di);
 Vd = V2d*1.0e6; pd = p2d; Td = T2d; md = m2d;
 p2d = pd; T2d = Td;
       cout << setprecision(2)<<"\n "<<theta;
       outfile << setprecision(2)<<"\n "<<theta;
       cout << setprecision(2)<<"\t"<<Vd;
       outfile << setprecision(2)<<"\t"<<Vd;
       cout <<setprecision(2)<<"\t"<<pd;
       outfile <<setprecision(2)<<"\t"<<pd;
       cout <<setprecision(3)<<"\t"<<Td;
       outfile <<setprecision(3)<<"\t"<<Td;
//     cout <<setprecision(6)<<"\t\t\t"<<md;
//     outfile <<setprecision(6)<<"\t\t\t"<<md;
       getch();
       }
// process d-4 (polytropic expansion process):
      Vd4i = Vd*1.0e-6; pd4i = pd; Td4i = Td; md4i = md;
      Td4 = Td4i; Wdd = W2d*1.0e-3;
 if (theta > theta_d && theta <= theta_4)
 {
 rad = theta*(PI/180.0);
 xpd4 = r*((1.0-cos(rad))+((Rrl*sin(rad)*sin(rad))/2.0));
 Vsd4 = Ap*xpd4; Vd4 = Vcl+Vsd4;
 cpd4 = 1.005; cvd4 = 0.718; Rd4 = cpd4-cvd4; nd4 = 1.25; //
cpd4/cvd4;
 RVd4 = Vd4i/Vd4; pd4 = pd4i*pow(RVd4,nd4); md4 = md4i; nd4T = nd4-
1.0;
 Td4 = Td4i*pow(RVd4,nd4T);
 Wd4 = (md4*Rd4*1.0e3*(Td4-Td4i))/(1.0-nd4);
 V4 = Vd4*1.0e6; p4 = pd4; T4 = Td4; m4 = md4; n4 = nd4;
       cout << setprecision(2)<<"\n "<<theta;
       outfile << setprecision(2)<<"\n "<<theta;
       cout << setprecision(2)<<"\t"<<V4;
       outfile << setprecision(2)<<"\t"<<V4;
       cout <<setprecision(2)<<"\t"<<p4;
       outfile <<setprecision(2)<<"\t"<<p4;
```

```
       cout <<setprecision(3)<<"\t"<<T4;
       outfile <<setprecision(3)<<"\t"<< T4;
//     cout <<setprecision(6)<<"\t\t\t"<<m4;
//     outfile <<setprecision(6)<<"\t\t\t"<<m4;
       getch();
       }
// process 4-5 (polytropic exhaust-blowdown process):
      V44 = V4*1.0e-6; p44 = p4; T44 = T4; m44i = m4;
      T45 = T4; m45 = m4;
      W44 = Wd4*1.0e-3;
 if (theta > theta_4 && theta <= 540.0)
 {
 theta45 = theta; rad45 = theta45*(PI/180.0); t45 = rad45/OMEGA;
 xp45 = r*((1.0-cos(rad45))+((Rrl*sin(rad45)*sin(rad45))/2.0));
 Vs45 = Ap*xp45; V45 = Vcl+Vs45;
// for 300 K < T < 1000 K:
 a1O2r = 3.21; a2O2r = 0.11e-2; a3O2r = -0.06e-5; a4O2r = 0.13e-8;
 cpO2 = (a1O2r+(a2O2r*T45)+(a3O2r*T45*T45)+(a4O2r*T45*T45*T45))*Ru;
 a1N2r = 3.29; a2N2r = 0.14e-2; a3N2r = -0.04e-4; a4N2r = 0.06e-7;
 cpN2 = (a1N2r+(a2N2r*T45)+(a3N2r*T45*T45)+(a4N2r*T45*T45*T45))*Ru;
 n45 = 1.25;
 RHO45 = m45/V45;
 Acv45 = (2.0*Ap)+(PI*d*xp45);
 Aev_m = PI*dp_ev*(lev_max/2.0);
 me45 = ETAv*RHO45*Vs45; m45 = m44i-me45;
 Cg_mev = (Ap/Aev_m)*(Cpm/Kv_ev); Cg_mevsqr = (Cg_mev*Cg_mev)/2.0;
 delp45a = -(Cg_mevsqr*RHO45); delp45 = delp45a*1.0e-5;
 p45 = p44+delp45; Rp45 = p45/p44; Rn45Tp = (n45-1.0)/n45;
 T45 = T44*pow(Rp45,Rn45Tp); // (p45*1.0e5*V45)/(Z45*m45*R45*1.0e3);
 dQ45_W = hcv*Acv45*(TW-T45); dQ451 = 0.0; dQ452 = dQ45_W*t45;
 dW451 = ((p45-pcc)*1.0e5)*(V45-V44);
 dW45_other = 0.0; dW452 = dW45_other;
 dQ45 = dQ451+dQ452; dW45 = dW451+dW452;
 Q45 = dQ45; W45 = dW45;
 V5 = V45*1.0e6; p5 = p45; T5 = T45; m5 = m45; theta5 = theta45;
 T45 = T5; m45 = m5; dQ45 = Q45; dW45 = W45;
 m44i = m45;
       cout << setprecision(2)<<"\n "<<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(3)<<"\t"<<T5;
       outfile <<setprecision(3)<<"\t"<<T5;
//     cout <<setprecision(6)<<"\t\t\t"<<m5;
//     outfile <<setprecision(6)<<"\t\t\t"<<m5;
//     cout <<setprecision(2)<<"\t "<<Cg_mev;
       getch();
       }
// process 5-6 (polytropic exhaust-displacement process):
      p55 = p5; T55 = T5;
```

```
     W55 = dW451*1.0e-3;
 if (theta > 540.0 && theta <= theta_6)
 {
 theta56 = theta; rad56 = theta56*(PI/180.0);
 xp56 = r*((1.0-cos(rad56))+((Rrl*sin(rad56)*sin(rad56))/2.0));
 Vs56 = Ap*xp56; V56 = Vcl+Vs56;
 V6 = V56*1.0e6; p6 = p55/2.0; T6 = T55/2.0; theta6 = theta56;
      cout << setprecision(2)<<"\n "<<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(3)<<"\t"<<T6;
      outfile <<setprecision(3)<<"\t"<<T6;
//    cout <<setprecision(6)<<"\t\t\t"<<m6;
//    outfile <<setprecision(6)<<"\t\t\t"<<m6;
//    cout <<setprecision(2)<<"\t"<<Cg_mev;
      getch();
      }
      }
 cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
 getch();
 getch();
 clrscr();
 cout << "\n\n Check ! for 'Violation' ... ";
 outfile << "\n Check ! for 'Violation' ... ";
 cout << "\n ------------------------------ \n";
 outfile << "\n ------------------------------ \n";
 Wnet1 = Wcc+W22+Wdd+W44+W55;
 EFin1 = mFbb*QLHV*1.0e3; EFin1_kW = EFin1*n_cps; EFin_t =
EFin1_kW*Zc;
 Qin1 = (mFbb*QLHV*1.0e3)*ETAc; Qin1_kW = Qin1*n_cps; Qin_t =
Qin1_kW*Zc;
 Pi1_kW = Wnet1*n_cps; Pi_t = Pi1_kW*Zc;
 Pb1_kW = Pi1_kW*ETAm; Pb_t = Pb1_kW*Zc*1.4;
 Qout_t = Qin_t-Pb_t;
 mFbb_kgps = mFbb*n_cps; mF_kgph = (mFbb_kgps*3600.0)*Zc;
 bsfc_t = mF_kgph/Pb_t; Qincd = mFbb*(QLHV*1.0e3);
 bmep_t = ((Pb_t*1.0e3)/(Ec*n_cps))*1.0e-5;
 Tcd_m = (Tcc+2500.0)/2.0; cvcd_m = 0.718+(20.0e-5*Tcd_m);
 Tmin = T11; T33 = (Qincd*1.0e3)/(cvcd_m*1.0e3*mbb); Tmax = T33;
 ETAith_carn = 1.0-(Tmin/Tmax); ETAith_carnper = ETAith_carn*100.0;
 ETAith = Pi_t/Qin_t; ETAith_per = ETAith*100.0;
 ETAbth = Pb_t/EFin_t; ETAbth_per = ETAbth*100.0;
 OPD = Pb_t/Ec_liter;
     cout <<setprecision(2)<<"\n Qin = "<<Qin_t<<" kW ";
     cout <<setprecision(2)<<" ; Wnet = "<<Pb_t<<" kW ";
     cout <<setprecision(2)<<" ; Qout = "<<Qout_t<<" kW ";
     cout <<setprecision(2)<<"\n\n ETAith_carn =
"<<ETAith_carnper<<" % ";
     cout <<setprecision(2)<<" ; ETAith = "<<ETAith_per<<" % ";
```

```
      cout <<setprecision(2)<<" ; ETAbth = "<<ETAbth_per<<" % ";
      cout <<setprecision(2)<<"\n\n Rc = "<<Rc<<":1 ";
      cout <<setprecision(2)<<" ; d = "<<d_mm<<" mm ";
      cout <<setprecision(2)<<" ; Rsd = "<<Rsd<<":1 ";
      cout <<setprecision(2)<<"\n\n bmep = "<<bmep_t<<" bar ";
      cout <<setprecision(2)<<" ; bsfc = "<<bsfc_t<<" kg/kWh ";
      cout <<setprecision(2)<<"\n\n OPD = "<<OPD<<" kW/liter ";
 cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
 getch();
 getch();
 clrscr();
 cout << "\n\n mass fraction burned (xb) and arbitrary heat release
(Qr): ";
 outfile << "\n\n mass fraction burned (xb) and arbitrary heat
release (Qr): ";
 cout << "\n ----------------------------------------------------
----- \n";
 outfile << "\n ----------------------------------------------------
----- \n";
 cout <<"\n theta"; outfile <<"\n theta";
 cout <<"\txb( )" ; outfile <<"\txb( )" ;
 cout <<"\tQr(kJ)" ;       outfile <<"\tQr(kJ)" ;
 for (theta = theta_sc; theta <= theta_ec; theta = theta+deltheta)
 {
 thetascdc = (theta-theta_sc)/theta_dc;
 a2d = 5.908; mC = 2.0; mpow = mC+1.0;
 exp2d = -a2d*pow(thetascdc,mpow); xb = 1.0-exp(exp2d);
 Qin2d = (Qin1*1.0e3)*xb; Qr = Qin2d;
 cout << setprecision(2)<<"\n "<<theta;
 cout << setprecision(2)<<"\t "<<xb;
 cout << setprecision(2)<<"\t "<<Qr;
 outfile << setprecision(2)<<"\n "<<theta;
 outfile << setprecision(2)<<"\t "<<xb;
 outfile << setprecision(2)<<"\t "<<Qr;
 getch();
 }
 cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
 getch();
 getch();
 clrscr();
 cout << "\n\n Power (Energy) Flow and Efficiencies: ";
 outfile << "\n Power (Energy) Flow and Efficiencies: ";
 cout << "\n --------------------------------------- \n";
 outfile << "\n --------------------------------------- \n";
      cout <<setprecision(2)<<"\n EFin = "<<EFin_t<<" kW ";
      cout <<setprecision(2)<<" ; Qin = "<<Qin_t<<" kW ";
      cout <<setprecision(2)<<" ; ETAc = "<<ETAc_per<<" % ";
      cout <<setprecision(2)<<"\n\n Pi = "<<Pi_t<<" kW ";
      cout <<setprecision(2)<<" ; ETAith = "<<ETAith_per<<" % ";
//    cout <<setprecision(2)<<" ; ETAith_carn = "<<ETAith_carnper<<"
% ";
      cout <<setprecision(2)<<"\n\n Pb = "<<Pb_t<<" kW ";
```

```
     cout <<setprecision(2)<<" ; ETAm = "<<ETAm_per<<" % ";
 cout << "\n\n\t\t Press 'enter' twice to proceed ... ";
 getch();
 getch();
 clrscr();
 cout << "\n\n Engine Performance: ";
 outfile << "\n\n Engine Performance: ";
 cout << "\n -------------------------- \n";
 outfile << "\n -------------------------- \n";
     Pbmax = (bmep_t*1.0e5*Ec*n_cps)*1.0e-3;
 N_min = 700.0; N_max = N_Pbmax; delN = (N_max-N_min)/10.0;
 for (N = N_min; N <= N_max; N = N+delN)
 {
 x = N/N_Pbmax; OMEGA = (2.0*PI*N)/60.0;
 Pb = Pbmax*(-(x*x*x)+(1.4*x*x)+(0.6*x));
// Pb = Pbmax*(-(x*x*x)+(1.13*x*x)+(0.87*x));
 Tb = (Pb*1.0e3)/OMEGA;
 bsfc = bsfc_Pbmax*((x*x)-(1.55*x)+1.55);
 cout << setprecision(2)<<"\n\t "<<Pb;
 cout << setprecision(2)<<"\t "<<Tb;
// cout << setprecision(2)<<"\t "<<bmep;
 cout << setprecision(2)<<"\t "<<bsfc;
 cout << setprecision(2)<<"\t "<<N;
 outfile << setprecision(2)<<"\n\t "<<Pb;
 outfile << setprecision(2)<<"\t "<<Tb;
// outfile << setprecision(2)<<"\t "<<bmep;
 outfile << setprecision(2)<<"\t "<<bsfc;
 outfile << setprecision(2)<<"\t "<<N;
 getch();
 }
 cout << "\n\n\n\t\t Press 'enter' twice to exit ! ";
 getch();
 getch();
 getche();
 }
```