Feasibility Study of NOx Reduction by Exhaust Gas Recirculation (EGR) in Bio Diesel Fueled C I Engine

By

Nirav Joshi

08MMET20

DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

May 2010

Feasibility Study of NOx Reduction by Exhaust Gas Recirculation (EGR) in Bio Diesel Fueled C I Engine

Major Project Report

Submitted in partial fulfillment of the requirements

for the degree of

MASTER OF TECHNOLOGY

IN

MECHANICAL ENGINEERING

(THERMAL ENGINEERING)

By

Nirav Joshi

08MMET20

DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

May 2010

Declaration

This is to certify that

i) The thesis comprises my original work towards the Degree of Master of Technology in Thermal Engineering at Nirma University and has not been submitted elsewhere for a degree.

ii) Due acknowledgement has been made in the text to all other material used.

Nirav Joshi

Certificate

This is to certify that the Major Project entitled "Feasibility Study of NOx Reduction by Exhaust Gas Recirculation (EGR) in Bio Diesel Fueled C I Engine" submitted by **Niravkumar Shankarlal Joshi (08MMET20)**, towards the partial fulfilment of the requirements for the Degree of Master of Technology in Mechanical Engineering(Thermal Engineering) of Institute of Technology, Nirma University, Ahmedabad is the record of work carried out by him under my supervision and guidance. In my opinion, the submitted work has reached a level required for being accepted for examination. The results embodied in this major project, to the best of my knowledge, haven't been submitted to any other university or institution for award of any degree or diploma.

Prof. N K Shah Co-Guide and APME Institute of Technology, Nirma University.	Prof. A M Lakdawala Co-Guide and APME Institute of Technology, Nirma University.	Dr. R N Patel Guide and Professor Institute of Technology, Nirma University.

Prof. V R Iyer Head of Department Mechanical Engg. Dept. Institute of Technology, Nirma University.	Dr. K Kotecha Director Institute of Technology, Nirma University.

Acknowledgements

First of all, I wish to thank Institute of Technology, Nirma University, whose course curriculum provided me an opportunity for an exposure to the field of project work related to Thermal Engineering.

With great pleasure I wish to express my deep gratitude to Prof. V R Iyer, (Head of Department, Mechanical Engineering, Institute of Technology, Nirma University) for his keen interest, constant encouragement and valuable guidance for this work.

I would like to express thanks and gratitude to my project guide Dr. R N Patel, Professor, Mechanical Engineering Department, IT, NU, and co guides Prof. A M Lakdawala, APME, IT, NU and Prof. N K Shah, APME, IT, NU for providing me guidance from initiation of this work till end of the project. Without them it is not possible for me to complete the task.

I am very much thankful to the faculty members of Mechanical Engineering Department for their expertise knowledge and guidance throughout my study, which has provided me valuable insight in number of areas.

I would like to express my thanks to entire workshop staff for their true support during preparation of experimental setup for the project.

I am very much thankful to Central Salt and Marine Chemicals Research Institute (CSMCRI), Bhavnagar for providing the required quantity of Biodiesel of known chemical properties to carry out the experimental work.

Last but certainly note the least, I wish to express my profound appreciation to all my friends and well wishers for all their direct or indirect help in this project work. It is indeed a pleasure for me to express my sincere gratitude to those who have always helped me throughout my project work.

- Nirav Joshi

08MMET20

Abstract

The aim of this project work is to check the feasibility of Exhaust Gas Recirculation (EGR) technique for NOx reduction in a compression ignition engine fueled by biodiesel diesel blend.

Biodiesel is an oxygenated fuel. When biodiesel is used in an I C Engine, the formation of NOx becomes higher. Different techniques are available for reduction in NOx formation, but Exhaust Gas Recirculation (EGR) is the most convenient because it can be implemented to any I C Engine without much modification in the engine design and noticeable reduction in NOx formation can be achieved.

A deep literature review had been carried out from the research papers published nationally and internationality. Experimental setup had been prepared as per the requirement. During Initial experiments, fuel composition had been changed by supplying various diesel-biodiesel blends.

After comparison of results, it was found that 10% Biodiesel (on volume basis) can be mixed with conventional diesel without much degradation in engine parameters.

After getting optimized biodiesel-diesel blend, experiments were carried out with 5%, 10% and 15% Exhaust Gas Recirculation (EGR). At the end of work, it was found that 10% EGR acts most effectively by 80% reduction in NOx formation at part load conditions.

Key words: EGR, Biodiesel, Biodiesel-Diesel blend, NOx reduction, Emission Control

Contents

List of Tables

List of Figures

Nomenclature

P	break power (kW);
α	power adjustment factor;
k	ratio of indicated power;
p	barometrik pressure (kPa);
p_s	saturation vapour pressure (kPa);
Φ	relative humidity;
T	absolute air inlet temperature, (K);
η_m	mechanical efficiency.
Subscript "r"	values under the standard reference condition
Subscript "x"	values under the ambient site condition

Chapter 1

Introduction

Diesel engines have been widely used as power of engineering machinery, automobile, and shipping equipment for its drivability and thermal efficiency. At the same time, diesel engine are major contributors of various types of air pollutant emissions such as carbon monoxide (CO), oxides of nitrogen (NOx), soot and other harmful compounds. With the increasing concern of environmental protection and more stringent government regulation on exhaust emissions, reductions in engine emissions become a major research task in engine development. Simultaneous reduction in nitric oxides emissions and particulate matter is quite difficult due to the soot/NOx trade off and is often accompanied by fuel consumption penalties. Thus, improvement of fuel properties is also essential for the suppression of diesel pollutant emissions along with the optimization of combustion related design factors and exhaust after treatment equipment.[1]

The interest in renewable energy sources for energy production is not new. Many studies have been conducted to qualify various oil and their blends from plants and vegetables as alternative renewable energy sources. Biodiesel may also help in reducing the net production of CO_2 from combustion sources and our dependence on fossil fuels. Often the vegetable oils investigated for their suitability as biodiesel are those which occur abundantly in the country of testing. Therefore, soybean oil is of primary interest as biodiesel source in the United States while many European countries are concerned with rapeseed oil, and countries with tropical climate prefer to utilize coconut oil, hazelnut or palm oil. Other vegetable oils, including sunflower, rubber, etc., have also been investigated. Furthermore, other sources of biodiesel studied include animal fats, salmon oil and or waste cooking oils.[2]

Use of efficient diesel engines need encouragement in the future since they consume less fuel and significantly reduce effective green house gases like carbon dioxide. Ever increasing diesel consumption, large outflow of foreign exchange and concern for environment have prompted developing countries like India to search for a suitable environmental friendly alternative to diesel fuel. The country has to simultaneously address the issues of energy insecurity, increasing oil prices and large-scale unemployment.

Straight Vegetable Oils (SVO) even though projected as an engine friendly fuel by many researchers have recently lost its attraction. Being highly viscous and less volatile, SVO's will result in poor spray atomization, vaporization, and pose serious threat to the engine health in the long run. More over many SVO's are edible oils whose continuous supply cannot be ensured in India.

Straight Vegetable Oils (SVO) are chemically modified into bio-diesel through a transesterification process. Bio-diesel thus obtained has properties close to diesel fuel and is found to be engine friendly. In spite of several advantages, Jatropha based bio-diesel (JBD) is found to emit higher NOx compared to diesel fuel. Higher NOx level in the JBD exhaust as reported by several researchers, is a serious issue to be addressed before its wide spread implementation. The researchers also found higher NO emissions when the JBD was tested in the laboratory. Higher NOx emission from JBD is probably due to their higher bulk modulus and boiling point. Inherent oxygen in its structure can also aggravate the situation.[3]

Four major emissions produced by internal combustion engines are hydrocarbons (HC), carbon monoxide (CO), oxides of nitrogen (NOx), and solid particulates. Hydrocarbons are fuel molecules which did not get burned and smaller non equilibrium particles of partially burned fuel. Carbon monoxide occurs when not enough oxygen is present to fully react all carbon to CO_2 or when incomplete air-fuel mixing occurs due to the very short engine cycle time. Oxides of nitrogen are created in an engine when high combustion temperatures cause some normally stable N_2 to dissociate into monatomic nitrogen N, which then combines with reacting oxygen. Solid particulates are formed in compression ignition engines and are seen as black smoke in the exhaust of these engines. Other emissions found in the exhaust of engines include aldehydes, sulfur, lead, and phosphorus.

Two methods are being used to reduce harmful engine emissions. One is to improve the technology of engines and fuels so that better combustion Occurs and fewer emissions are generated. The second method is after treatment of the exhaust gases. This is done by using thermal converters or catalytic converters that promote chemical reactions in the exhaust flow. These chemical reactions convert the harmful emissions to acceptable CO_2, H_20, and N_2.

Exhaust gases of an engine can have up to 2000 ppm of oxides of nitrogen. Most of this will be nitrogen oxide (NO), with a small amount of nitrogen dioxide (NOx), and traces of other nitrogen-oxygen combinations. These are all grouped together as NOx, with x representing some suitable number. NOx is a very undesirable emission, and regulations that restrict the allowable amount continue to become more stringent. Released NOx reacts in the atmosphere to form ozone and is one of the major causes of photochemical smog.

1.1 NOx Reduction Strategies

Even though some cetane improving additives are capable of reducing NOx, the amount of reduction is reported to be inadequate. Moreover, most of the additives are expensive and can promote auto-oxidation in bio-diesel. Extensive studies have revealed that NOx reduction by altering fuel properties is highly limited.

Retarded injection is an effective method employed in diesel engines for NOx control. However, this method leads to increased fuel consumption, reduced power, increased HC and excess smoke. Water injection on the other hand is prone to corrosion. In addition, it adds to the weight of the engine system for maintaining a water storage tank. It is also difficult to retain water at a desired value during cold climate.

Exhaust Gas Recirculation (EGR) is an effective method for NOx control. The exhaust gases mainly consist of inert carbon dioxide, nitrogen and possess high specific heat. When recirculated to engine inlet, it can reduce oxygen concentration and act as a heat sink. This process reduces oxygen concentration and peak combustion temperature, which results in reduced NOx.[3]

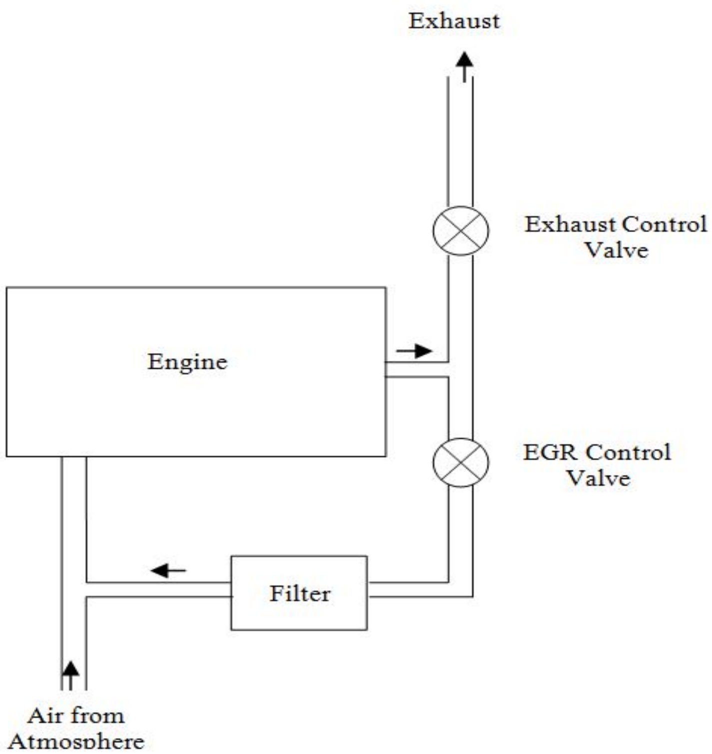

Figure 1.1: EGR System

To study the effects of EGR on the performance and emissions of automotive engines, the system shown in Fig. 1.1 gives an example of the EGR system. To remove the smoke, a particulate trap filter is used. Part of the EGR pipe is made from flexible stainless steel to avoid transfer of engine vibration to the exhaust system and then to the measuring instruments. Glass wool is used to insulate the EGR pipe to minimize the reduction in gas temperature.

1.2 Organization of Thesis

The thesis has been organized in five chapters. The abstract of the thesis and the key words have been presented before the contents of the thesis.

Chapter one is introduction to the project work. This covers the overall picture of the project including motivation for the project work.

In chapter two, literature review regarding oxygenated fuels, diesel-biodiesel blends and techniques for NOx reduction has been given, which shall be immensely helpful for the further research work in the direction.

Chapter three includes the discussion about different physicochemical properties of diesel, biodiesel, diesel-biodiesel blend followed by instrumentation of the test rig used for the experimentation and methodology of experiments (IS 10000 Part IV) have been discussed. The procedure of statistical and uncertainty analysis done in this work have also been included in this chapter.

In chapter four all the results obtained during the experimentation have been discussed in detail.

Chapter five is where concluding remarks are given. The scope and motivation for the future work in the direction are also elaborated

The thesis also includes resource of the references. This is followed by the Appendix section in which various tables and other reference documents are given.

Chapter 2

Literature Review

Based on problem definition, before carrying out a project, it is required to know the work carried out by different researchers nationally and internationally. Number of journals and conference papers were reviewed, out of which papers with concern topic are briefly summarized as under.

Saleh H E [2] used Jojoba methyl ester (JME)as a renewable fuel in numerous studies evaluating its potential use in diesel engines. These studies showed that this fuel is good gas oil substitute but an increase in the nitrogenous oxides emissions was observed at all operating conditions. The aim of this study mainly was to quantify the efficiency of Exhaust Gas Recirculation (EGR) when using JME fuel in a fully instrumented, two-cylinder, naturally aspirated, four-stroke direct injection diesel engine. The tests were carried out in three sections. Firstly, the measured performance and exhaust emissions of the diesel engine operating with diesel fuel and JME at various speeds under full load are determined and compared. Secondly, tests were performed at constant speed with two loads to investigate the EGR effect on engine performance and exhaust emissions including nitrogenous oxides (NOx), carbon monoxide (CO), unburned hydrocarbons (HC) and exhaust gas temperatures. Thirdly, the effect of cooled EGR with high ratio at full load on engine performance and emissions was examined. The results showed that EGR is an effective technique for reducing NOx emissions with JME fuel especially in light-duty diesel engines. With the application of the EGR method, the CO and HC concentration in the engine-out emissions increased. For all operating conditions, a better trade-off between HC, CO and NOx emissions can be attained within a limited EGR rate of 5-15 % with very little economy penalty.

Pradeep and Sharma [3] carried out an experimental work in this field. They have concluded that diesel engines running on JBD are found to emit higher oxides of nitrogen, NOx. HOT EGR, a low cost technique of Exhaust Gas Recirculation, is effectively used in this work to overcome this environmental penalty. Practical problems faced while using a COOLED EGR system are avoided with HOT EGR. Results indicated higher nitric oxide (NO) emissions when a single cylinder diesel engine was fueled with JBD, without EGR. NO emissions were reduced when the engine was operated under HOT EGR levels of 5-25 %. However, EGR level was optimized as 15 % based on adequate reduction in NO emissions, minimum possible smoke, CO, HC emissions and reasonable brake thermal efficiency. Smoke emissions of JBD in the higher load region were lower than diesel, irrespective of the EGR levels. However, smoke emission was higher in the lower load region. CO and HC emissions were found to be lower for JBD irrespective of EGR levels. Combustion parameters were found to be comparable for both fuels.

Abd-Alla [4] G H, reviewed the potential of Exhaust Gas Recirculation (EGR) to reduce the exhaust emissions, particularly NOx emissions, and to delimit the application range of this technique. A detailed analysis of previous and current results of EGR effects on the emissions and performance of Diesel engines, spark ignition engines and duel fuel engines is introduced. From the deep analysis, it was found that adding EGR to the air flow rate to the Diesel engine, rather than displacing some of the inlet air, appears to be a more beneficial way of utilizing EGR in Diesel engines. This way may allow exhaust NOx emissions to be reduced substantially.

In spark ignition engines, substantial reductions in NO concentrations are achieved with 10 to 25 % EGR. However, EGR also reduces the combustion rate, which makes stable combustion more difficult to achieve. At constant burn duration and brake mean effective pressure, the brake specific fuel consumption decreases with increasing EGR. The improvement in fuel consumption with increasing EGR is due to three factors: firstly, reduced pumping work; secondly, reduced heat loss to the cylinder walls; and thirdly, a reduction in the degree of dissociation in the high temperature burned gases. In dual fuel engines, with hot EGR, the thermal efficiency is improved due to increased intake charge temperatures and reburning of the unburned fuel in the recirculated gas. Simultaneously, NOx is reduced and smoke is reduced to almost zero at high natural gas fractions. Cooled EGR gives lower thermal efficiency than hot EGR but makes possible lower NOx emissions.

Zheng et al.[5] worked on this topic. According to their work, Exhaust Gas Recirculation (EGR) is effective to reduce nitrogen oxides (NOx) from Diesel engines because it lowers the flame temperature and the oxygen concentration of the working fluid in the combustion chamber. However, as NOx reduces, particulate matter (PM) increases, resulting from the lowered oxygen concentration. When EGR further increases, the engine operation reaches zones with higher instabilities, increased carbonaceous emissions and even power losses. In this research, the paths and limits to reduce NOx emissions from Diesel engines are briefly reviewed, and the inevitable uses of EGR are highlighted. The impact of EGR on Diesel operations is analyzed and a variety of ways to implement EGR are outlined.
At the end of their work, they made a conclusion that Diesel exhaust contains sulfuric salts and other abrasive and corrosive substances. It has been argued whether EGR should be applied to Diesel engines because of the increased piston-cylinder wearing. Heavy uses of EGR could also deteriorate the energy efficiency, operational stability and PM generation of the engine. However, the concern over increased wearing and deteriorated performance has soon given way to stringent emission regulations. In

stark contrast, the current concern is on how aggressively EGR should be applied to all speeds and all loads; although EGR increased wearing continues to be a problem affecting engine durability and performances. To date, EGR is still the most viable technique that can reduce NOx dramatically. Energy efficient after treatment systems dealing with NOx and PM simultaneously are still in the early development stages. The inability of available catalytic after treatment technologies further encourages aggressive uses of EGR.

Lapuerta et al.[6] collected and analyzed the body of work written mainly in scientific journals about diesel engine emissions when using biodiesel fuels as opposed to conventional diesel fuels. Since the basis for comparison is to maintain engine performance, the first section is dedicated to the effect of biodiesel fuel on engine power, fuel consumption and thermal efficiency. The highest consensus lies in an increase in fuel consumption in approximate proportion to the loss of heating value. In the subsequent sections, the engine emissions from biodiesel and diesel fuels are compared, paying special attention to the most concerning emissions: nitric oxides and particulate matter, the latter not only in mass and composition but also in size distributions. In this case the highest consensus was found in the sharp reduction in particulate emissions. Most of studies report slight increases in NOx emissions when using biodiesel fuels. The reason most frequently pointed out is that the injection process is slightly advanced with biodiesel. The physical properties of biodiesel or the response of the electronic unit could cause such an advance. Some authors propose delaying injection as a mean to eliminate the increase in NOx emissions, with a minor penalty in particulate emissions.

Mahlaa et al.[7] shown that NOx emission is higher in case of biodiesel fueled engine. Exhaust Gas Recirculation is an effective technique to emission of NOx from diesel engines because of lowering in peak cylinder temperature and oxygen content during combustion. The only penalty for using EGR is increase in smoke. The aim of the present research work is to utilize the B20 blend of jatropha methyl ester and cooled EGR simultaneously in order to reduce pollutant from the diesel engine. A single cylinder, air cooled, four stroke, natural aspirated DI diesel engine capable of producing 5.9 KW@1500 r.p.m. was used for experiments. Emissions of NOx, CO, HC and smoke opacity were recorded and compared with baseline diesel. Peak Cylinder pressure and heat release rate were recorded with crank angle at full load condition. Various performance parameters was evaluated such as brake thermal efficiency, BSFC, BSEC were calculated. Results indicate the improved brake thermal efficiency and reduction in NOx emission with the application of EGR and usage of B20 blend of biodiesel. On the basis of experimental data, it was found that biodiesel and EGR both can be used in compression ignition engine to simutanously reduce NOx and smoke emissions. Peak cylinder pressure was lower than baseline diesel level. Delay period was longer because of decrease in flame speed. Exhaust emissions such as HC and CO are also found to decrease. The B20 blend of biodiesel together with 14 percent EGR was found to be useful in improving the brake thermal efficiency, reduces exhaust emissions and the BSEC. Biodiesel contains 10 12 percent oxygen content due to which it undergoes more complete combustion which leads to higher NOx mission. This is controlled by employing the EGR method. EGR helps in reducing NOx emissions by reducing the availability of extra oxygen for the formation of NOx and absorbing the heat during combustion thereby reducing the temperature during the combustion. Thus, diesel engine can be useful operated with B20 blend of biodiesel along with application of EGR to reduce the NOx emission without increasing the smoke.

NOx (Oxides of Nitrogen) is considered as one of the most objectionable pollutant from diesel engine. Exhaust Gas Recirculation (EGR), has long been of interest to engine designers, researchers, and regulating authorities in abating NOx. Implementation of EGR for naturally aspirated stationary diesel engines are relatively simple and a study of the application of high level EGR on such an engine is carried to study its impact on the other performance parameters. The results of this investigation, carried out by Hebbar and Bhat[8] give insight into the effect of EGR level on the development of gaseous emissions as well as mechanisms of its formation. Reductions in NOx amount are found to be remarkable with EGR but combustion quality deteriorates at higher loads and higher percentages of EGR due to a significant decrease of A/F ratio. EGR up to 60 percent is found optimum with 190 bar injection pressure without sacrificing thermal efficiency and increase in unburned HC significantly.

Vegetable methyl ester was added in ethanol-diesel fuel to prevent separation of ethanol from diesel in this study carried out by Chen et al.[9] The ethanol blend proportion can be increased to 30 percent in volume by adding the vegetable methyl ester. Engine performance and emissions characteristics of the fuel blends were investigated on a diesel engine and compared with those of diesel fuel. Experimental results show that the torque of the engine is decreased by 6-7 percent for every 10 percent (by volume) ethanol added to the diesel fuel without modification on the engine. Brake specific fuel consumption (BSFC) increases with the addition of oxygen from ethanol but equivalent brake specific fuel consumption (EBSFC) of oxygenated fuels is at the same level of that of diesel. Smoke and particulate matter (PM) emissions decrease significantly with the increase of oxygen content in the fuel. However, PM reduction is less significant than smoke reduction. In addition, PM components are affected by the oxygenated fuel. When blended fuels are used, nitrogen oxides (NOx) emissions are almost the same as or slightly higher than the NOx emissions when diesel fuel is used. Hydrocarbon (HC) is apparently decreased when the engine was fueled with ethanol-ester-diesel blends. Fuelling the engine with oxygenated diesel fuels showed increased carbon monoxide (CO) emissions at low and medium loads, but reduced CO emissions at high and full loads, when compared to pure diesel fuel. NOx emissions of the oxygenated blends showed a slight increase or at close level to that of diesel fuels. E10B increases NOx emission slightly comparing with diesel at all load, E20B and E30B emit the same NOx level as diesel. The trade-off relation of

PM and NOx is broken by the oxygenated fuels.

The present investigation was to study the effect of Exhaust Gas Recirculation (EGR) on homogeneous charge ignition engine by Miller Jothi et al.[10] A stationary four stroke, single cylinder, direct injection (DI) diesel engine capable of developing 3.7 kW at 1500 rpm was modified to operate in HCCI mode. In the present work the diesel engine was operated on 100 percent Liquified Petroleum Gas (LPG). The LPG has a low cetane number (¡3), therefore Diethyl ether (DEE) was added to the LPG for ignition purpose. DEE is an excellent ignition enhancer (cetane number ¿125) and has a low auto ignition temperature (160 C). Experimental results showed that by EGR technique, at part loads the brake thermal efficiency increases by about 2.5 percent and at full load, NO concentration could be considerably reduced to about 68 percent as compared to LPG operation without EGR. However, higher EGR percentage affects the combustion rate and significant reduction in peak pressure at maximum load.

Ibrahim and Bari [11] used the stoichiometric air-fuel mixture with Exhaust Gas Recirculation (EGR) technique in a spark-ignition natural gas engine for experimental investigation. Engine performance and NO emissions were studied for both atmospheric and supercharged inlet conditions. It was found that the use of EGR has a significant effect on NO emissions. NO emissions decreased by about 50 percent when EGR dilution increased from zero with an inlet pressure of 101 kPa to close to the misfire limit with an inlet pressure of 113 kPa. In addition, the use of EGR effectively suppressed abnormal combustion which occurred at higher inlet pressure. The use of higher inlet pressure in the presence of EGR improved engine performance significantly. Engine brake power increased by about 20 percent and engine fuel consumption decreased by about 7 percent while NO emissions decreased by about 12 percent when 5 percent of EGR dilution was employed with an inlet pressure of 113 kPa compared to using undiluted stoichiometric inlet mixture with an inlet pressure of 101 kPa.

2.1 Problem Definition

The diesel engine presents rather different problems from the spark-ignition engine because it always operates with considerable excess air, so that CO emissions are not a significant problem, and the close control of air/fuel ratio, so significant in the control of gasoline engine emissions, is not required. On the other hand, particulates are much more of a problem and NOx production is substantial. NOx emissions are very sensitive to maximum cylinder temperature and to the excess air factor. This has prompted the use of increased levels of turbo charging with improved after-cooling, as well as the use of retarded injection which results in reduced peak pressures and temperatures but, beyond a certain point, in increased fuel consumption.

Biodiesel has a high cetane number than diesel fuel, no aromatics, no sulfur and contains 10-12 % oxygen by weight. These characteristics of biodiesel reduce the harmful emissions of unburned hydrocarbon (HC), CO, and particulates than the diesel fuel. Several researchers found increased NOx emission as compare to petrol-diesel. For reduction in NOx emission some exhaust gas after treatment such as, Exhaust Gas Recirculation (EGR) can be used with biodiesel. EGR helps in reducing NOx emissions by reducing the availability of extra oxygen for the formation of NOx and absorbing the heat during combustion thereby reducing the temperature during the combustion. Since the specific heat capacity of exhaust gas is more it absorbs more heat thereby reducing the high temperature available during the time of combustion.

2.2 Objective of the Project

The literature results indicate that the emission of nitric oxide (NO) is higher when a single cylinder diesel engine is fuelled with JBD, without EGR. The emission study of biodiesel fueled CI indicates that there is a reduction in NOx emission when engine is operated with EGR. However, no work is reported on the optimization of the biodiesel-diesel blend with different percentage of EGR.

On the basis of the above literature survey, following objective is concluded.

The project work is to check feasibility of EGR in C.I. engine fuelled with bio-diesel blend. In India, the use of bio-diesel is still not accepted commercially. When the bio-diesel blend will be used commercially, the formation of NOx will become higher. The aim of my project is to reduce the NOx formation in C.I. engine fuelled with bio-diesel blend with the use of EGR. Hence, the experimental work carried out can be useful for the actual commercial design of the C.I. engine to accept the use of bio-diesel as an alternative fuel. Aim of this priject is to find out the effect of biodiesel on physico-chemical properties of the biodiesel-diesel blends and optimization of biodiesel-diesel blend and percentage recirculation of exhaust gas for minimum emissions.

The experimental work under this project includes comparision of properties of different blends and base fuels and analysis of the effect of EGR on performance, emission and combustion parameters of C.I. engine fueled with bio-diesel-diesel blend will be carried out on single-cylinder, four-stroke, air cooled engine developing 5 kW at 1500 rpm. To ensure repeatability of the experiments, testing of engine will be carried out as per Indian standard 10000 Part IV, which lays down the guidelines for declaring power, efficiency and fuel consumption and specifies relevant correction factors which are required for adjusting the observed reading to the standard reference condition.

Chapter 3

Methodology of Work

3.1 Plan of the experiments

The main aim of the experimentation is to find out feasibility of Exhaust Gas Recirculation (EGR) using biodiesel as a partial substitute of diesel oil in C.I engine to minimize NOx formation. The experimental work under this project consists of two parts, initial experimental work to find out optimum biodiesel - diesel blend and in second phase, experimental work for finding the optimum percentage of EGR for reduction of NOx emission.

Initial experimental work includes preparation of experimental setup, various biodiesel-diesel blends and measurement of various engine parameters by running the engine with different fuel blends through entire load range.

Work in line with the actual objective of the project is planned out as follows,

- Generation of base line performance data from the C.I engine fuelled by diesel, diesel-biodiesel blends at manufacturer injection pressure without and with EGR

- Analysis of the effect of EGR and diesel-biodiesel blends on performance of C I Engine at optimum result values

3.2 Variable Parameters for Experiments

For entire project work, different parameters are varying among their respective range. The variable parameters are fuel composition, Exhaust Gas Recirculation (EGR) percentage and the load condition. Table 3.1 shows all the combination for all the variable parameters. The main thing is fuel composition. The experiment will start with 100 percent diesel as fuel and performed for the various combination of the Biodiesel and Diesel. Similarly, the EGR percentages will also vary from 0 to 15 percent. All the combinations for fuel and EGR will be tested for the entire range of the engine starts from no load condition to full load condition.

Table 3.1: Variable Parameters for Experiment

Variable Parameters	**Range of variations**						
Biodiesel – Diesel blend	Diesel (%)	100	95	90	85	80	75
	Biodiesel (%)	00	05	10	15	20	25
EGR (%)	—	0	5	10	15	—	—
Load (%)	—	0	20	40	60	80	100

3.3 Experimental setup

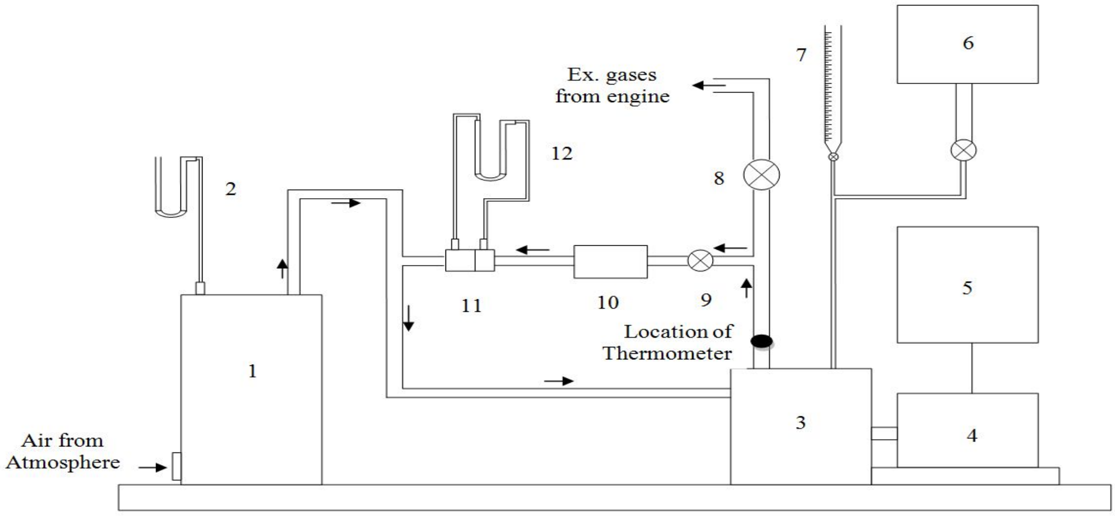

Figure 3.1: Schematic diagram of experimental setup

(1) Air box with orifice

(2) manometer for air flow measurement

(3) engine

(4) A.C. Alternator

(5) load bank

(6) Fuel tank

(7) burette

(8) flow control valve

(9) EGR valve

(10) filter

(11) orifice

(12) manometer for measurement of recirculated gas flow

A single cylinder 4 - stroke air-cooled diesel engine(4) developing 5.5 kW at 1500 rpm was used for experimental work. Engine details along with alternator, are given in Table A1 and A2 of Appendix A. The schematic diagram of the experimental set up is shown in Figure 3.1. An electrical dynamometer(4) was used for loading the engine. A manometer(2) with water as manometric fluid, connected to a large tank(1) of 0.2 m^3 which is 275 times the swept volume of the engine, was attached to the engine to make air flow measurement.Air flow was measured with the help of pressure difference produced by a round, sharp edge orifice of 32 mm diameter. The fuel flow was measured on a volumetric basis using a 50 ml capacity burette(7) and stopwatch.Thermometer with dial indicator (range 0°C to 600°C) was used for measuring the exhaust gas temperature. an electrical load bank(5) is prepared using bulbs of 200 watts and 100 watts capacity. Electrical connections ware made in such way that 20% load interval can be achieved for entire load range. For recirculation of exhaust gases, exhaust control valve(8) and EGR valve(9) are used. Recirculated gases passes through a filter(10). Flow of recirculated gases measured with the help of orifice(11) connected to a manometer(12) with water as manometric fluid.

3.4 Methodology of testing

To insure the repeatability of the experiments, testing of the engine was done as per the Indian standard (IS: 10000 Part IV), which lays down the guidelines for declaring power, efficiency and fuel consumption and specifies relevant correction factors which are required for adjusting the observed readings to the standard reference conditions, as specified in IS: 10000 (Part II).[12]

The standard reference conditions are:

Reference pressure Pr = 100 kPa

Reference temperature Tr = 300 K

Reference relative humidity Φr = 0.6

Power adjustment factor 'α' - is the ratio of power output under the site conditions to the power output under standard reference conditions.

$$P_r = \alpha . P_r \tag{3.1}$$

$$\alpha = k - 0.7(1-k)\left(\frac{1}{\eta_m} - 1\right) \tag{3.2}$$

$$k = \left(\frac{p_x - a.\phi_x.p_{sx}}{p_r - a.\phi_r.p_{sr}}\right)^m \times \left(\frac{T_r}{T_x}\right)^n \tag{3.3}$$

The factor 'a' and exponents 'm' and 'n' have numerical value as given in Figure B5 of Appendix B.

Specific fuel consumption adjustment factor 'β' - is ratio of the specific fuel consumption under site condition to the specific fuel consumption under standard reference condition.

$$(b.s.f.c)_x = \beta.(b.s.f.c)_r \tag{3.4}$$

$$\beta = k/\alpha; \tag{3.5}$$

The power and specific fuel consumption adjustment factors α and β can be calculated using Appendices Figure B1, B2, B3, B4 of IS: 10000 part IV given in Appendix B.

3.5 Statistical analysis

The experiments were conducted six times with base fuels and optimal blends to carry out statistical analysis at 95% confidence level. The procedure of the same was as follows.

- The mean(x_m), deviation$|d_i|$ and standard deviation(S) of the results at different load were calculated by,

$$x_m = \frac{1}{n}\sum_{i=1}^{n} x_i \tag{3.6}$$

$$S = \left[\frac{1}{n}\sum_{i=1}^{n}(x_i - x_m)^2\right]^{1/2} \tag{3.7}$$

$$|d_i| = |x_i - x_m| \tag{3.8}$$

- Chauvenet's criterion was applied to eliminate dubious data points; the deviation of the individual points was then compared with the standard deviation in accordance with the information in Figure B6 of Appendix B, and the dubious points were eliminated. For the final data, presentation a new mean and standard deviation were computed with the dubious points eliminated from the calculation.

- The confidence interval was estimated by Student's t- Distribution

$$\Delta = \frac{tS}{\sqrt{n}} \tag{3.9}$$

Δ = Confidence interval

t= Random variable; values of the same are given in Figure B7 of Appendix B at different degree of freedom, ν = n-1

n= number of observations.

3.6 Uncertainty analysis

The uncertainty in measurement is estimated based on the procedure given by Kline and McClintock (1953). The uncertainty in measurement is defined as

$$\omega_R = \sqrt{\left(\frac{\partial R}{\partial V_1}\omega_1\right)^2 + \left(\frac{\partial R}{\partial V_2}\omega_2\right)^2 + \ldots \left(\frac{\partial R}{\partial V_n}\omega_n\right)^2} \tag{3.10}$$

R is the result of which uncertainty is to be estimated. ω_R is the uncertainty in the result. V_i (i=1 to n) are the variables of which R is a function.

Defining the uncertainty in percentage, the equation modifies to

$$\frac{\omega_R}{R} = \frac{1}{R}\sqrt{\left(\frac{\partial R}{\partial V_1}\omega_1\right)^2 + \left(\frac{\partial R}{\partial V_2}\omega_2\right)^2 + \ldots \left(\frac{\partial R}{\partial V_n}\omega_n\right)^2} \times 100\% \tag{3.11}$$

The uncertainties in the measurement were estimated from the resolution of the instrument or provided by the manufacturer.

The uncertainty in temperature measurement by thermocouple, $\omega_t = \pm 4$ °C.

The uncertainty in voltage measurement by voltmeter, $\omega_v = \pm 1$ volt

The uncertainty in ampere measurement, $\omega_A = \pm 0.1$A

The uncertainty in time measurement, $\omega_t = 0.1$ sec

The uncertainty in volume flow measurement, $\omega_m = 1$ ml

Chapter 4

Results and Discussion

4.1 Test fuel

Jetropha Biodiesel was used as the test fuel in different proportion with diesel. The biodiesel was blended with commercial diesel in 5% (B5D), 10% (B10D), 15% (B15D), 20% (B20D) and 25% (B25D). The properties of diesel were available readily and for Biodiesel, property values were provided by CSMCRI, Bhavnagar from where the biodiesel is procured. The properties of the test fuels are given in Table.

Table 4.1: Properties of Diesel, Biodiesel and Diesel-Biodiesel Blends

Type of fuel	**CV (kJ/kg)**	**Density (kg/m^3)**
Diesel	42500	840
Biodiesel	40160	880
B5D	42383	842
B10D	42266	844
B15D	42149	846
B20D	42032	848
B25D	41915	850

4.2 Properties of Biodiesel and Diesel

This entire project is based on the effect of biodiesel on the NOx formation. Biodiesel required for the experiments will be procured from CSMCRI, Bhavnagar. They have already provided the properties of the biodiesel which they will supply for the project work. Table 3.8 shows the comparison of the important properties for the Biodiesel and Diesel. From comparison we can identify that all the major properties of Biodiesel is very much equivalent to that of diesel. The only point of importance is the Viscosity and the density. The viscosity and density of Biodiesel is higher than Diesel. This is the reason why the conventional diesel engines cannot run directly on 100 percent Biodiesel without any modification.

Table 4.2: Comparison of Biodiesel and Diesel properties

Properties	**Biodiesel**	**Diesel**
Density @ 15°C(kg / m^3)	880.0	840.0
Viscosity at 40°C (mm^2 / s)	4.34	3.00
Flash point (°C)	160	49
Cetane number	54.5	52.9
Gross calorific value (Kcal/kg)	9562	10700

4.3 Variation in EGR percentages

Exhaust gas coming out from the engine is mixed with the air supplied to the engine. The air supplied to the C I Engine is considered as 100%. The amount of recirculation of the exhaust gas varied from 0 to 15% with the interval of 5% i.e. 0%, 5%, 10% and 15%.

4.4 Test procedure

Constant speed engine test were carried out at different loads on the engine and the following performance and emission were analyzed.

- Brake specific fuel consumption. Kg/kwh
- Break specific energy consumption, kJ/kWh
- Brake thermal efficiency, %
- Exhaust gas temperature, °C
- Oxide of nitrogen emission, ppm

4.5 Experimental Results

4.5.1 Effect of biodiesel-diesel blend on Break Specific Fuel Consumption

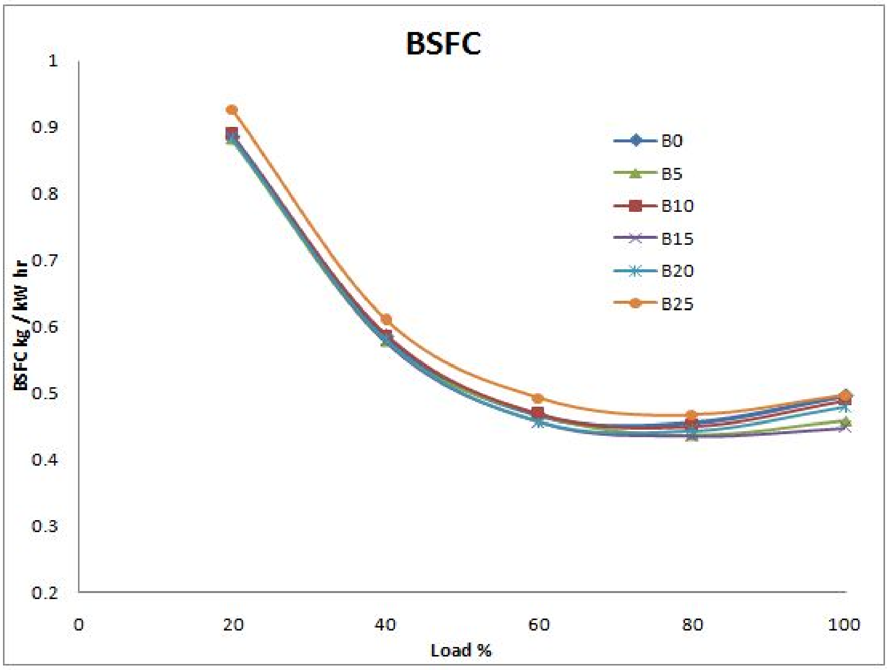

Figure 4.1: Effect of Biodiesel blends on BSFC

Figure 4.1 shows the brake specific fuel consumption (BSFC) variation for the diesel and diesel-biodiesel blends with respect to engine loads. In general, the BSFC values of the blend were slightly higher than those of the base fuel i.e at all loads, more fuel is needed, with the blending of biodiesel with diesel, to produce the same amount of energy due to its lower heating value in comparison to the base fuel. However, for B10 the specific fuel consumption was nearly equal to the values for diesel. For B5, at part load condition, the BSFC decreased as compare to diesel by 1.51%. For B25, BSFC increased noticeably by 5% due to decrease in calorific value of the fuel. The common observation was that the brake specific fuel consumption of the engine, operating on any test fuels decreased with increasing load. Higher mechanical efficiency at high load explained such trends in the BSFC.[13]

4.5.2 Effect of biodiesel-diesel blend on Break Specific Energy Consumption

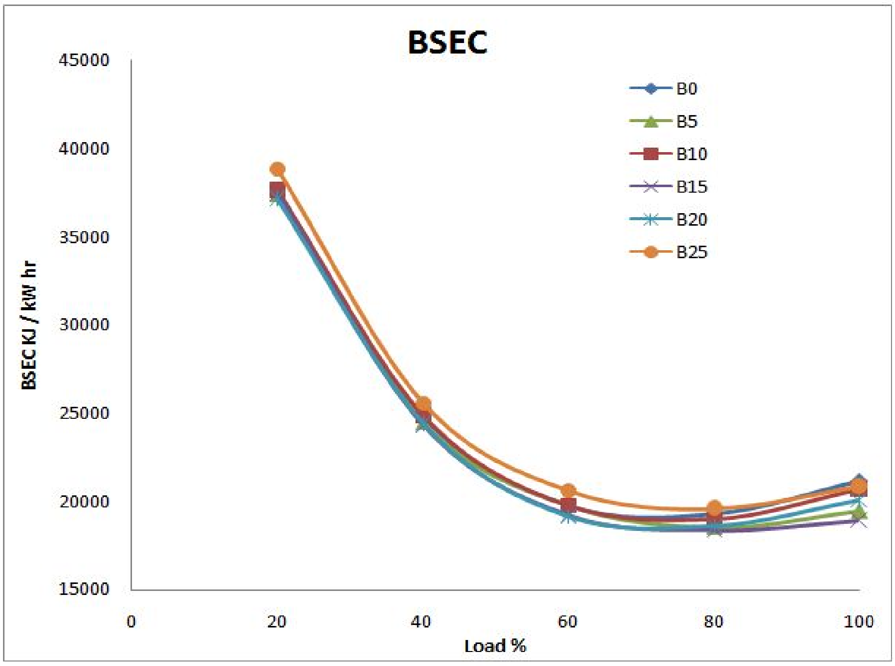

Figure 4.2: Effect of Biodiesel-diesel blends on BSEC

Figure 4.2 shows the brake specific energy consumption (BSEC) variation for the diesel and diesel-biodiesel blends with respect to engine loads. In general, the BSEC values of the blend were slightly lower than those of the base fuel. For B10, the specific energy consumption was nearly equal to the values for pure diesel. For B5, at part load condition, the BSEC decreased as compare to diesel. For B25, BSEC increased.

4.5.3 Effect of biodiesel-diesel blends on brake thermal efficiency

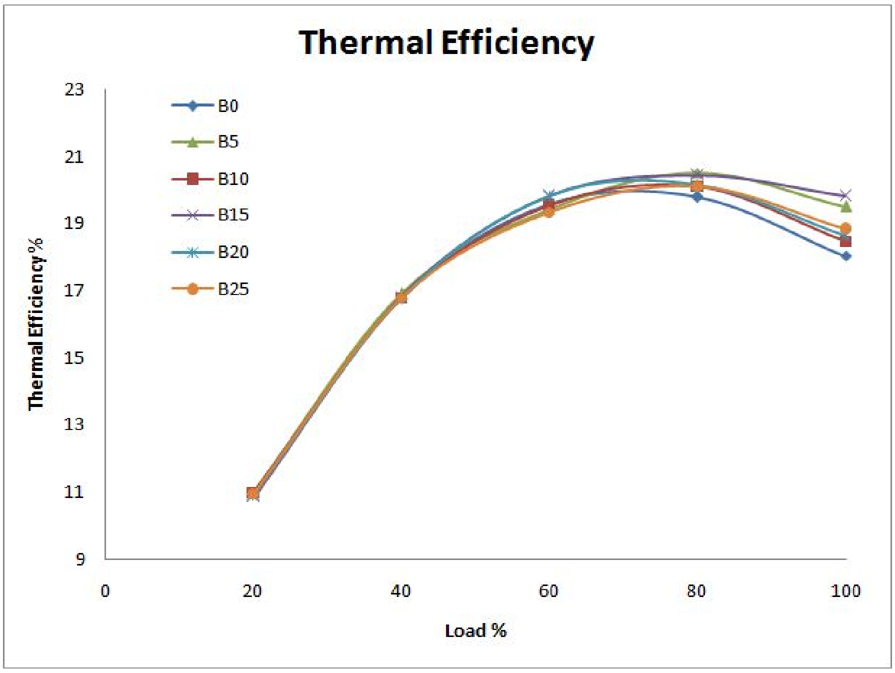

Figure 4.3: Effect of Biodiesel-diesel blends on brake thermal efficiency

The thermal efficiency distribution is shown in Figures 4.3 for diesel and diesel-biodiesel blends. With diesel-biodiesel blends brake thermal efficiency shows an increase with biodiesel addition at higher loads is due to improvement of diffusive combustion phase on account of oxygen enrichment. Based on these reasons the energy consumption rate of blends decreases. With 10% biodiesel addition to diesel, around 2.4% rise in thermal efficiency can be achieved as compared to diesel for almost entire range of load. Although the BSFC values of the blends at partial loads were slightly higher than those with the diesel, the thermal efficiencies were slightly higher than diesel due to higher oxygen content in the chemical composition of the fuel.[13]

4.5.4 Effect of biodiesel-diesel blends on exhaust gas temperature

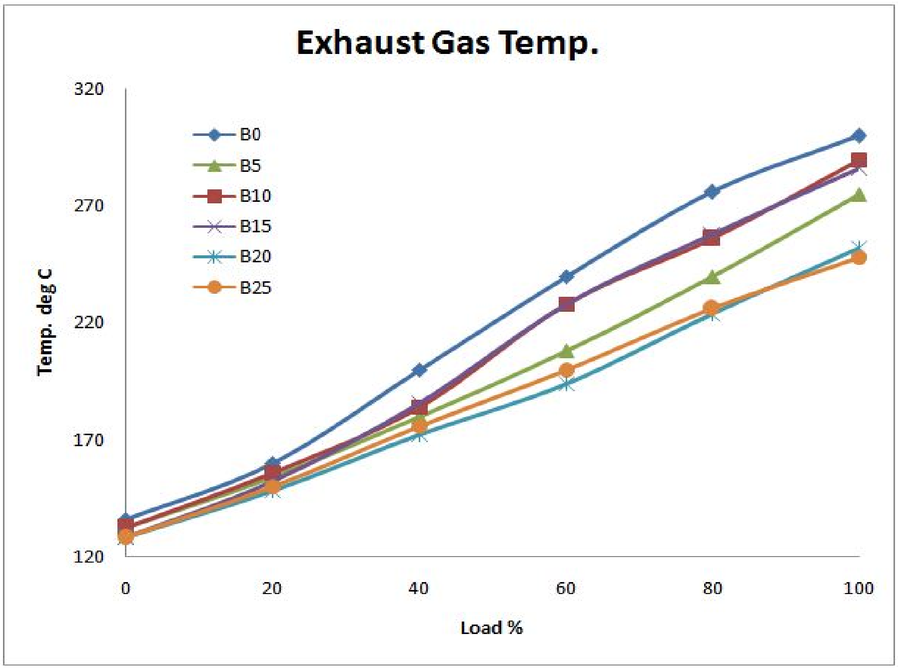

Figure 4.4: Effect of Biodiesel-diesel blends on exhaust gas temperature

The variation of exhaust temperature with respect to applied load for different fuels tested is shown in Figures 4.4. The biodiesel contains some amount (11% by weight) of oxygen molecules; it also takes part in the combustion. When biodiesel concentration was increased, the exhaust gas temperature slightly decreased. The nitrogen oxide emission is directly related to the engine combustion temperature, which in turn is indicated by the prevailing exhaust gas temperature. Here With decrease in temperature, NOx emission should decrease but higher availability of oxygen at the time of combustion leads to higher formation of NOx. For diesel, the exhaust gas temperature is very high for entire range of load. As the biodiesel contain increased in the fuel, the temperature decreased.

4.5.5 Effect of biodiesel-diesel blends on carbon dioxide emission

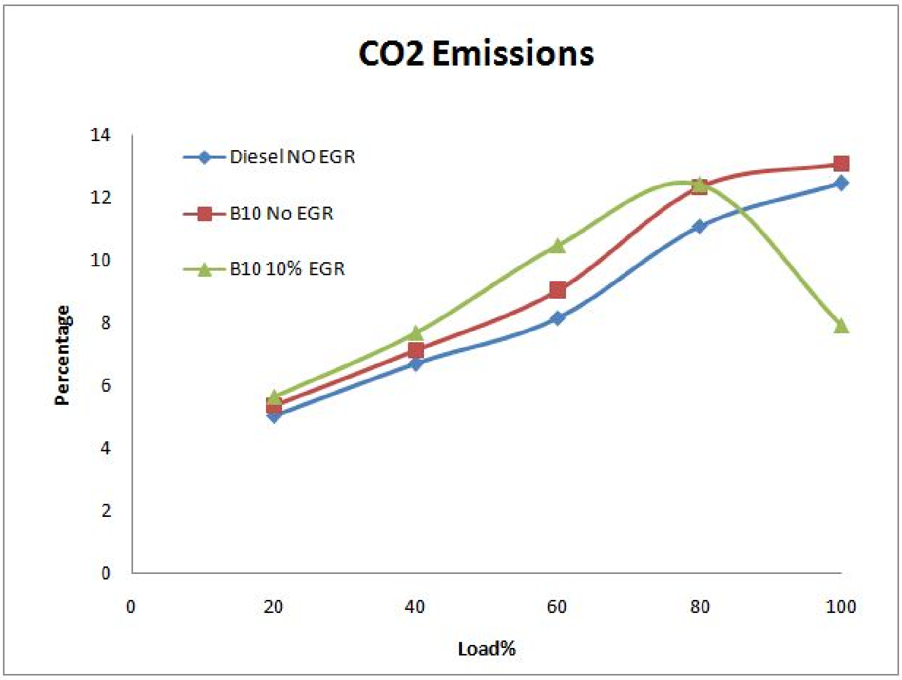

Figure 4.5: Effect of Biodiesel-diesel blends on carbon dioxide emission

Figure 4.5 shows the variation of CO2 emission with engine load for different fuels and with and without EGR. It is known that amount of carbon dioxide emitted is proportional to the amount of fuel burned. At low load as BSFC is more, the amount of CO2 emission is less (in percentage) at low load compare to higher load. In addition, at higher load, the CO2 emissions with the 10% biodiesel blend was 12.16% higher than those with base fuels due to the increase in the mass of fuel injected using the blends. Higher CO2 in the exhaust emission is an indication of the complete combustion of fuel. The carbon monoxide emissions were found negligible at lower load and than increase with increase in load. This is typical with all internal combustion engines since at low load the air fuel ratio is too lean and air fuel ratio decreases with increase in load. It is interesting to note that, the engine emitted

5.35% more CO with blends compared to base fuels at higher load. It is because the presence of oxygen molecules in biodiesel makes the air fuel mixture too lean resulting in more CO emission.

4.5.6 Effect of Exhaust Gas Recirculation on Break Specific Fuel Consumption

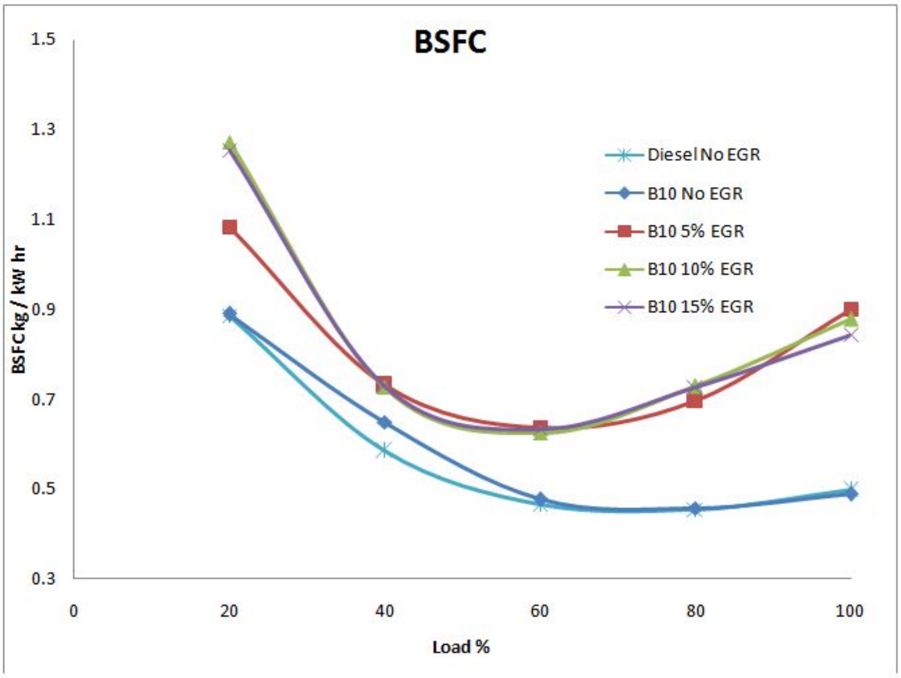

Figure 4.6: Effect of Exhaust Gas Recirculation on BSFC

Figure 4.6 indicates the change in break specific fuel consumption for various EGR percentage. For Diesel the fuel consumption is minimum. When biodiesel diesel blend is used, the fuel consumption is slightly increased. For entire range of EGR percentage, the BSFC is comparatively higher from Diesel and Biodiesel-diesel blend without EGR. In general the 12.38% of rise is noticed for 10% EGR as compare to 0% EGR for biodiesel-diesel blend.

4.5.7 Effect of Exhaust Gas Recirculation on Break Specific Energy Consumption

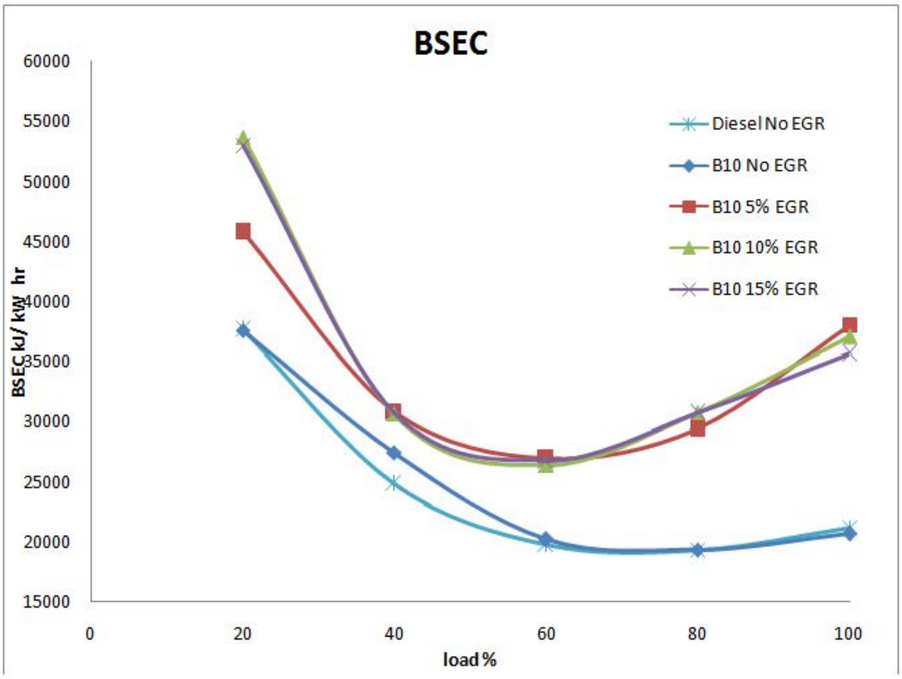

Figure 4.7: Effect of Exhaust Gas Recirculation on BSEC

Break Specific Energy Consumption for the various EGR percentages is shown in Figure 4.7. As BSEC is directly depends to the BSFC, the nature of the plot for BSEC is similar to the plot of BSFC. Higher BSEC is noticed for the various EGR percentages as compare to the engine tests carried out without EGR.

4.5.8 Effect of Exhaust Gas Recirculation on Break Power

Figure 4.8 shows the effect of EGR on the break power of the engine fueled with 10% biodiesel diesel blend. It clearly indicates that the power produced is higher for diesel for which engine is designed. Break power is reduced by 2.88% while using

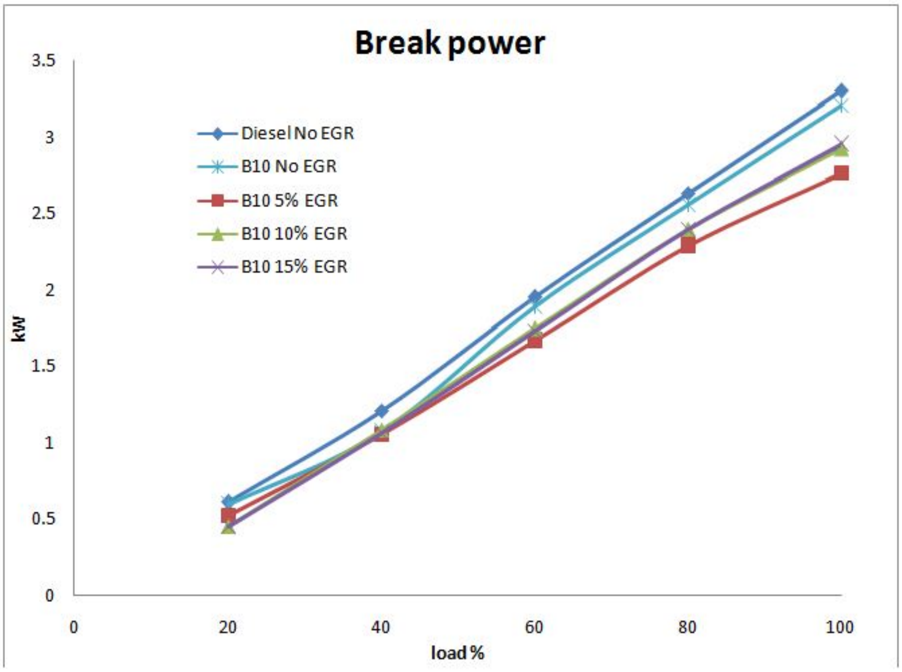

Figure 4.8: Effect of Exhaust Gas Recirculation on Break Power

10% biodiesel diesel blend due to decrease in calorific value of the fuel. When 10% EGR is applied, the power is reduced by 8.08% in comparision with no EGR for 10% biodiesel blend.

4.5.9 Effect of Exhaust Gas Recirculation on Exhaust Gas Temperature

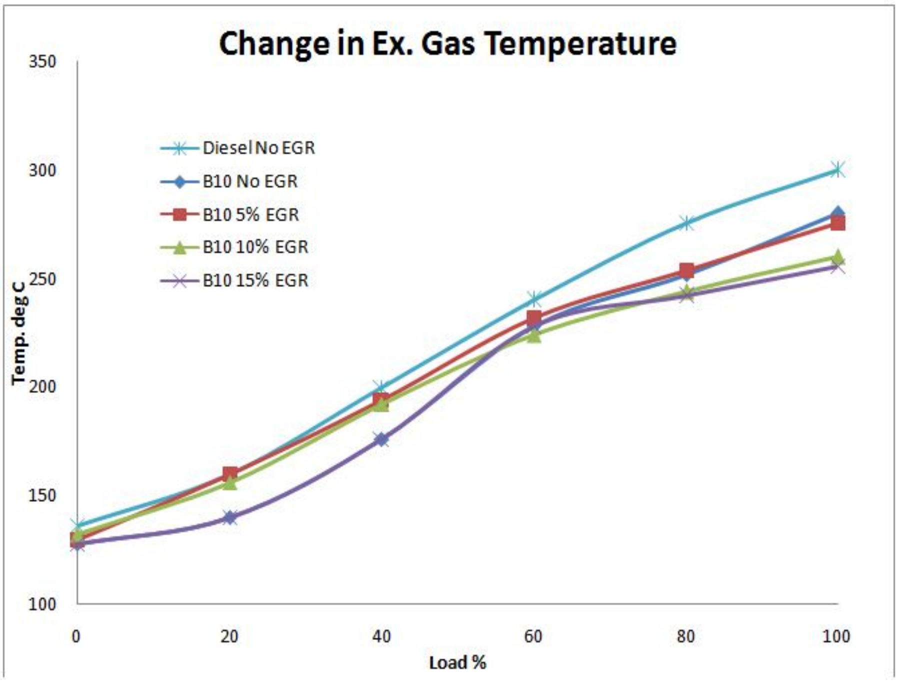

Figure 4.9: Effect of Exhaust Gas Recirculation on Exhaust Gas Temperature

The variation of exhaust gas temperature with respect to applied load for different percentage of EGR is shown in Figure 4.9 for diesel and diesel-biodiesel blend. When exhaust gases coming out from the engine cylinder are recirculated by mixing with the inlet air, the overall cycle temperature comes down. For diesel and Biodiesel both, the exhaust gas temperature is higher at no EGR as compare to 10% and 15% EGR.

4.5.10 Effect of biodiesel-diesel blends on oxides of nitrogen emission

Figures 4.10, 4.11, 4.12 and 4.13 indicates the oxide of nitrogen emission trends for diesel and biodiesel blends at different engine loads with and without EGR. There are mainly three factors which effect the NOx emissions oxygen concentration, combustion temperature, and reaction time. From Fig. 4.10, The NOx emission with diesel-biodiesel blends were observed more compared to diesel.From Figure 4.10, 4.11, 4.12, 4.13 it is clearly evident that NOx emission increase by around 62% with addition with biodiesel in diesel. As increases concentration biodiesel increases availability of oxygen. At a same time addition of biodiesel makes the combustion homogeneous hence there is chance to increase in NOx emission in comparison with diesel fuel.
It is also observed that NOx emission decreases with increase in % of EGR in comparison with base line fuel without EGR. The reason for the same may be reduction in concentration of external oxygen with increasing the % of EGR. The external oxygen supplied with air is mainly responsible for NOx emission, As with EGR, this concentration is decreasing, NOx emission reduces.

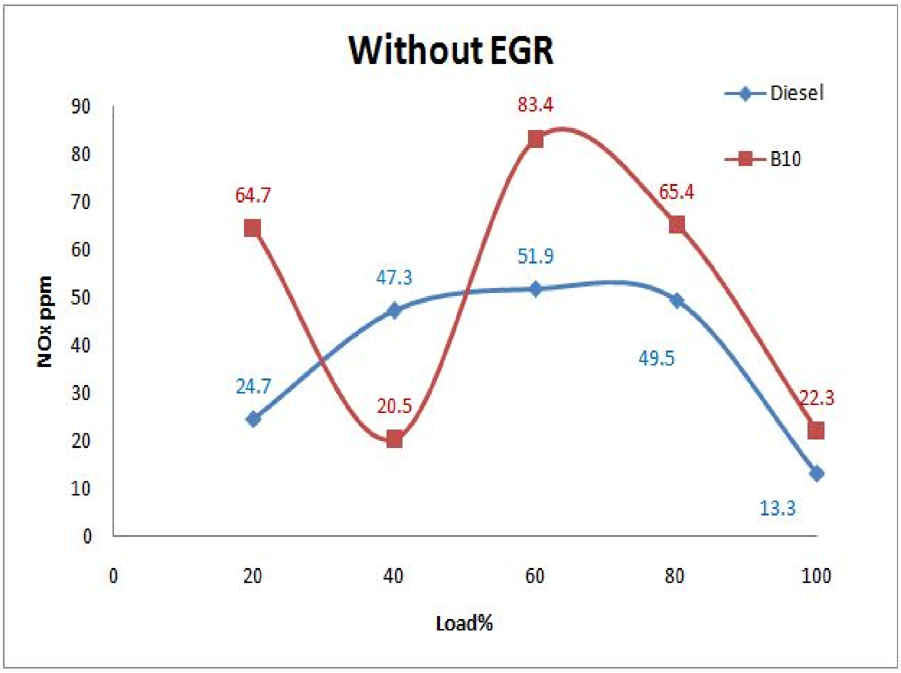

Figure 4.10: Effect of Biodiesel on NOx emission

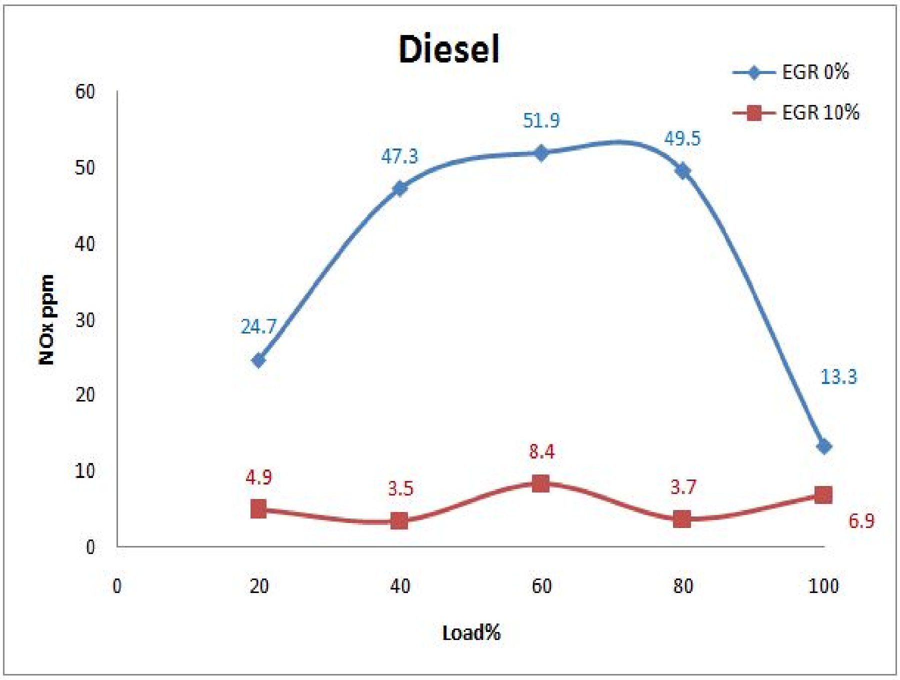

Figure 4.11: Effect of EGR on NOx emission (Diesel)

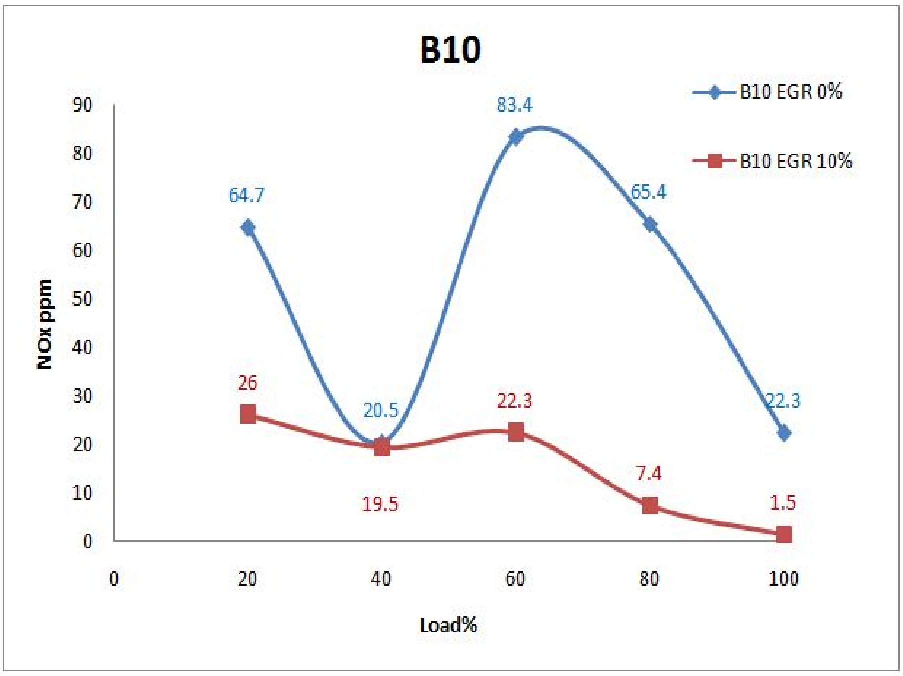

Figure 4.12: Effect of EGR on NOx emission (Diesel-biodiesel blend)

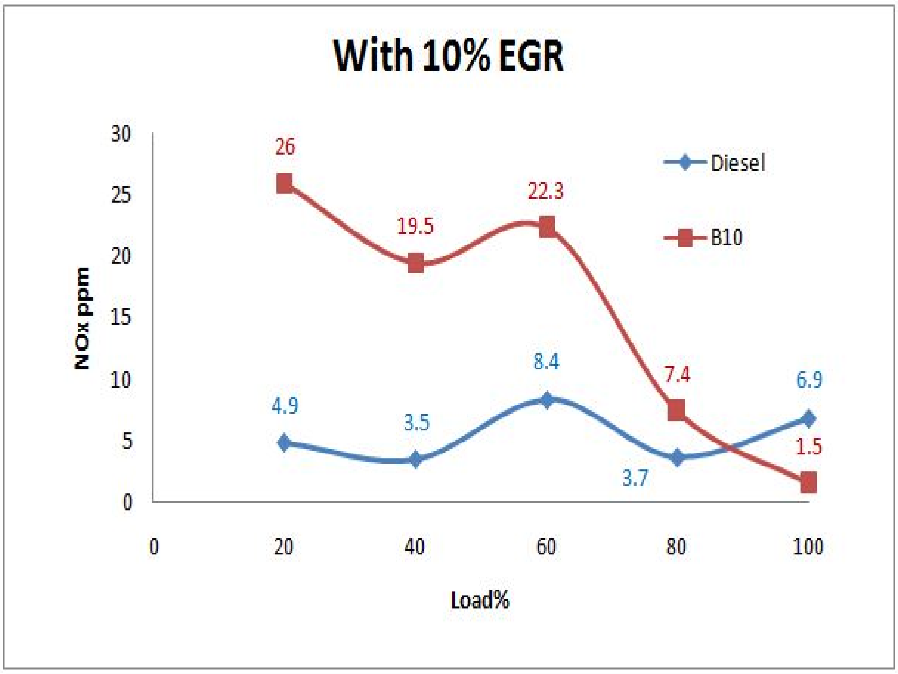

Figure 4.13: Effect of EGR on NOx

Chapter 5

Conclusion

This entire work is concentrated around the minimum formation of NOx from a C I Engine fueled with Biodiesel Diesel blend with the help of Exhaust Gas Recirculation. Biodiesel is more viscous (4.34 mm^2/sec) as compare to diesel (3.0 mm^2/sec). For 10% Biodiesel-Diesel blend(volume basis) the density become 3.134 mm^2/sec. For higher blends of biodiesel, the modification in injection system of engine is required due to increase in viscosity of fuel.

Calorific value of Biodiesel is less (40160 kJ/kgk) as compare to diesel (42500 kJ/kgk). Decrese in calorific value results in higher consumption of fuel for biodiesel-diesel blend as compare to diesel.

From performance test analysis, it is evident that when 10% biodiesel-diesel blend is used, the BSFC is decreased by around 1%, BSEC is increased by 1.15%, Thermal efficiency of engine is increased by 1.53% and exhaust gas temperature is reduced by 7.24% at part load condition. By considering all these performance parameters, it was concluded that the 10% biodiesel (on volume basis) can be mixed with diesel as an optimum blend.

The formation of NOx emission becomes higher when biodiesel-diesel blend is used. This emission can be reduced by using EGR. For engine operating with 10% biodiesel-diesel blend and EGR, the BSFC is increased by 30% and BSEC is increased by 32% at part load condition.It was difficult to find the optimum % of EGR. To find the effect of EGR on NOx emission, analysis of the exhaust gas was carried out. The results of gas analysis shows that, when 10% biodiesel-Diesel blend is used, the Nox emission increased by 63% and When 10% EGR is implemented to the cycle, the NOx emission is decreased by 92% and 70% for diesel and biodiesel-diesel blend respectively at part load condition.86% reduction in NOx emission has been achieved by 10% EGR for optimized fuel blend at part load condition.

5.1 Possibility of work in Future

Work carried out under this project can include some more parameters for further discussion and experimentation.

Combustion analysis can be carried out for different fuel blends and EGR percentage to find out the variation in different parameters like maximum cycle pressur, maximum cycle temperature, combustion delay period etc.

Modification in fuel injection system is required when biodiesel-diesel blends with higher percentage of biodiesel will be supplied. After modification, the experiment can be performed for 100% biodiesel also. Change in injection timing is also required with modification in fuel injection system.

Effect of cooling of exhaust gases can be analyzed by using an appropriate heat exchanger which works as EGR cooler.

A rigorous analysis of exhaust gases is required at all the combination of variable parameters using 5 gas analyzer.

References

[1] J. B. Heywood, *Internal combustion engine fundamentals.* New York: McGraw Hill, 5 ed., 1988.

[2] H. E. Saleh, "Effect of exhaust gas recirculation on diesel engine nitrogen oxide reduction operating with jojoba methyl ester," *Renuable Energy*, vol. 34, pp. 2178–2186, 2009.

[3] V. Pradeep and R. P. Sharma, "Use of hot egr for nox control in a compression ignition engine fuelled with bio-diesel from jatropha oil," *Renuable Energy*, vol. 32, pp. 1136–1154, 2007.

[4] G. H. Abd-Alla, "Using exhaust gas recirculation in internal cobustion engines: a review," *Energy Conversion and Management*, vol. 43, pp. 1027–1042, 2002.

[5] M. Zheng, G. T. Reader, and J. GaryHawley, "Diesel engine exhaust gas recirculationa review on advanced and novel concepts," *Energy Conversion and Management*, vol. 45, pp. 883–900, 2004.

[6] M. Lapuerta, O. Armas, and J. R. Fernandez, "Effect of biodiesel fuels on diesel engine emissions," *Progress in Energy and combustion science*, vol. 34, pp. 198–223, 2008.

[7] S. K. Mahlaa, L. Das, and M. Babu, "Effect of cooled egr on performance and exhaust emission characteristics of biodiesel fueled engine," in *Proceedings of the Third International Conference on Thermal Engineering: Theory and Applications*, (Amman, Jordan), May 2007.

[8] G. S. Hebbar and A. K. Bhat, "Analysis of preformance and emission of a naturally aspirated stationary di diesel engine with exhaust gas recirculation," *Progress in Energy and combustion science.*

[9] H. Chen, J. Wang, S. Shuai, and W. Chen, "Study of oxygenated biomass fuel blends on a diesel engine," *Fuel*, vol. 87, pp. 3462–3468, 2008.

[10] N. K. Millerjothi, G. Nagarajan, and S. Renganarayanan, "Lpg fueled diesel engine using diethyl ether with exhaust gas recirculation," *International Journal of Thermal Sciences*, vol. 47, pp. 450–457, 2008.

[11] A. Ibrahim and S. Bari, "An experimental investigation on the use of egr in a supercharged natural gas si engine," *Fuel*, Artical in press.

[12] J. P. Holman, *Experimental methods for engineers.* New York: McGraw Hill, 5 ed., 1989.

[13] A. M. Lakdawala, "To identify patential of dee as supplimentry fuel for c i engine," Master's thesis, Indian Institute of Technology, Delhi, 2007.

Appendix A

Table A.1: Specifications for Diesel Engine

Make	Markon
No. of cylinder	One
No. of cycle strokes	Four
Bore x stroke	92 mm x 110 mm
Power	5 kW
Speed	1500 rpm
Electric start	
Air cooled	

Table A.2: Specifications for A.C. Generator

Make	Markon
Single phase	240 volts, 50 Hz
Rated output	5 kW/5 kVA
Speed	1500 rpm
Direct coupled with Diesel engine	
Engine and generator both mounted on a single frame	

Appendix B

IS : 10000 (Part IV) - 1980

APPENDIX A

(*Clauses* 2.3.1 *and* H-1.7)

DETERMINATION OF THE POWER ADJUSTMENT FACTOR (α)

A-1. The table below gives values of the power adjustment factor (α) for known values of the ratio of indicated power (k) and mechanical efficiency (η_m).

A-2. The value of (k) can be determined from Appendix B.

A-3. The value of η_m is stated by the manufacturer (*see* **2.3**, Note 3).

k	α					
	η_m					
	0·70	0·75	0·80	0·85	0·90	0·95
0·50	0·350	0·383	0·413	0·438	0·461	0·482
0·52	0·376	0·408	0·436	0·461	0·483	0·502
0·54	0·402	0·433	0·460	0·483	0·504	0·523
0·56	0·428	0·457	0·483	0·506	0·526	0·544
0·58	0·454	0·482	0·507	0·528	0·547	0·565
0·60	0·480	0·507	0·530	0·551	0·569	0·585
0·62	0·506	0·531	0·554	0·573	0·590	0·606
0·64	0·532	0·556	0·577	0·596	0·612	0·627
0·66	0·558	0·581	0·601	0·618	0·634	0·648
0·68	0·584	0·605	0·624	0·641	0·655	0·668
0·70	0·610	0·630	0·648	0·663	0·677	0·689
0·72	0·636	0·655	0·671	0·685	0·698	0·710
0·74	0·662	0·679	0·695	0·708	0·720	0·730
0·76	0·688	0·704	0·718	0·730	0·741	0·751
0·78	0·714	0·729	0·742	0·753	0·763	0·772
0·80	0·740	0·753	0·765	0·775	0·784	0·793
0·82	0·766	0·778	0·789	0·798	0·806	0·813
0·84	0·792	0·803	0·812	0·820	0·828	0·834
0·86	0·818	0·827	0·836	0·843	0·849	0·855
0·88	0·844	0·852	0·859	0·865	0·871	0·876
0·90	0·870	0·877	0·883	0·888	0·892	0·896
0·92	0·896	0·901	0·906	0·910	0·914	0·917
0·94	0·922	0·926	0·930	0·933	0·935	0·938
0·96	0·948	0·951	0·953	0·955	0·957	0·959
0·98	0·974	0·975	0·977	0·978	0·978	0·979
1·00	1·000	1·000	1·000	1·000	1·000	1·000
1·02	1·026	1·025	1·024	1·023	1·022	1·021
1·04	1·052	1·049	1·047	1·045	1·043	1·042
1·06	1·078	1·074	1·071	1·067	1·065	1·062
1·08	1·104	1·099	1·094	1·090	1·086	1·083
1·10	1·130	1·123	1·118	1·112	1·108	1·104
1·12	1·156	1·148	1·141	1·135	1·129	1·124
1·14	1·182	1·173	1·165	1·157	1·151	1·145
1·16	1·208	1·197	1·188	1·180	1·172	1·166
1·18	1·234	1·222	1·212	1·202	1·194	1·187
1·20	1·260	1·247	1·235	[illegible]	1·216	1·207

Figure B.1:

APPENDIX B

(*Clauses* 2.3.1, 3.2.2, A-2, F-2 *and* H-1.4)

DETERMINATION OF THE RATIO OF INDICATED POWER (k)

B-1. Formula (3) in **2.3** can be written as:

$$k = (R_1)^{y_1} (R_2)^{y_2} (R_3)^{y_3}$$

where

$$R_1 = \frac{p_x - a\,\phi_x . p_{sx}}{p_r - a\,\phi_r . p_{sr}},$$

$$R_2 = \frac{T_r}{T_x}, \text{ and}$$

$$R_3 = \frac{T_{cr}}{T_{cx}}$$

and $y_1 = m$, $y_2 = n$, $y_3 = q$

B-2. The value of R (dry air pressure ratio) $= \frac{p_x - a.\phi_x.p_{sx}}{p_r - a.\phi_r.p_{sr}}$ can be obtained from Appendix C and other values of R can be calculated.

B-3. The values of m, n, q are obtained from Table 1.

B-4. The table below then gives values of R^y for known ratios R and known factors y.

B-5. The value of k is then obtained by multiplying together the appropriate values of R^y.

R	R^y								
	y								
	0·5	0·55	0·57	0·7	0·75	0·86	1·2	1·75	2
0·60	0·775	0·755	0·747	0·699	0·682	0·645	0·542	0·409	0·360
0·62	0·787	0·769	0·762	0·716	0·699	0·663	0·564	0·433	0·384
0·64	0·800	0·782	0·775	0·732	0·715	0·681	0·585	0·458	0·410
0·66	0·812	0·796	0·789	0·748	0·732	0·700	0·607	0·483	0·436
0·68	0·825	0·809	0·803	0·763	0·749	0·718	0·630	0·509	0·462
0·70	0·837	0·822	0·816	0·779	0·765	0·736	0·652	0·536	0·490
0·72	0·849	0·835	0·829	0·795	0·782	0·754	0·674	0·563	0·518
0·74	0·860	0·847	0·842	0·810	0·798	0·772	0·697	0·590	0·548
0·76	0·872	0·860	0·855	0·825	0·814	0·790	0·719	0·619	0·578
0·78	0·883	0·872	0·868	0·840	0·830	0·808	0·742	0·647	0·608
0·80	0·894	0·885	0·881	0·855	0·846	0·825	0·765	0·677	0·640
0·82	0·906	0·897	0·893	0·870	0·862	0·843	0·788	0·707	0·672
0·84	0·917	0·909	0·905	0·885	0·877	0·861	0·811	0·737	0·706
0·86	0·927	0·920	0·918	0·900	0·893	0·878	0·834	0·768	0·740
0·88	0·938	0·932	0·930	0·914	0·909	0·896	0·858	0·800	0·774
0·90	0·949	0·944	0·942	0·929	0·924	0·913	0·881	0·832	0·810
0·92	0·959	0·955	0·954	0·943	0·939	0·931	0·905	0·864	0·846
0·94	0·970	0·967	0·965	0·958	0·955	0·948	0·928	0·897	0·884
0·96	0·980	0·978	0·977	0·972	0·970	0·966	0·952	0·931	0·922
0·98	0·990	0·989	0·989	0·986	0·985	0·983	0·976	0·965	0·960
1·00	1·000	1·000	1·000	1·000	1·000	1·000	1·000	1·000	1·000
1·02	1·010	1·011	1·011	1·014	1·015	1·017	1·024	1·035	1·040
1·04	1·020	1·022	1·023	1·023	1·030	1·034	1·048	1·071	1·082
1·06	1·030	1·033	1·034	1·042	1·045	1·051	1·072	1·107	1·124
1·08	1·039	1·043	1·045	1·055	1·059	1·068	1·097	1·144	1·166
1·10	1·049	1·054	1·056	1·069	1·074	1·085	1·121	1·182	1·210
1·12	1·058	1·064	1·067	1·083	1·089	1·102	1·146	1·219	1·254
1·14	1·068	1·075	1·078	1·096	1·103	1·119	1·170	1·258	1·300
1·16	1·077	1·085	1·088	1·110	1·118	1·136	1·195	1·297	1·346
1·18	1·086	1·095	1·099	1·123	1·132	1·153	1·220	1·336	1·392
1·20	1·095	1·106	1·110	1·135	1·147	1·170	1·245	1·376	1·440

Figure B.2:

APPENDIX C

(*Clauses* 2.3.1, 3.2.2, A-2 *and* H-1.3)

DETERMINATION OF DRY AIR PRESSURE RATIO

C-1. The dry air pressure ratio $\frac{p_x - a.\phi_x.p_{sx}}{p_r - a.\phi_r.p_{sr}}$ used in formula (3) in **2.3** is given in the table below for the value of $a = 1$ of formula references A, E and G, and for different values of total barometric pressure (p_x) and water vapour pressure ($\phi_x.p_{sx}$).

C-1.1 If the water vapour pressure is not known it can be obtained from the air temperature and relative humidity by the use of Appendix D.

Altitude	Barometric Pressure p_x	$\frac{p_x - a.\phi_x.p_{sx}}{p_r - a.\phi_r.p_{sr}}$													
		$\phi_x.p_{sx}$ (kPa)													
m	kPa	0	1	2	3	4	5	6	7	8	9	10	11	12	13
0	101·3	1·04	1·02	1·01	1·00	0·99	0·98	0·97	0·96	0·95	0·94	0·93	0·92	0·91	0·90
100	100·0	1·02	1·01	1·00	0·99	0·98	0·97	0·96	0·95	0·94	0·93	0·92	0·91	0·90	0·89
200	98·9	1·01	1·00	0·99	0·98	0·97	0·96	0·95	0·94	0·93	0·92	0·91	0·90	0·89	0·88
400	96·7	0·99	0·98	0·97	0·96	0·95	0·94	0·93	0·92	0·91	0·90	0·89	0·88	0·87	0·86
600	94·4	0·96	0·95	0·94	0·93	0·92	0·91	0·90	0·89	0·88	0·87	0·86	0·85	0·84	0·83
800	92·1	0·94	0·93	0·92	0·91	0·90	0·89	0·88	0·87	0·86	0·85	0·84	0·83	0·82	0·81
1 000	89·9	0·92	0·91	0·90	0·89	0·88	0·87	0·86	0·85	0·84	0·83	0·82	0·81	0·80	0·79
1 200	87·7	0·90	0·89	0·88	0·87	0·86	0·85	0·84	0·82	0·81	0·80	0·79	0·78	0·77	0·76
1 400	5·6	0·87	0·86	0·85	0·84	0·83	0·82	0·81	0·80	0·79	0·78	0·77	0·76	0·75	0·74
1 600	3·5	0·85	0·84	0·83	0·82	0·81	0·80	0·79	0·78	0·77	0·76	0·75	0·74	0·73	0·72
1 800	1·5	0·83	0·82	0·81	0·80	0·79	0·78	0·77	0·76	0·75	0·74	0·73	0·72	0·71	0·70
2 000	9·5	0·81	0·80	0·79	0·78	0·77	0·76	0·75	0·74	0·73	0·72	0·71	0·70	0·69	0·68
2 200	77·6	0·79	0·78	0·77	0·76	0·75	0·74	0·73	0·72	0·71	0·70	0·69	0·68	0·67	0·66
2 400	75·6	0·77	0·76	0·75	0·74	0·73	0·72	0·71	0·70	0·69	0·68	0·67	0·66	0·65	0·64
2 600	73·7	0·75	0·74	0·73	0·72	0·71	0·70	0·69	0·68	0·67	0·66	0·65	0·64	0·63	0·62
2 800	71·9	0·73	0·72	0·71	0·70	0·69	0·68	0·67	0·66	0·65	0·64	0·63	0·62	0·61	0·60
3 000	70·1	0·72	0·71	0·70	0·69	0·68	0·67	0·66	0·64	0·63	0·62	0·61	0·60	0·59	0·58
3 200	68·4	0·70	0·69	0·68	0·67	0·66	0·65	0·64	0·63	0·62	0·61	0·60	0·59	0·58	0·57
3 400	66·7	0·68	0·67	0·66	0·65	0·64	0·63	0·62	0·61	0·60	0·59	0·58	0·57	0·56	0·55
3 600	64·9	0·66	0·65	0·64	0·63	0·62	0·61	0·60	0·59	0·58	0·57	0·56	0·55	0·54	0·53
3 800	63·2	0·65	0·64	0·63	0·62	0·60	0·59	0·58	0·57	0·56	0·55	0·54	0·53	0·52	0·51
4 000	61·5	0·63	0·62	0·61	0·60	0·59	0·58	0·57	0·56	0·55	0·54	0·53	0·52	0·51	0·50
4 200	60·1	0·61	0·60	0·59	0·58	0·57	0·56	0·55	0·54	0·53	0·52	0·51	0·50	0·49	0·48
4 400	58·5	0·60	0·59	0·58	0·54	0·56	0·55	0·54	0·53	0·52	0·51	0·50	0·49	0·48	0·47
4 600	56·9	0·58	0·57	0·56	0·55	0·54	0·53	0·52	0·51	0·50	0·49	0·48	0·47	0·46	0·45
4 800	55·3	0·57	0·55	0·54	0·53	0·52	0·51	0·50	0·49	0·48	0·47	0·46	0·45	0·44	0·43
5 000	54·1	0·55	0·54	0·53	0·52	0·51	0·50	0·49	0·48	0·47	0·46	0·45	0·44	0·43	0·42

Figure B.3:

IS : 10000 (Part IV) - 1980

APPENDIX D

(*Clauses* 2.3.1, 3.2.2, C-1.1 *and* H-1.2)

DETERMINATION OF WATER VAPOUR PRESSURE

D-1. The water vapour pressure ($\phi_x . p_{sx}$) is given in the table below for different values of the air temperature t_x in degrees Celsius and relative humidity ϕ_x.

t_x (°C)	$\phi_x . p_{sx}$ (kPa)				
	ϕ_x				
	1	0·8	0·6	0·4	0·2
−10	0·3	0·2	0·2	0·1	0·1
−5	0·4	0·3	0·2	0·2	0·1
0	0·6	0·5	0·4	0·2	0·1
5	0·9	0·7	0·5	0·4	0·2
10	1·2	1	0·7	0·5	0·2
15	1·7	1·4	1	0·7	0·3
20	2·3	1·9	1·4	0·9	0·5
25	3·2	2·5	1·9	1·3	0·6
27	3·6	2·9	2·1	1·4	0·7
30	4·2	3·4	2·5	1·7	0·8
32	4·8	3·8	2·9	1·9	1
34	5·3	4·3	3·2	2·1	1·1
36	6	4·8	3·6	2·4	1·2
38	6·6	5·3	4	2·7	1·3
40	7·4	5·9	4·4	3	1·5
42	8·2	6·6	4·9	3·3	1·6
44	9·1	7·3	5·5	3·6	1·8
46	10·1	8·1	6·1	4	2
48	11·2	8·9	6·7	4·5	2·2
50	12·3	9·9	7·4	4·9	2·5

Figure B.4:

Engine Type	Condition		Factor	Exponents	
			a	m	n
Compression ignition oil and dual fuel engine	Non-turbo charged	Power limited by air excess	1	1	0.75
		Power limited by thermal reasons	0	1	1
Spark ignition engines using gaseous fuel	Non turbo charged		1	0.86	0.55
Spark ignition engines using liquid fuel	Naturally aspirated		1	1	0.5

Figure B.5: Values of constant used in equation 3.3

Number of Readings, n	Ratio of Maximum Acceptable Deviation to Standard Deviation, d_{max}/σ
3	1.38
4	1.54
5	1.65
6	1.73
7	1.80
10	1.96
15	2.13
25	2.33
50	2.57
100	2.81
300	3.14
500	3.29
1000	3.48

Figure B.6: Chauvenet's criterion for rejecting a reading

Subscript designates percent confidence level.

Degrees of freedom ν	t_{50}	t_{80}	t_{90}	t_{95}	t_{98}	t_{99}	$t_{99.9}$
1	1.000	3.078	6.314	12.706	31.821	63.657	636.619
2	0.816	1.886	2.920	4.303	6.965	9.925	31.598
3	0.765	1.638	2.353	3.182	4.541	5.841	12.941
4	0.741	1.533	2.132	2.776	3.747	4.604	8.610
5	0.727	1.476	2.015	2.571	3.365	4.032	6.859
6	0.718	1.440	1.943	2.447	3.143	3.707	5.959
7	0.711	1.415	1.895	2.365	2.998	3.499	5.405
8	0.706	1.397	1.860	2.306	2.896	3.355	5.041
9	0.703	1.383	1.833	2.262	2.821	3.250	4.781
10	0.700	1.372	1.812	2.228	2.764	3.169	4.587
11	0.697	1.363	1.796	2.201	2.718	3.106	4.437
12	0.695	1.356	1.782	2.179	2.681	3.055	4.318
13	0.694	1.350	1.771	2.160	2.650	3.012	4.221
14	0.692	1.345	1.761	2.145	2.624	2.977	4.140
15	0.691	1.341	1.753	2.131	2.602	2.947	4.073
16	0.690	1.337	1.746	2.120	2.583	2.921	4.015
17	0.689	1.333	1.740	2.110	2.567	2.898	3.965
18	0.688	1.330	1.734	2.101	2.552	2.878	3.922
19	0.688	1.328	1.729	2.093	2.539	2.861	3.883
20	0.687	1.325	1.725	2.086	2.528	2.845	3.850
21	0.686	1.323	1.721	2.080	2.518	2.831	3.819
22	0.686	1.321	1.717	2.074	2.508	2.819	3.792
23	0.685	1.319	1.714	2.069	2.500	2.807	3.767
24	0.685	1.318	1.711	2.064	2.492	2.797	3.745
25	0.684	1.316	1.708	2.060	2.485	2.787	3.725
26	0.684	1.315	1.706	2.056	2.479	2.779	3.707
27	0.684	1.314	1.703	2.052	2.473	2.771	3.690
28	0.683	1.313	1.701	2.048	2.467	2.763	3.674
29	0.683	1.311	1.699	2.045	2.462	2.756	3.659
30	0.683	1.310	1.697	2.042	2.457	2.750	3.646
40	0.681	1.303	1.684	2.021	2.423	2.704	3.551
60	0.679	1.296	1.671	2.000	2.390	2.660	3.460
120	0.677	1.289	1.658	1.980	2.358	2.617	3.373
∞	0.674	1.282	1.645	1.960	2.326	2.576	3.291

Figure B.7: Value of student's t for use in equation 4.7